

Energy efficiency and
the quality of energy
in the food processing industry



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Energy efficiency and the quality of energy in the food processing industry

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Preface

The production of foods involves a considerable use of energy and materials. Common processes in the food processing industry are cooling, freezing and heat transfer. To improve the efficiency and sustainability of processes, it is important to know the losses occurring in these processes and their causes. This can be done by taking into account the quality of energy. According to the quality of energy concept, e.g. work is more valuable than heat, because work can be transformed completely to heat but not vice versa. The quality of energy (also: exergy) appears to be a valuable tool in improving the energy efficiency and sustainability of processes.

The book starts with the analysis of a dairy factory as a case study. The second chapter of the book explains the significance of the quality of energy for optimizing processes, followed by the causes and reduction of exergy losses in Chapter 3. This theory is applied to refrigerated facilities in Chapter 4 and to the refrigeration and heat pumping in food processing plants in Chapter 5. Chapter 6 deals with exergy efficiency and exergy loss visualization in value diagrams. The exergy losses due to fluid flow and heat transfer are presented in Chapter 7.

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1 Energy analysis of a dairy factory

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For a leading dairy factory in the Netherlands, KWA has performed an energy saving study, pinch analysis and a global exergy analysis. The dairy factory produces cheese. A by-product of this process is whey.

The production process consists of several processes:

- standardization
- pasteurization
- curdled milk preparation
- pressing of cheese
- pickle baths
- cheese ware house
- whey evaporation

In the dairy sector thermal vapour compression is often used for evaporation. In this dairy factory mechanical vapour recompression is applied.

The dairy factory uses electricity and natural gas for all utilities, processes and buildings. The energy balance shows the main users of heat and electricity, namely: pasteurization, whey evaporation, air conditioning of the cheese ware house and refrigeration. Extra attention has been paid to these main users in the energy saving study. One of the measures, which were the results of the energy saving study, was co-generation and heating of the air conditioning of the cheese ware house by a heat pump.

The pinch analysis was performed for the heat exchangers network of the pasteurization. The pinch analysis shows that the present heat exchangers network were reasonably optimal. The global exergy analysis was performed for the whole dairy factory. The focus in the exergy analysis was drawn to the generation of steam, whey evaporation and refrigeration.

2 The quality of energy

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2.1 Introduction

Characteristic for our society is a massive consumption of goods and energy. Continuation of this way of life in the long term is only possible if the use of raw materials and fuel will be reduced considerably. This will require that goods and useful energy are produced in a more sustainable way. Waste of raw material and energy should be avoided and losses during conversion and transport need to be reduced as far as possible. The aim for a sustainable production of goods and ready for use energy is a great challenge of our future society.

For this ambition thermodynamics (= the science of energy conversion) will show the direction we have to go. In case of energy conversion this is obvious. The purpose of such systems is the conversion of energy from one form into the other. In case of chemical processes this seems to be less obvious. Since the properties of matter are completely determined by their energetic conditions, thermodynamics will provide the basis for the analysis and optimization of the chemical conversion of matter. Through applying thermodynamics for energy conversion systems as well as chemical processes the minimal amount of energy required can be determined. Moreover thermodynamics are useful to show where losses do occur and to determine their magnitude. To make clear how the quantity “exergy” can be used for this purpose examples that are concerned with energy conversion are shown mainly. The same considerations are applicable to chemical processes.

Why the introduction of an additional quantity “exergy” and what is the meaning of this quantity? In practice thermodynamics are often used to determine the performance of processes and plants. In case of chemical processes this implies determining the required amount of raw material and energy per unit product; in case of energy conversion plants, determining conversion efficiencies is the central issue. Determining the performance in these cases can be easily done without using the quantity exergy. However the application of the quantity exergy will be useful if differences in performance between alternative plants have to be explained or if plants have to be optimised. To illustrate this, Table 2.1 shows characteristic thermal efficiencies for a few conversion options, assuming natural gas as the primary fuel.

Table 2.1 *Thermal efficiencies for a few conversion options*

Plant	efficiency
electric power station	55 %
central heating boiler	95 %
combined heat and power (CHP) plant	80 %

It appears that power production with an electrical power plant leads to much lower conversion efficiencies than the production of hot water in a central heating boiler. Also in case of combined production of heat and power in so-called CHP plants, higher conversion efficiencies can be achieved. These efficiencies are determined by using amounts of energy only. However the second law of thermodynamics learns that energy is not only characterized by its amount but also by its quality. This quality is very important in case of energy conversion, as loss of quality can occur during conversions without immediate loss of energy to the environment. In thermodynamics the property “entropy” has been introduced to enable quantifying losses in quality. Entropy however is not a direct measure for the quality of energy, while such a direct measure is very useful for assessing and optimizing energy conversion systems. The quality of energy can be indicated in a for engineers comprehensible way as “the potential to convert energy into work”: this is the amount of work that can be obtained from an amount of energy under ideal conversion conditions (using reversible processes). This potential to produce work is called “exergy”. The quantity exergy originates from the application of thermodynamics on energy conversion systems and must not be seen as a quantity of classical thermodynamics. The importance of this quantity will become clear during analyzing and optimizing energy systems as well as chemical processes.

The quantity exergy is based of course on thermodynamic concepts. To determine exergy values as well as to explain differences in exergy values and exergy losses, basic knowledge of thermodynamics is indispensable. Before focusing on the quantity exergy the necessary thermodynamic concepts will be discussed.

2.2 Thermodynamic concepts

2.2.1 Main laws of thermodynamics

Thermodynamics are based on experience, experience with nature that shows which conversions from one kind of energy into the other are possible and which are not. In the following, several kinds of energy will play a role like: kinetic energy, potential energy, internal energy, heat, work, electrical energy and chemical energy. The most well known statements about these conversions, based on experience with nature, are defined as the first and second law of thermodynamics. The first law of thermodynamics says:

“energy can’t be lost, neither it can be obtained from nothingness”

This law indicates that for an arbitrary system the difference between the amount of energy that is transferred into the system and the amount that is transferred from the system, should equal the amount that is accumulated in the system during the process. This law makes it possible to draw up energy balances for any arbitrary system.

This first law, also indicated as the conservation of energy principle, shows in fact the similarity between the different kinds of energy. However this law cannot be applied without considering other phenomena during energy conversion. Different kinds of energy not only exhibit similarities but also differences; not every energy conversion is

possible if the first law of thermodynamics is satisfied. Nature shows that specific kinds of energy can be converted in every other kind of energy, but there are also kinds of energy for which this is not true. This experience is defined in the second law of thermodynamics. Carnot has formulated this characteristic of nature in the middle of the nineteenth century as follows:

“work can always be converted completely into heat, but heat cannot always be converted entirely into work”

Also Clausius, Thomson and Planck have formulated this characteristic of nature each in his own words. Actually the meaning of all these formulations is the same. They can be summarized as follows:

“work is a kind of energy that can be converted completely into heat by any arbitrarily process; however it is impossible to design a process that is able to convert the produced heat completely into work in order to restore the initial state entirely”

The second law of thermodynamics shows that work can be converted completely in any other kind of energy but that this is not the case with heat. In our strongly mechanized society, that consumes work on a large scale, work is more valuable than heat. Therefore based on the second law of thermodynamics the value of an amount of energy is supposed to equal the maximum amount of work that can be obtained from this amount of energy.

The meaning of these thermodynamic laws can be illustrated by considering an arbitrarily closed system for the production of work as shown in Fig. 2.1. Within this system a power cycle takes place that receives an amount of heat Q_1 from a hot reservoir at temperature T_1 and discharges an amount of heat Q_2 to a cold reservoir at temperature T_2 . It is assumed that the temperature at which heat is transferred into or from the system is the same as the temperature of the corresponding reservoir. In case of a reversible power cycle the produced power is called W_{rev} . The efficiency of the reversible process is defined as:

$$\eta = \frac{W_{\text{rev}}}{Q_1} \quad (2.1)$$

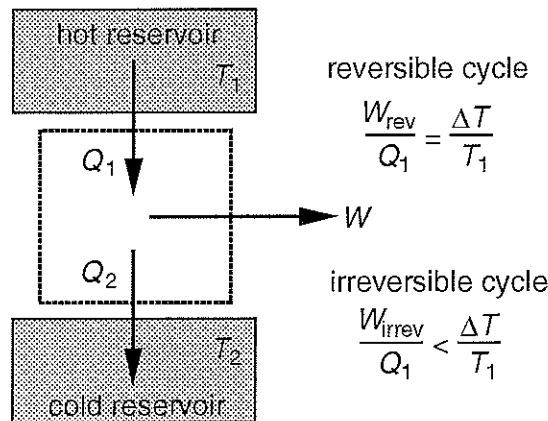


Figure 2.1 Application of the second law of thermodynamics on a closed power cycle

It has been proved that the efficiency of such a reversible power cycle only depends on the temperature of both heat reservoirs. This has resulted in the definition of the thermodynamic temperature scale. This temperature scale has been chosen such that the efficiency of a reversible power cycle equals the ratio between the temperature difference ($\Delta T = T_1 - T_2$) of the reservoirs and the temperature T_1 at which heat is transferred into the power cycle:

$$\eta = \frac{W_{\text{rev}}}{Q_1} = \frac{\Delta T}{T_1} \quad (2.2)$$

Using the thermodynamic temperature scale the amount of work from a reversible power cycle can be determined with:

$$W_{\text{rev}} = \left(1 - \frac{T_2}{T_1}\right) \cdot Q_1 \quad (2.3)$$

The term $(1 - T_2/T_1)$ is often called the Carnot-efficiency. Carnot has derived this equation assuming a reversible cycle containing an ideal gas and applying the ideal gas temperature scale. Today this equation is defined by the choice of the thermodynamic temperature scale. The term $(1 - T_2/T_1)$ indicates the maximum amount of work that can be obtained from the amount of heat Q_1 transferred into the system while using a cold reservoir at temperature T_2 .

2.2.2 The property enthalpy and entropy

Thermodynamic properties are used to define the thermodynamic state of a system. A short discussion of the properties enthalpy and entropy is presented here as these properties are in particular important to determine the exergy of an amount of matter.

The property enthalpy of an arbitrary amount of substance is defined as:

$$H = U + p \cdot V \quad (2.4)$$

The enthalpy of an amount of substance can thus be written as the sum of the internal energy and the product of pressure and volume of the substance. Differentiating this equation gives:

$$dH = dU + p \cdot dV + V \cdot dp \quad (2.5)$$

By applying the first law on a compressible substance in a closed system undergoing a reversible process Eq. (2.6) can be derived. This equation says that the amount of heat, supplied as the result of an infinitesimal change of state, equals the sum of the change of internal energy of the substance and the product of pressure and the change in volume:

$$dQ = dU + p \cdot dV \quad (2.6)$$

From the Eqs. (2.5) and (2.6) it results that the relation between the change in enthalpy and the supplied heat to the substance is:

$$dQ = dH - V \cdot dp \quad (2.7)$$

It appears that as long as the pressure of the substance is constant, the change in enthalpy equals the amount of heat supplied to the substance.

Another property, closely related to the supplied heat, is entropy. To clarify the meaning of this property the system shown in Figure 2.1 will be used. In thermodynamics it is postulated that heat is positive as it is transferred into the considered system. Furthermore it is assumed that work has a positive sign as it is produced by the system and delivered to the environment. As a consequence of the first law of thermodynamics it must be that:

$$W_{\text{rev}} = Q_1 + Q_2 \quad (2.8)$$

In this case the value of Q_2 will be negative. By rewriting Eq. (2.2) it appears that:

$$\frac{Q_1 + Q_2}{Q_1} = \frac{T_1 - T_2}{T_1} \quad (2.9)$$

This equation can be reorganized resulting in:

$$\frac{Q_1}{T_1} + \frac{Q_2}{T_2} = 0 \quad (2.10)$$

Up to now it is assumed that the system in Fig. 2.1 undergoes a reversible power cycle. The amount of work that can be obtained with an irreversible power cycle from the same amount of heat Q_1 will always be less than with the reversible process, thus:

$$\frac{W_{\text{rev}}}{Q_1} < \frac{\Delta T}{T_1} \quad (2.11)$$

In a similar way as has been done for the reversible case for the irreversible it can be derived that:

$$\frac{Q_1}{T_1} + \frac{Q_2}{T_2} < 0 \quad (2.12)$$

To obtain one general applicable equation, for reversible as well as irreversible processes, the Eqs. (2.10) and (2.12) can be combined into the equation:

$$\sum \frac{Q}{T} \leq 0 \quad (2.13)$$

In case of systems undergoing a cycle that transfers heat at changing temperature levels one can prove that the following equation applies:

$$\oint \frac{dQ}{T} \leq 0 \quad (2.14)$$

Suppose that a closed system undergoes a process that brings the from condition 1 to condition 2 system in a reversible way and after that in a different but also reversible way from condition 2 to condition 1. The closed system then undergoes a reversible cycle, for which the following equation is valid:

$$\int_{1 \text{ (rev)}}^2 \frac{dQ}{T} + \int_{2 \text{ (rev)}}^1 \frac{dQ}{T} = 0 \quad (2.15)$$

This equation is valid for any other reversible way through which the closed system goes from condition 1 to condition 2. From this it may be concluded that $\int_1^2 dQ/T$ during a reversible transition from condition 1 to condition 2 always will have the same value. This means that the system has a property for which the change in value during transition from condition 1 to condition 2 is determined by summing up dQ/T over the covered way. This property is called the entropy and is represented by the symbol S :

$$S_2 - S_1 = \int_1^2 \frac{dQ}{T} \quad \text{or} \quad dS = \frac{dQ}{T} \quad (2.16)$$

2.2.3 The meaning of ΔS_{irrev}

Suppose that the closed system undergoes a cycle where the transition from condition 1 to condition 2 is irreversible, while the transition from condition 2 to condition 1 is reversible. As the total cycle is irreversible it applies that:

$$\int_{1 \text{ (irrev)}}^2 \frac{dQ}{T} + \int_{2 \text{ (rev)}}^1 \frac{dQ}{T} < 0 \quad (2.17)$$

Since the transition from condition 1 to 2 is supposed to be reversible, from Eq. (2.16) it follows:

$$\int_{2 \text{ (rev)}}^1 \frac{dQ}{T} = S_1 - S_2 \quad (2.18)$$

As a consequence of Eqs. (2.17) and (2.18) it results:

$$\int_{1 \text{ (irrev)}}^2 \frac{dQ}{T} < S_2 - S_1 \quad (2.19)$$

For this equation can also be written:

$$S_2 - S_1 = \int_{1 \text{ (irrev)}}^2 \frac{dQ}{T} + C \quad (2.20)$$

In this equation C must be positive. The size of C depends on the extent of the irreversibilities during transition from condition 1 to condition 2. In case of a highly irreversible transition C will have a high value; the value of C will be low in case of a little irreversible transition. The value of C will become 0 only in case of a reversible transition. Therefore C can be seen as a contribution of the irreversibilities to the entropy increase and is called the irreversible entropy increase (or entropy production) indicated as ΔS_{irrev} . So, from the second law it appears that any closed system can be characterized by a property “entropy”; the change of this property during a transition from condition 1 to condition 2 results from the following equation:

$$S_2 - S_1 = \int_1^2 \frac{dQ}{T} + \Delta S_{\text{irrev}} \quad (2.21)$$

On a differential basis this equation takes the form:

$$dS = \frac{dQ}{T} + dS_{\text{irrev}} \quad (2.22)$$

To derive this equation it is assumed that the heat flow dQ is positive if this flow is transferred into the system. The temperature T in this equation is the temperature of the heat flow at the system border. By applying this equation these assumptions have to be considered carefully.

2.3 Exergy

2.3.1 Definition of the quantity exergy

In the introduction the quantity exergy is indicated as “the potential to convert energy into work”. Obviously the meaning of this quantity is limited to kinds of energy for which the potential to produce work does not equal the energy content, that is kinds of energy that can not be converted fully into work. From the types of energy mentioned in section 2.2.1 the kinetic energy, potential energy and electrical energy can be converted completely into work by applying reversible conversion systems. This means that the exergy content of these kinds of energy is the same as the energy content. So the quantity exergy is mainly important for heat, internal energy and chemical energy. Heat, work and flows of matter have to be considered for the analysis of chemical plants and energy conversion plants. An exergy analysis of these types of plants will require

exergy values for heat and flows of matter. In this section equations will be derived that can be used to determine these exergy values.

In general it may be expected that a thermodynamic system which is not in equilibrium with its environment, has the potential to produce work. In this respect an amount of matter can be seen as a thermodynamic system. When such a system is brought into equilibrium with the environment it will be able to produce work and thus the potential of the system to produce work will be reduced. The maximum amount of work will be derived when the system is brought into equilibrium with the environment in a reversible way. This maximum amount of work is defined as the exergy of the system. As a consequence of this definition it is obvious that the exergy of a system is zero when it is fully in equilibrium with the environment.

2.3.2 The exergy of heat

In section 2.2.1 it is concluded that an amount of heat derived from a hot reservoir at temperature T_1 , can only partly be converted into work. The remainder is transferred to a heat reservoir at a lower temperature. The produced work is maximal in case of a reversible cycle. Suppose that an amount of heat dQ at temperature T is supplied to a reversible power cycle that uses the environment at temperature T_0 as the cold reservoir. The work obtained from this cycle becomes:

$$dW_{\text{rev}} = \left(1 - \frac{T_0}{T}\right) \cdot dQ \quad (2.23)$$

The remaining heat is transferred to the environment at temperature T_0 . This heat cannot exchange energy with the environment any longer and is therefore supposed to be in (thermal) equilibrium with the environment. This means that the amount of work dW_{rev} produced by the cycle is the exergy dEx of the quantity of heat dQ .

In practice heat is usually extracted from fluid flows that are cooled down during heat transfer. Extracting heat then occurs under continuously changing temperature conditions. When the condition of the heat source changes from condition 1 to condition 2 during a heat transfer process, the exergy of the heat derived from the heat source becomes:

$$Ex_Q = \int_1^2 dEx_Q = \int_1^2 \left(1 - \frac{T_0}{T}\right) \cdot dQ \quad (2.24)$$

This equation can be used generally to determine the exergy of an amount of heat.

2.3.3 The exergy of an amount of matter

The exergy of an amount of matter is defined by the amount of work that will be obtained from a system that brings the matter into equilibrium with the environment by reversible processes. To determine the amount of work unambiguously the system for the conversion of the matter must be defined more accurately. It is agreed that such a system should have the following characteristics:

- a) only reversible processes take place in the system

- b) the system is open, has a constant volume and a steady flow is entering and leaving the system
 - c) the kinetic and potential energy of the matter is not taken into consideration (is assumed to remain unchanged)
 - d) heat can be transferred only from or to the environment (at temperature T_0)
 - e) for the conversion of matter into environmental components matter from the environment (at environmental conditions) should be used only
 - f) at the outlet of the system, matter is entirely in equilibrium with the environment
- The exergy of an amount of matter is defined as the net amount of work that is obtained when the amount of matter has passed such a system.

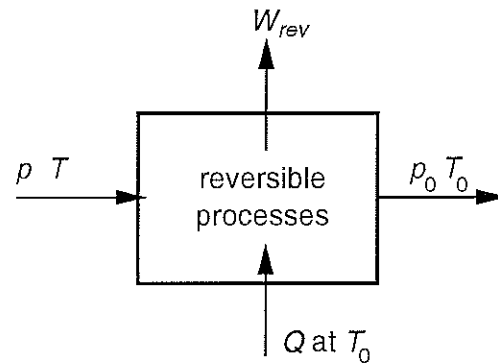


Figure 2.2 System with reversible processes that brings an amount of matter into equilibrium with the environment

First the exergy of an amount of matter will be determined without considering changes in the chemical composition of this matter. As will be explained later the exergy derived in this way is also called the thermo-mechanical exergy.

Figure 2.2 shows a system that brings an amount of matter in (thermo-mechanical) equilibrium with the environment. This means that the matter leaves the system at environmental temperature and pressure.

Applying the first law to the system shown in Figure 2.2 and neglecting the changes in kinetic and potential energy of the matter at inlet and outlet of the system it becomes:

$$Q = (H_0 - H) + W_{rev} \quad (2.25)$$

Using Eq. (2.22) for a reversible process gives:

$$dQ = T \cdot dS$$

As the system can only exchange heat with the environment, for this heat can be written:

$$Q = T_0 \cdot (S_0 - S) \quad (2.26)$$

An equation for work produced by the system can be obtained by combining Eqs. (2.25) and (2.26):

$$W_{\text{rev}} = (H - H_0) - T_0 \cdot (S - S_0) \quad (2.27)$$

This represents the work obtained when the amount of matter is brought into equilibrium by applying a system as defined for determining the exergy of matter. Thus the exergy of an amount of matter can be determined with Eq. (2.27):

$$Ex = (H - H_0) - T_0 \cdot (S - S_0) \quad (2.28)$$

At the right side of this equation only properties of state are used. Some of these properties are depending on the condition of the environment. Only after defining the environment unambiguously the quantity exergy can also be seen as a property.

Equation (2.28) can be used to determine the exergy of an amount of matter. In case of a fluid flow an amount of matter is passing per unit of time. The exergy of a fluid flow then becomes:

$$Ex_{\text{flow}} = \Phi_m \cdot (h - h_0) - T_0 \cdot (s - s_0) \quad (2.29)$$

2.3.4 Exergy losses

The driving forces applied in the considered processes determine the magnitude of exergy losses. On the other hand driving forces are determining equipment size and the time processes need to develop. Therefore exergy losses have to be weighted against equipment size and capital costs. Finally operational costs and other relevant performance factors, like e.g. emissions, have to be balanced against capital costs. During design and optimization of plants, the determination of exergy losses will be very useful, because they show how much exergy, or potential to produce work, will be lost in a specific unit of the plant. At normal operation many processes in large plants can be considered to take place in open, steady flow systems as e.g. in heat exchangers, compressors, turbines, furnaces or reactors. To derive a general equation for the exergy loss of such processes Fig. 2.3 will be used. It is assumed that the same amount of matter that enters the system will also leave the system during a considered period of time.

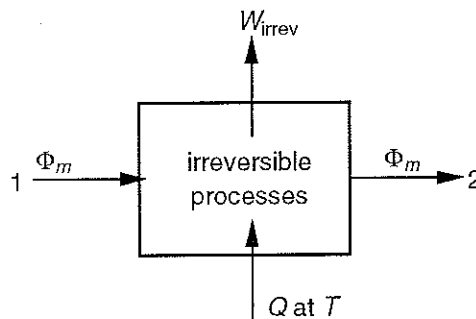


Figure 2.3 Open system with steady flow

The exergy loss in the system due to the irreversible processes can be determined by using the exergy balance of the system:

$$Ex_{\text{loss}} = Ex_{\text{in}} - Ex_{\text{out}} \quad (2.30)$$

The exergy loss of the system in Fig. 2.3 can be determined with this equation if the exergy flows entering and leaving the system are known. Exergy is entering the system with mass flow 1, with exergy Ex_1 , and a heat flow at temperature T . Exergy leaves the system with mass flow 2, with exergy Ex_2 , and the flow of work W_{irrev} . Equation (2.30) can then be written as:

$$Ex_{\text{loss}} = Ex_1 + \int_1^2 \left(1 - \frac{T_0}{T}\right) \cdot dQ - Ex_2 - W_{\text{irrev}} \quad (2.31)$$

The exergy change of the flow of matter passing the system can be calculated using equation (2.29):

$$Ex_1 - Ex_2 = (H_1 - H_2) - T_0 \cdot (S_1 - S_2) \quad (2.32)$$

As it may be assumed that changes in potential energy and kinetic energy can be neglected, the energy balance of the system in Fig. 2.3 can be written as:

$$H_2 - H_1 + W_{\text{irrev}} - Q = 0 \quad (2.33)$$

Combining Eqs. (2.31), (2.32) and (2.33) delivers the following equation for the exergy loss of the system:

$$Ex_{\text{loss}} = T_0 \cdot \left((S_2 - S_1) - \int_1^2 \frac{dQ}{T} \right) \quad (2.34)$$

The system in Fig. 2.3 converts an amount of matter at state 1 in an amount of matter at state 2. The entropy change due to this conversion can be determined by applying Eq. (2.21). Combining this equation with Eq. (2.34) gives:

$$Ex_{\text{loss}} = T_0 \cdot \Delta S_{\text{irrev}} \quad (2.35)$$

From this equation it appears that the exergy loss in a open system with steady flow equals the product of ambient temperature and the irreversible entropy increase (also called “entropy production”) of the flow of matter through the system. Equation (2.35) is, in a somewhat different form, also called the Gouy-Stodola-equation.

2.4 Dividing exergy in thermo-mechanical and chemical exergy

The exergy of an amount of matter is determined by using a system that brings the matter in equilibrium with the environment. Full equilibrium with the environment is achieved when the temperature of the considered amount of matter is the same as the temperature of the environment and the matter is fully converted into components that

are available in the environment while the pressure of the separate components equals the partial pressure of the respective component in the environment.

For the calculation of the exergy of an amount of matter it is required to select a system that will be able to bring the matter in full equilibrium with the environment. In order to derive useful equations it is necessary to divide the required conversion system in a number of sub-systems for which the produced work easily can be determined. Then the total work delivered by all sub-systems together equals the exergy of the amount of matter. Dividing the conversion system in sub-systems is in particular necessary for the calculation of the exergy of mixtures. A first division is made here to distinguish between the thermo-mechanical exergy and the chemical exergy. Assuming that a gas mixture is available at a pressure and temperature that differ from the environmental conditions, the mixture will be brought at environmental conditions. An equation for calculating the work generated by performing this conversion with a suitable system is derived in section 2.3.3. The work from this conversion is called the thermo-mechanical exergy as it brings the mixture in thermo-mechanical equilibrium with the environment. During this conversion the composition of the gas mixture will remain unchanged. Next the mixture must be brought into full equilibrium with the environment by converting the mixture isothermally into environmental components at the partial pressure of the respective components in the environment. The work generated during this conversion is called the chemical exergy, because it is the maximum amount of work that can be derived by changing the chemical composition of the mixture keeping temperature and total pressure constant. Obviously all sub-systems together have to fulfil the characteristics as described in section 2.3.3. The exergy of the mixture equals the sum of the mechanical exergy and chemical exergy, thus:

$$Ex_{\text{mixture}} = Ex_{\text{tm}} + Ex_{\text{ch}} \quad (2.36)$$

In the following a procedure is presented for the calculation of the chemical exergy of gas mixtures. In this procedure the calculation of the exergy of a gas mixture occurs in two steps:

1. determining the exergy of a gas mixture based on known exergy values for the respective elements at standard pressure and temperature;
2. determination of the exergy of the elements based on a predefined environmental composition.

An overview of the necessary sub-systems for the first step is shown in Fig. 2.4. In order to convert a gas mixture into the respective elements, the consisting components must be separated first; this occurs in system 2 of Fig. 2.4. Therefore it is supposed that the components can be isolated from the mixture by applying reversible membranes through which only one specific component can pass without friction while all other components are hold up. After passing the membrane a component is available at a pressure that equals its partial pressure in the mixture. Then this separation will occur without the need for any work. After separation the components have to be pressurized isothermally and reversibly till environmental pressure. In these conditions it is assumed that components will fulfil the ideal gas laws. Then the work required for the compression can be calculated using the following equation:

$$W_{\text{rev}} = -\int_1^0 V \cdot dp = -\int_1^0 \frac{R \cdot T_0}{p} \cdot dp = -R \cdot T \cdot \int_1^0 \frac{dp}{p} \quad (2.37)$$

Using this equation the work necessary for the separation of one mole of mixture (the work done by sub-system 2) becomes:

$$W_{\text{rev},2} = \sum_i -y_i \cdot R_m \cdot T_0 \cdot \ln \frac{p_0}{y_i \cdot p_0} \quad (2.38)$$

In this equation y_i is the mole fraction of the respective components in the gas mixture.

In a second step, sub-system 3a, the isolated components are reversibly dissociated into their elements. The reactions necessary for this conversion are reverse formation reactions. As the work from a formation reaction equals the negative value of the change in the Gibbs-free energy, the work for the dissociation of one mole of a component becomes:

$$W_{\text{rev}} = \Delta_f g_{\text{mol},298}^0 \quad (2.39)$$

Then for the work from the dissociation reactions to convert all components from one mole mixture, it can be written:

$$W_{\text{rev},3a} = \sum_i y_i \Delta_f g_{\text{mol},i,298}^0 \quad (2.40)$$

After sub-system 3a all compounds in the gas mixture are converted into their elements at standard pressure and temperature. As it is assumed that the exergy of these elements is known, the chemical exergy of the mixture can be calculated by adding up the different contributions:

$$ex_{\text{mol},\text{ch}} = W_{\text{rev},2} + W_{\text{rev},3a} + \sum_i v_i \cdot ex_{\text{mol},\text{element}} \quad (2.41)$$

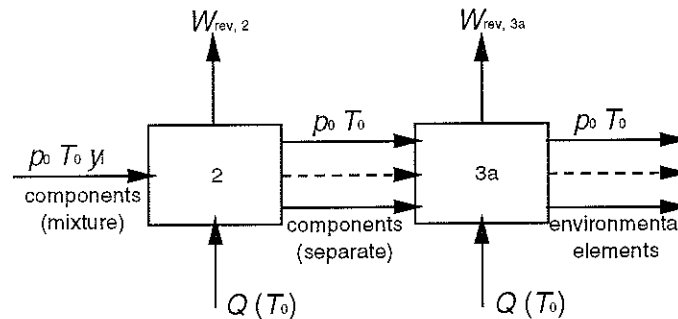


Figure 2.4 Overview of sub-systems for the calculation of the exergy of a gas mixture, assuming that the exergy of available elements is known

In equation 2.41 it is assumed that the molar exergy of the elements is known. The exergy values of the elements will be determined here in a second step. Therefore the element must be converted reversibly and isothermally into a compound or element that exists in the (defined) environment. In Fig. 2.5 the necessary sub-systems for the calculation of the exergy of an element are shown. For the conversion of elements into an environmental compound, only matter from the environment can be used.

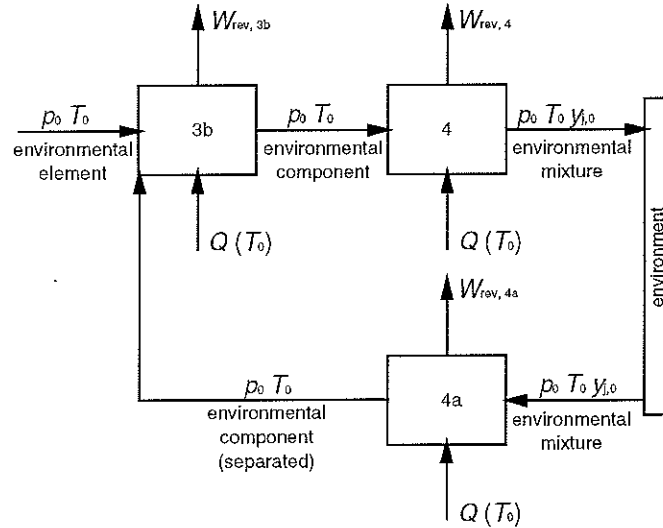


Figure 2.5 Overview of sub-systems for the calculation of the exergy of elements

In order to bring an element in full equilibrium with the environment, it must be converted first in an environmental component if the element does not exist as such in the environment. If for instance the surrounding air is used as the defined environment to determine exergy values, the elements carbon (C) and hydrogen (H₂) must be converted into CO and H₂O respectively by using oxygen from the environment. In case of elements like O₂ and N₂, which do exist as such in the environment, no conversion is necessary. The conversion of the element into an environmental compound occurs in sub-system 3b (see figure 2.5) and is actually the formation reaction of the environmental compound. The work from a formation reaction equals the negative value of the change in Gibbs-free energy. As the formation reaction of one mole of compound requires ν mole of the considered element, the work from the conversion in sub-system 3b becomes:

$$W_{\text{rev},3b} = -\frac{1}{\nu} \cdot \Delta_f g_{\text{mol},298}^0 \quad (2.42)$$

As the conversion of 1 mole of the considered element requires ν_0 mole of an environmental component, this component has to be separated from the environmental mixture at its partial pressure and reversibly compressed till standard pressure. This extraction of the component is supposed to occur in sub-system 4a. The work from this sub-system, the work from compression, is:

$$W_{\text{rev},4a} = -\nu_0 \cdot R_m \cdot T_0 \cdot \ln \frac{y_j \cdot P_0}{p_0} \quad (2.43)$$

In this equation y_j is the mole fraction of the respective component in the environment.

The compound resulting from the formation reaction in sub-system 3b becomes available at standard pressure. The reversible discharge of this compound to the environment requires that it first must be expanded to its partial pressure in the environment. After this expansion it will be discharged without pressure loss to the environment via a reversible membrane. The reversible work from the isothermal expansion becomes:

$$W_{\text{rev},4} = -\frac{1}{\nu} \cdot R_m \cdot T_0 \cdot \ln \frac{P_0}{y_j \cdot P_0} \quad (2.44)$$

The exergy of the element then equals the reversible work from the sub-systems necessary for the reversible conversion of the element into an environmental component, so:

$$ex_{\text{mol,element}} = W_{\text{rev},3b} + W_{\text{rev},4} + W_{\text{rev},4a} \quad (2.45)$$

With the Eqs. (2.42) till (2.45) the chemical exergy of one mole element can be calculated, provided that a composition of the environment consisting of gaseous components is defined and that at least one of these components contains the considered element.

It must be noticed that some authors apply a further subdivision of the chemical exergy into chemical exergy and mixing exergy. Here the chemical exergy includes the chemical exergy as well as the mixing exergy. Dividing these terms has only limited advantages, but can result in serious confusion with regard to the accurate definition of the mixing exergy.

3 What about losses?

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3.1 Introduction

In the context of the first law of thermodynamics, it seems strange to speak about exergy losses. Doesn't this law stress the conservation of energy? We will try to develop some feeling about this point by discussion of a simplified example of a counter current heat exchanger. From this we will see what actually is lost and what causes these losses. We continue with a simple and a more complete derivation of the relation between losses occurring in all real processes and their causes. When we understand what is lost and how that is related to the operating conditions of our processes, we are perhaps able to meaningfully reduce these losses in order to improve our processes.

3.2 A heat exchange process

Consider a counter current heat exchanger where a hot and a cold stream exchange thermal energy, the hot stream cools down and the cold stream heats up. This type of energy transfer is very common in the process industry. At a certain point in the heat exchanger, the hot stream has a temperature T_h and the cold stream has a temperature T_c . In this case the driving force for thermal energy transfer is usually taken as the temperature difference $T_h - T_c$. In order to transfer an amount of thermal energy in a reasonable amount of time we have to design the heat exchanger such that the temperature difference between both streams remains large enough. We, for example, decide about the minimum temperature difference that we allow to occur in the heat exchanger, and this has its consequences for the heat exchange surface area needed. Assume that from the hot stream at T_h , a small amount of thermal energy δQ per second is transferred to the cold stream at T_c , and that $T_c > T_0$, the temperature of the environment. In the best case we have no thermal energy leaks to the environment. The first law of thermodynamics is certainly not violated by the process we consider, because the thermal energy abstracted from the hot stream is completely transferred to the cold stream. Is it possible to speak about losses in this case?

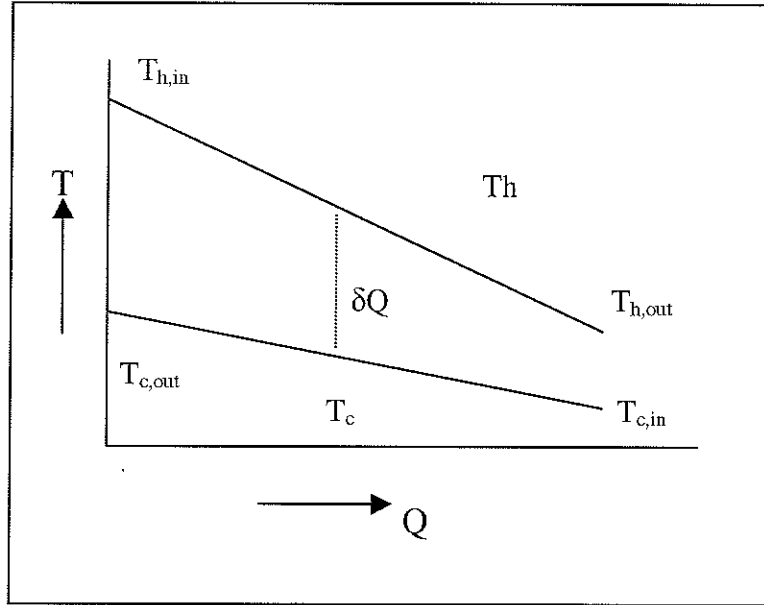


Figure 3.1 A heat exchange process

We will now consider this heat exchange process from a completely different point of view. Assume that we want to produce the maximum amount of power from an amount of thermal energy δQ per second at the temperature T_h of the hot stream. The lowest temperature that we have available is the temperature of our environment T_0 . We can then operate a Carnot engine between T_h and T_0 , to produce the amount of work δW_h per second, given in Eq. (3.1):

$$\delta \dot{W}_h = \delta \dot{Q} \left(1 - \frac{T_0}{T_h} \right) \quad (3.1)$$

However, if the amount of thermal energy δQ per second is transferred to the cold stream at T_c , we can produce an amount of work δW_c per second in the same way as before from a Carnot engine, as shown in Eq. (3.2):

$$\delta \dot{W}_c = \delta \dot{Q} \left(1 - \frac{T_0}{T_c} \right) \quad (3.2)$$

If we subtract Eq. (3.2) from Eq. (3.1), we obtain:

$$\delta \dot{W}_h - \delta \dot{W}_c \equiv \delta \dot{W}_{lost} = T_0 \delta \dot{Q} \left(\frac{1}{T_c} - \frac{1}{T_h} \right) \quad (3.3)$$

The right hand side of Eq. (3.3) is > 0 because $T_c < T_h$ and the other terms are positive too. This means that also the left-hand side of Eq. (3.3) has to be positive. The amount of power we could have obtained from δQ per second at T_h is larger than the amount of power obtainable from δQ per second at T_c . The operation of transfer of thermal

energy from T_h to T_c is accompanied with a decrease in the amount of power obtainable from the system. The difference between δW_h and δW_c is called the amount of work lost δW_{lost} due to the process of thermal energy transfer from a higher to a lower temperature. A closer observation of Eq.3.3 shows that δQ is the amount of thermal energy transferred per second and $\Delta 1/T$, the difference between $1/T_c$ and $1/T_h$, can be interpreted as the cause, the driving force, for transfer of thermal energy. The larger $\Delta 1/T$, the larger the amount of work lost due to the process considered.

The second law of thermodynamics, applied to a control volume, can be expressed in formula form as follows:

$$\frac{dS}{dt} = \sum \dot{m}_{in} s_{in} - \sum \dot{m}_{out} s_{out} + \sum \int \frac{\delta \dot{Q}_{in}}{T} - \sum \int \frac{\delta \dot{Q}_{out}}{T} + \Delta \dot{S}_{irrev} \quad (3.4)$$

In our system, we consider two control volumes, one at T_h and one at T_c . For the first, Eq. (3.4) can be worked out, the first term is the accumulation term, which in our case of a steady state process is zero. If we have only one hot stream, the mass balance gives $\dot{m}_{in} = \dot{m}_{out}$. Assume that the amount of thermal energy transferred between the hot and cold stream δQ is so small that we reasonably can assume that both temperatures T_h and T_c remain constant, and further that the transfer process in each of the streams occurs reversible. In that case $\Delta \dot{S}_{irrev} = 0$. For the hot stream Eq. (3.4) reduces to:

$$0 = \dot{m} s_{in} - \dot{m} s_{out} - \frac{\delta \dot{Q}}{T_h} \quad (3.5)$$

The entropy change of the hot stream is the difference between the out- and ingoing entropy streams and is shown in Eq. (3.6):

$$\Delta \dot{S}_{hot} = -\frac{\delta \dot{Q}}{T_h} = \dot{m}(s_{h,out} - s_{h,in}) \quad (3.6)$$

In the same way, the entropy change of the cold stream is given by:

$$\Delta \dot{S}_{cold} = +\frac{\delta \dot{Q}}{T_c} = \dot{m}(s_{c,out} - s_{c,in}) \quad (3.7)$$

The total entropy change is equal to $\Delta \dot{S}_{irrev}$ per second, as shown in Eq. (3.8):

$$\Delta \dot{S}_{total} = \Delta \dot{S}_{hot} + \Delta \dot{S}_{cold} = \Delta \dot{S}_{irrev} \quad (3.8)$$

Eqs.(3.8), (3.6) and (3.7) can be combined to Eq. (3.9):

$$\Delta \dot{S}_{irrev} = \delta \dot{Q} \left(\frac{1}{T_c} - \frac{1}{T_h} \right) \quad (3.9)$$

From Eq. (3.9) we see that $\Delta S_{\text{irrev}} > 0$, as it should be for a real, possible, process, in accordance with the second law of thermodynamics. This example shows us that the entropy generated in the process considered is given by the product of the thermal energy stream and the driving force of that process:

$$\Delta \dot{S}_{\text{irrev}} = \delta \dot{Q} \Delta \left(\frac{1}{T} \right) \quad (3.10)$$

From Eq. (3.3) and Eq. (3.10), we see that we can write:

$$\dot{W}_{\text{lost}} = T_0 \Delta \dot{S}_{\text{irrev}} \quad (3.11)$$

Eq. (3.11) is the so-called Gouy-Stodola-relation. The power lost in a process is equal to the product of the actual temperature of the environment in which the process takes place, and the amount of entropy generated per second in the process.

What we have seen from this example is that thermal energy initially available at T_h , after transfer to a lower temperature T_c , has a lower potential to obtain work from it than it originally had. There is no loss of energy, in accordance with the first law, but we can do less with it. Before we consider other driving forces and streams that lead in an analogous way to a loss in work potential, we apply the following linear relation between the stream and the driving force to our example:

$$\delta \dot{Q} = (T_h - T_c) U dA \quad (3.12)$$

In Eq. (3.12), U is the overall heat transfer coefficient, in S.I. units $\text{Wm}^{-2}\text{K}^{-1}$, and dA is the surface area through which the thermal energy is transferred from the hot to the cold stream. Eq. (3.12) is often used in practice and has to be integrated over the whole heat exchange surface area. The driving force for thermal energy transfer is usually chosen as ΔT in stead of $\Delta 1/T$ as suggested here. From Eq. (3.12) we see that for a designed heat transfer rate, we can increase the heat transfer area, as the most reasonable option, and/or the overall heat transfer coefficient U , to reduce the driving force for heat transfer. Substitution of Eq. (3.12) into Eq. (3.3) leads to:

$$\delta \dot{W}_{\text{lost}} = (T_h - T_c) U dA \left(\frac{1}{T_c} - \frac{1}{T_h} \right) = U dA \left(\frac{(T_h - T_c)^2}{T_h T_c} \right) \quad (3.13)$$

As we can see from Eq. (3.13), the potential power lost due to heat transfer is proportional to the temperature difference squared. Reduction of ΔT by increase of dA and/or U can lead to a decrease in the amount of power lost due to transfer of thermal energy.

3.3 Towards the equilibrium state

The next example that we will consider is a rigid, completely isolated and closed system in which a flexible and permeable membrane divides the system in two parts: part 1 and part 2, initially at different conditions of pressure, temperature and chemical potentials of the substances present. We assume that no chemical reactions take place. The internal energy U of the whole system remains constant, as well as the number of moles of each component i n_i , and the volume V . Due to some transfer of the components through the membrane, both entropy values of part 1 and part 2 will change.

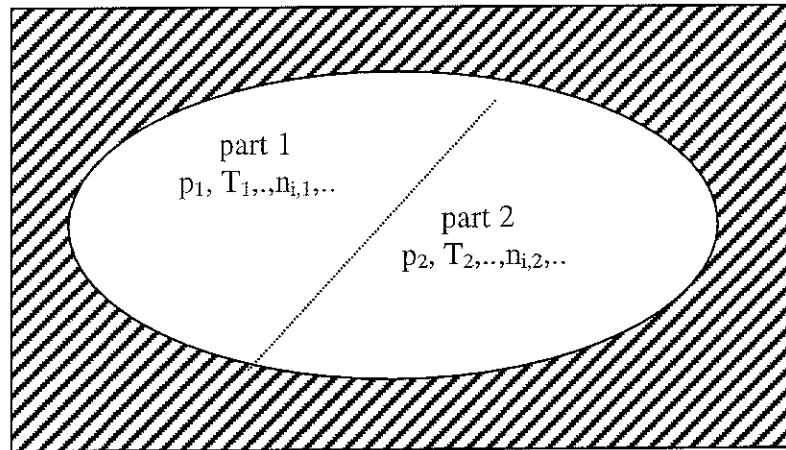


Figure 3.2 Isolated and closed system consisting of two parts

The change of part 1 is given by:

$$d\dot{S}_1 = \frac{1}{T_1} d\dot{U}_1 + \frac{p_1}{T_1} d\dot{V}_1 - \sum \frac{\mu_{i,1}}{T_1} d\dot{n}_{i,1} \quad (3.14)$$

For part 2, we can write an expression analogous to Eq. (3.14), see Eq. (3.15):

$$d\dot{S}_2 = \frac{1}{T_2} d\dot{U}_2 + \frac{p_2}{T_2} d\dot{V}_2 - \sum \frac{\mu_{i,2}}{T_2} d\dot{n}_{i,2} \quad (3.15)$$

Because U , V and each n_i for the total system remain constant, the following relations must hold:

$$d\dot{U}_1 = -d\dot{U}_2; d\dot{V}_1 = -d\dot{V}_2 \text{ and } d\dot{n}_{i,1} = -d\dot{n}_{i,2} \text{ for each } n_i \quad (3.16)$$

The total change in entropy can be obtained from the summation of Eq. (3.14) and Eq. (3.15), and subsequent substitution of the expressions from Eq. (3.16). This leads to Eq. (3.17):

$$\Delta \dot{S}_{\text{irrev}} = d\dot{S}_1 + d\dot{S}_2 = d\dot{U}_1 \left(\frac{1}{T_1} - \frac{1}{T_2} \right) + d\dot{V}_1 \left(\frac{p_1}{T_1} - \frac{p_2}{T_2} \right) - \sum \left(\frac{\mu_{i,1}}{T_1} - \frac{\mu_{i,2}}{T_2} \right) d\dot{n}_{i,1} \quad (3.17)$$

Each of the three terms on the right hand side of Eq. (3.17) must be > 0 for any real process. If $T_1 < T_2$ energy must flow from part 2 to part 1, and dU_1 must be > 0 . For the second term we have that if $p_2/T_2 > p_1/T_1$ then $dV_1 > 0$, as a consequence of the transport of matter and energy from part 2 to part 1. In the same way we have for the third term that if $\mu_{i,2}/T_2 > \mu_{i,1}/T_1$ we must have that $d\dot{n}_{i,1} > 0$. This means that component i is transported from part 2 to part 1.

We can identify in Eq. (3.17) three different driving forces and their conjugated streams: $\Delta 1/T$ and a stream of internal energy, $\Delta p/T$ and a volume stream, and finally $\Delta(\mu_i/T)$ and the mole stream n_i . As a consequence of the processes taking place in the system, T_2 decreases and T_1 increases until the temperature of both part 2 and part 1 is the same: T_f . In this case there will be no net transport of thermal energy from part 2 to part 1 any more. Also p_2 decreases and p_1 increases until the pressure in both part 2 and part 1 is the same: p_f . The conjugated volume streams become both $0 \text{ m}^3\text{s}^{-1}$. Finally also any net mole transport ceases if $\mu_{i,1} = \mu_{i,2}$ for each component i . In this situation we say that the system we considered has gone to its equilibrium state. Without any net forces a system is in the equilibrium state if everywhere in the system the temperature, the pressure and the chemical potentials of each component have a constant value. In this case there are no driving forces and resulting streams, and entropy generation stops.

We have seen examples of different types of driving forces and conjugated streams, the product of which contributes to entropy generation. The product of the temperature of the environment: T_0 , and the sum of all entropy generation contributions is equal to the loss in the amount of power the system potentially can produce. In the next paragraph, we will discuss a more general approach to the determination of the total amount of entropy generated in a system in a non equilibrium situation. This will give us insight in the causes of entropy generation and thus also in the loss in power, the exergy, as a consequence of all processes. The power that initially could have been obtained is larger than the amount of power that we finally could have produced from the system.

3.4 A microscopic view on entropy generation

Classical thermodynamics deals with macroscopic systems but several phenomena that occur in the process industry are microscopic in nature and can be better understood and modeled starting at a microscopic level of description. Irreversible thermodynamics starts from a so small microscopic level that parameters as temperature, pressure, chemical potential and so on are still well defined. This means that, especially for the gas phase, the characteristic length scale must be larger than 1

μm under the process conditions normally occurring in the process industry. The procedure often followed is to work with microscopic density values for the extensive thermodynamic properties used to describe the system studied. To calculate the total change as a consequence of the processes taking place in the system, we must integrate over the whole volume of the system. The processes in the system are driven by appropriately defined driving forces, in this case considered as gradients, and give rise to conjugated flows, in this case considered as the conjugated fluxes. As is mostly done, the entropy generation density is defined as follows:

$$\sigma \equiv \frac{\Delta \dot{S}_{\text{irrev}}}{V} \quad (3.18)$$

We expect to find an expression of the following form:

$$\sigma = \sum X_j J_j \quad (3.19)$$

In Eq. (3.19) X_j is the gradient of the driving force of type j and J_j is its conjugated flux. If the gradients are not too large, the fluxes are linearly dependent on the gradients. This is the linear regime that is extensively investigated in the area of irreversible thermodynamics. Onsager and Casimir contributed a lot to this. The driving forces can be of different tensorial character and can only couple to other driving forces of the same character. Thus scalar driving forces can only couple to other scalar driving forces. In this case flux J_j can for example be expressed as follows:

$$J_j = \sum L_{j,k} X_k \quad (3.20)$$

The constants $L_{j,k}$ in Eq. 3.20 are the so called coupling constants. Eq. (3.20) indicates that flux J_j depends in general on all gradients of the same type. These constants must be determined from experiment. Onsager has shown that $L_{j,k} = L_{k,j}$. This reduces the number of independent constants with a factor two. The constants $L_{j,k}$ are also called conductivities.

To derive an expression for σ , the entropy generation density, we start from the Gibbs expression as given in Eq. 3.21:

$$dS = \frac{1}{T} dU + \frac{p}{T} dV - \sum \frac{\mu_i}{T} dn_i \quad (3.21)$$

When we transform this expression to a relation between the density values of the extensive variables. We will use the normally used symbol with a subscript v : for example $S_v = S/V$, except for n_i/V we use c_i , the concentration of component i , Eq. (3.21) will get the following form:

$$dS_v = \frac{1}{T} dU_v - \sum \frac{\mu_i}{T} dc_i \quad (3.22)$$

The time derivative of Eq. (3.22) is given in Eq. (3.23):

$$\frac{dS_v}{dt} = \frac{1}{T} \frac{dU_v}{dt} - \sum \frac{\mu_i}{T} \frac{dc_i}{dt} \quad (3.23)$$

To derive an expression for σ from Eq. (3.23), we first have to find an expression for dc_i/dt . This derivative gives the accumulation of the concentration of component i in the small control volume that we consider to be our system. This accumulation term can be written in its most simple form as follows:

$$\frac{dc_i}{dt} = -\nabla J_i + \left(\frac{dc_i}{dt} \right)_{gen} \quad (3.24)$$

The first term on the right hand side of Eq. (3.24) is the contribution due to streams of i , to and from our system. In this term is, $\nabla = \partial/\partial x + \partial/\partial y + \partial/\partial z$, the nabla operator. In the case that we have to apply this operator to a vector, $\nabla = \mathbf{i}\partial/\partial x + \mathbf{j}\partial/\partial y + \mathbf{k}\partial/\partial z$, where \mathbf{i} , \mathbf{j} and \mathbf{k} are the unit vectors in the x -, y - and z -direction. Eventually this first term can be split in a diffusion and a convection contribution:

$$-\nabla J_i = -\nabla \dot{c}_i v_i = -\nabla \dot{c}_i (v_i - v) - \nabla \dot{c}_i v = -\nabla J_{i,dif} - \nabla J_{i,con} \quad (3.25)$$

The last term in Eq. (3.24) is the generation term of substance i in our system. Substance i can be generated as a consequence of chemical reactions in which i takes part. The conversion of a chemical reaction k is defined as:

$$\frac{d\varepsilon_k}{dt} \equiv \frac{1}{v_{k,i}} \frac{dc_{k,i}}{dt} \quad (3.26)$$

In Eq. (3.26), $v_{k,i}$ is the stoichiometric constant of substance i taking part in reaction k . This coefficient is positive for products and negative for reactants. Numerically, these constants are equal to the coefficients of the substances taking part in a chemical reaction.

If we have n different reactions in which i is converted, the last term of Eq. (3.24) can be expressed in more detail as follows:

$$\left(\frac{dc_i}{dt} \right)_{gen} = \sum_{k=1}^n v_{k,i} \frac{d\varepsilon_k}{dt} \quad (3.27)$$

The expression $d\varepsilon_k/dt$, in Eq. (3.26) and Eq. (3.27), is the speed of reaction k : r_k . Finally, Eq. (3.24) can be written as:

$$\frac{dc_i}{dt} = -\nabla J_i + \sum_{k=1}^n v_{k,i} r_k \quad (3.28)$$

Next, we focus on the determination of dU_v/dt . We will start from the first law of thermodynamics, in balance form:

$$\frac{dE_v}{dt} + \nabla \mathbf{J}_{E_v} = 0 \quad (3.29)$$

Eq. (3.29) shows clearly that the energy generation term is zero. Energy is conserved. The energy density E_v is a summation of several contributions as shown in Eq. (3.30):

$$E_v = \frac{1}{2} \sum_i c_i M_i v_i^2 + U_v + \sum_l \sum_i c_i c_{l,i} \varphi_l \quad (3.30)$$

The first term on the right hand side of Eq. (3.30) is the kinetic energy contribution, M_i is the molar mass of substance i , and v_i is the velocity of that same substance, the second term is the internal energy density without the presence of external force fields, and the last term is the potential energy contribution due to external force fields l . The last term of Eq. (3.30) is the contribution to the total energy density due to the presence of external force fields. φ_l is the potential of force field l and $c_{l,i}$ is a constant characteristic for the combination of the concentration of component i and the force field l . In case of the gravity field of the earth $c_{\text{gravity},i} = M_i$, the molar mass of substance i .

U_v is a function of temperature and the concentrations of the substances present in the system, and can be written in differential form as follows:

$$dU_v = c_{v,v} dT + \sum_i \frac{\partial U_v}{\partial c_i} dc_i \equiv c_{v,v} dT + \sum_i u_{v,i} dc_i \quad (3.31)$$

In Eq. (3.31) $c_{v,v}$ is the heat capacity density at constant density, and $u_{v,i}$ is the partial molar internal energy density of substance i . If we transfer Eq. (3.31) in its time derivative, and substitute Eq. (3.28) in that equation, we get:

$$\frac{dU_v}{dt} = c_{v,v} \frac{dT}{dt} + \sum_i u_{v,i} \left(-\nabla \mathbf{J}_i + \sum_k v_{k,i} r_k \right) \quad (3.32)$$

We will substitute Eq. (3.32) in the derivative of Eq. (3.30) with respect to time. This leads to Eq. (3.33):

$$\frac{dE_v}{dt} = \sum_i \frac{1}{2} \frac{dc_i}{dt} \left(M_i v_i^2 + \sum_l c_{l,i} \varphi_l \right) + c_{v,v} \frac{dT}{dt} + \sum_i u_{v,i} \left(-\nabla \mathbf{J}_i + \sum_k v_{k,i} r_k \right) \quad (3.33)$$

In Eq. (3.33) we can write the product $u_{v,i} \nabla \mathbf{J}_i$ as $\nabla u_{v,i} \mathbf{J}_i - \mathbf{J}_i \nabla u_{v,i}$ and for $\sum_i u_{v,i} v_{k,i} r_k$ we can write $\Delta_{r,k} u_{v,k}$, this is the change in internal energy density due to reaction k . Comparison of Eq. (3.33) with Eq. (3.29) shows that the right hand side of Eq. (3.33) is identical to $-\nabla \mathbf{J}_{E_v}$. The first term on the right hand side of Eq. (3.33) can be

identified as the time derivative of the kinetic and potential energy density contributions. Introduction of these changes in Eq. (3.33) gives:

$$-\nabla J_{E_v} = \frac{d(E_{v,kin} + E_{v,pot})}{dt} + c_{v,v} \frac{dT}{dt} + \sum_k \Delta_{r,k} u_{v,k} r_k - \sum_i (\nabla u_{v,i} J_i - J_i \nabla u_{v,i}) \quad (3.34)$$

$$\nabla J_{E_v} \equiv \nabla J_{Q_v} + \nabla \sum u_{v,i} J_i + \frac{d(E_{v,kin} + E_{v,pot})}{dt} \quad (3.35)$$

If we define:

Then, we see from Eq. (3.34) and Eq. (3.35) that we can write an expression for $-\nabla J_{Q_v}$. This expression is given in Eq. (3.36):

$$-\nabla J_{Q_v} = c_{v,v} \frac{dT}{dt} + \sum_k \Delta_{r,k} u_{v,k} r_k + \sum_i J_i \nabla u_{v,i} \quad (3.36)$$

We now write J_{U_v} explicitly, as shown in Eq. (3.37):

$$\nabla J_{U_v} \equiv \nabla J_{Q_v} + \nabla \sum u_{v,i} J_i - \frac{d(E_{v,kin} + E_{v,pot})}{dt} \quad (3.37)$$

From Eq. (3.37) it is possible to give another expression for $-\nabla J_{Q_v}$. This expression is shown in Eq. (3.38):

$$-\nabla J_{Q_v} \equiv -\nabla J_{U_v} + \nabla \sum u_{v,i} J_i - \frac{d(E_{v,kin} + E_{v,pot})}{dt} \quad (3.38)$$

After equating the right hand side of Eq. (3.36) and Eq. (3.38), the introduction of dU_v/dt from Eq. (3.32), and rearrangement of the resulting equation, we obtain:

$$\frac{dU_v}{dt} + \nabla J_{U_v} = -\frac{d(E_{v,kin} + E_{v,pot})}{dt} \quad (3.39)$$

From Eq. (3.39), we see that the internal energy density is increased by dissipation of the kinetic and potential energy density. We are now able to determine dS_v/dt from Eq. (3.23) and the expressions for dU_v/dt and dc_i/dt given in Eq. (3.39) and Eq. (3.28) respectively:

$$\frac{dS_v}{dt} = -\frac{1}{T} \nabla J_{U_v} - \frac{d(E_{v,kin} + E_{v,pot})}{dt} - \sum_i \frac{\mu_{v,i}}{T} \left(-\nabla J_i + \sum_k \nu_{k,i} r_k \right) \quad (3.40)$$

If we define: $A_{v,k} \equiv -\sum_i \nu_{k,i} \mu_{k,i}$ as the affinity of reaction k in the last term in Eq. (3.40), and rewrite the two nabla containing terms, we get the following result:

$$\frac{dS_v}{dt} = -\nabla \frac{J_{U_v}}{T} + J_{U_v} \nabla \frac{1}{T} + \nabla \sum \frac{\mu_{v,i} J_i}{T} - \sum J_i \nabla \frac{\mu_{v,i}}{T} + \sum_k \frac{A_{v,k}}{T} r_k - \frac{1}{T} \frac{d(E_{v,kin} + E_{v,pot})}{dt} \quad (3.41)$$

We will rewrite Eq. (3.41) such that the entropy production term is more clearly visible:

$$\frac{dS_v}{dt} + \nabla \left(\frac{J_{U_v}}{T} - \sum \frac{\mu_i J_i}{T} \right) = J_{U_v} \nabla \frac{1}{T} - \sum J_i \nabla \frac{\mu_{v,i}}{T} + \sum_k \frac{A_{v,k}}{T} r_k - \frac{1}{T} \frac{d(E_{v,kin} + E_{v,pot})}{dt} \quad (3.42)$$

The right hand side of Eq. (3.42) is an expression for σ , the entropy generation density. If we introduce the relation $J_{U_v} = J_{Q_v} + \sum c_i u_{v,i}$, in the left hand side of Eq. (3.42), we can transform that expression into a more well known form:

$$\frac{dS_v}{dt} + \nabla \left(\frac{J_{U_v}}{T} - \sum \frac{\mu_i J_i}{T} \right) = \frac{dS_v}{dt} + \nabla \left(\frac{J_{Q_v}}{T} + \sum \frac{J_i (u_{v,i} - \mu_{v,i})}{T} \right) \quad (3.43)$$

When we introduce the relation $u_{v,i} - \mu_{v,i} = T s_{v,i}$, we can simplify Eq. (3.43) even further:

$$\frac{dS_v}{dt} + \nabla \left(\frac{J_{U_v}}{T} - \sum \frac{\mu_i J_i}{T} \right) = \frac{dS_v}{dt} + \nabla \left(\frac{J_{Q_v}}{T} + \sum J_i s_{v,i} \right) \quad (3.44)$$

After the equal sign of the expression given in Eq. (3.44), we first see the accumulation term of entropy and then, in the second term, the transport of entropy by the flux of thermal energy divided by T , and in the third term, the contribution of all substances to the entropy of the streams.

In Eq. (3.42), as said, the right hand side of the expression represents the entropy generation density σ . We can change this expression a little bit by working out $\nabla(\mu_i/T)$ as follows:

$$\nabla \frac{\mu_{v,i}}{T} = \frac{\partial \frac{\mu_{v,i}}{T}}{\partial \frac{1}{T}} + \frac{(\nabla \mu_{v,i})_T}{T} = \frac{u_{v,i}}{T} + \frac{(\nabla \mu_{v,i})_T}{T} \quad (3.45)$$

When we substitute Eq. (3.45) into Eq. (3.42), we can write for σ :

$$\sigma = J_{Q_v} \nabla \frac{1}{T} - \sum J_i \frac{\nabla(\mu_{v,i})_T}{T} + \sum_k \frac{A_{v,k}}{T} r_k - \frac{1}{T} \frac{d(E_{v,kin} + E_{v,pot})}{dt} \quad (3.46)$$

In case we apply a static electric field to our system, leading to an electric current, the electrical energy is converted completely into thermal energy, and Eq. (3.46) changes to:

$$\sigma = J_{Q_v} \nabla \frac{1}{T} - \sum J_i \frac{\nabla(\mu_{v,i})_T}{T} + \sum_k \frac{A_{v,k}}{T} r_k - \frac{J_v}{T} \nabla E - \frac{1}{T} \frac{d(E_{v,kin} + E_{v,pot,remaining})}{dt} \quad (3.47)$$

The first term in Eq. (3.47) is the product of the heat flux and the gradient of $1/T$, the second term is the product of the flux of matter divided by T and the gradient of the chemical potential at constant temperature T , summed up over all substances, the third term contains the contribution due to the occurrence of chemical reactions. Per reaction, there is a contribution of the driving force of chemical reaction k , the affinity density divided by T , multiplied with the speed of that reaction. The fourth term is made explicit from the last term, the dissipation of kinetic and the remaining potential energies, and is the product of the electric current density divided by T and the gradient of the electric field \mathbf{E} .

If we consider a steady state flow process, with stream direction x , a constant cross sectional area A , and assume that mixing in directions radial with respect to x is complete, we can transform Eq. (3.47) to a form containing products of streams and driving forces:

$$\sigma A dx = A J_{Q_v} \frac{\Delta \frac{1}{T}}{dx} - \sum \frac{A J_i}{T} \frac{\Delta \mu_{i,T}}{dx} + \sum_k \frac{A_k}{T} r_k - \frac{A J_v}{T} \frac{dE}{dx} - \frac{1}{T} \frac{d(E_{kin} + E_{pot,remaining})}{dt} \quad (3.48)$$

Finally we get:

$$\Delta \dot{S}_{irrev} = \dot{Q} \Delta \frac{1}{T} - \sum \dot{n}_i \frac{\Delta \mu_{i,T}}{T} + \sum_k \frac{A_k}{T} r_k - \frac{I}{T} \Delta V - \frac{1}{T} \frac{d(E_{kin} + E_{pot,remaining})}{dt} \quad (3.49)$$

In Eq. (3.49) the last term can often be neglected, and in the process industry not many operations make use of an electric field. The dissipation of kinetic energy can be worked out further by using the expertise field of physical transport phenomena. In most cases we only have to retain the first three terms of the equation. For example for heat transfer, many separation processes, chemical reaction processes and/or

combinations of them, this more simplified form suffices.

We now apply Eq. (3.49) to the heat exchanger we considered before in paragraph 3.2, and take only the first term on the right hand side into account. We assume that Q/l per second is constant. In this expression is l the characteristic length of the heat exchanger. On integration over dx from $x = 0$ to $x = l$, we get:

$$\Delta\dot{S}_{irrev} = \frac{\dot{Q}}{l} \int_{x=0}^{x=l} \left(\frac{1}{T_{c,x}} - \frac{1}{T_{h,x}} \right) dx \quad (3.50)$$

To solve Eq. (3.50), we must express $T_{c,x}$ and $T_{h,x}$ as functions of x . At $x = 0$, $T_{c,x} = T_{c,out}$ and $T_{h,x} = T_{h,in}$, at $x = l$, $T_{c,x} = T_{c,in}$ and $T_{h,x} = T_{h,out}$. From these data, and the assumption that both $T_{c,x}$ and $T_{h,x}$ are linear functions of x , we can write:

$$T_{c,x} = \frac{(T_{c,in} - T_{c,out})}{l} x + T_{c,out} \quad (3.51)$$

$$T_{h,x} = \frac{(T_{h,out} - T_{h,in})}{l} x + T_{h,in} \quad (3.52)$$

Using Eq. (3.51) and Eq. (3.52) in combination with Eq. (3.50), we get the following result:

$$\Delta\dot{S}_{irrev} = \frac{\dot{Q}}{l} \left(\frac{l}{T_{c,in} - T_{c,out}} \ln T_{c,x} - \frac{l}{T_{h,out} - T_{h,in}} \ln T_{h,x} \right) \Big|_0^l \quad (3.53)$$

This equation must be solved further by application of the higher and the lower boundary conditions. The integration boundaries are indicated in Eq. (3.53), as shown after the last bracket. After this application we get:

$$\Delta\dot{S}_{irrev} = \frac{\dot{Q}}{l} \left(\frac{l}{T_{c,in} - T_{c,out}} \ln \frac{T_{c,in}}{T_{c,out}} - \frac{l}{T_{h,out} - T_{h,in}} \ln \frac{T_{h,out}}{T_{h,in}} \right) \quad (3.54)$$

To solve Eq. (3.54) further, we must realize that Q per second can be written in two different expressions, as shown in Eq. (3.55):

$$\dot{Q} = \dot{n}_c < c_{p,mol,c} > (T_{c,out} - T_{c,in}) = \dot{n}_h < c_{p,mol,h} > (T_{h,in} - T_{h,out}) \quad (3.55)$$

$<c_{p,mol,c}>$ and $<c_{p,mol,h}>$ are the mean molar heat capacity values at constant pressure for the cold and hot stream respectively. If we introduce the expressions from Eq. (3.55) in Eq. (3.54), we get the following result:

$$\Delta\dot{S}_{irrev} = \dot{n}_c < c_{p,mol,c} > \ln \frac{T_{c,out}}{T_{c,in}} + \dot{n}_h < c_{p,mol,h} > \ln \frac{T_{h,out}}{T_{h,in}} \quad (3.56)$$

Equation (3.56) is indeed the sum of the entropy increase of the cold stream and the entropy decrease of the hot stream, and is positive. As we have seen, the loss in the potential amount of power for our heat exchanger is given by the Gouy-Stodola relation, and is shown in Eq. (3.57):

$$\dot{W}_{lost} = T_o \Delta \dot{S}_{irrev} = T_o (\dot{n}_c < c_{p,mol,c} > \ln \frac{T_{c,out}}{T_{c,in}} + \dot{n}_h < c_{p,mol,h} > \ln \frac{T_{h,out}}{T_{h,in}}) \quad (3.57)$$

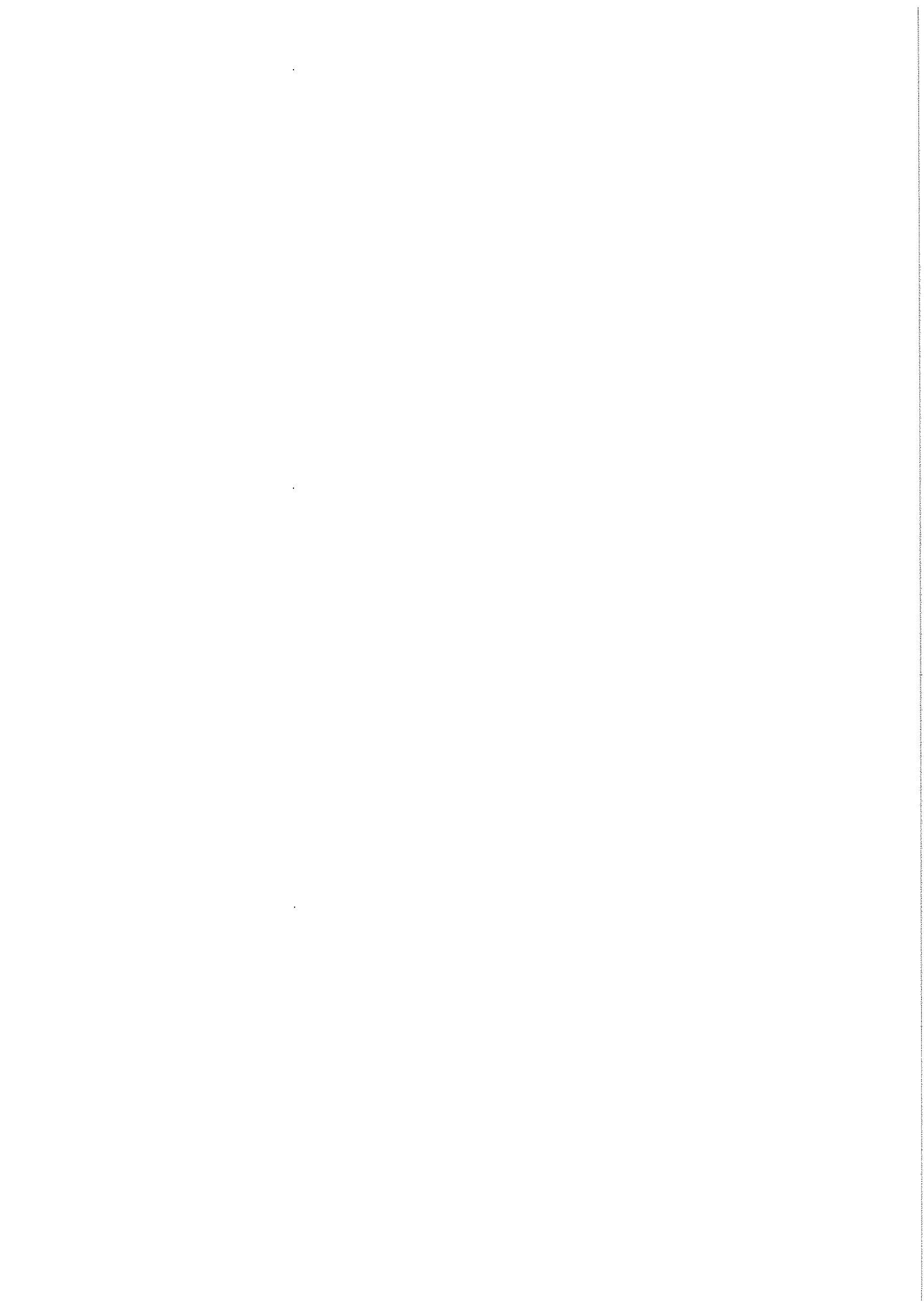
We have seen examples of the degradation of the amount of power before and after a process has taken place as a consequence of a decrease in temperature level. The example of the electric field has shown us that work was transferred completely into thermal energy due to friction. Energy related to a displacement in a certain direction was converted into thermal energy, being directly related to the random movement of molecules of the system. Losses are directly related to a larger input of energy carriers. This means that a reduction of the losses of processes directly reduces the energy costs.

3.5 Conclusions

- During a process the potential amount of power decreases
- The potential amount of power lost is linearly related to entropy generation
- Entropy generation is the sum of products of the driving forces and their conjugated streams, or
- Entropy generation is the sum of products of the gradients of the driving forces and their conjugated fluxes
- The reversible limit is the best case
- The most reasonable efficiency can be defined with respect to the reversible limit
- Power dissipation can be interpreted molecularly: directed movement changes into random movement (friction)
- Often, streams were chosen in the design of a process, that means that we have to investigate whether reduction of the driving forces is reasonably possible
- Driving force reduction leads to a reduction of the input of energy carriers
- Driving force reduction often leads to lower environmental impact
- Driving force reduction leads to lower operating costs and often to higher investment costs
- In the process industry, the potential amount of power input related to equipment is in the order of some % of the loss in the potential amount of power input related to the processes
- Analysis of the losses leads to process improvement, and certainly, if already applied in the design phase
- In answer to the question: ‘What about losses?’, we can say: ‘Scrutinizing our technological activities, understanding, and quantification of these losses, in order to improve our processes, are of utmost importance for society, and in particular for the engineer
- A useful paraphrase is: ‘All Joules are equal but some Joules are more equal than others

3.6 References

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4 Refrigerated facilities

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4.1 Introduction

Temperature abuse can occur when foods are not kept cold or are not placed in a refrigerated space and rapidly cooled to an adequate temperature (e.g. less than 5 °C). Refrigeration capacity is also required for rooms where frozen foods are stored. Buildings and facilities for cold storage have to be designed and built not only to meet production and hygiene requirements, but also to operate in an energy-efficient manner.

Exergy analysis can be applied to buildings, e.g. to provide insight into losses due to heat transfer through a room enclosure. Exergy analysis can also articulate intuitive notions which are not expressed in energy analysis, e.g., that thermal energy tends to be more valuable when its temperature is further from the environmental temperature, and that something is consumed (or lost) when the temperature of a substance or system is brought nearer to the environmental temperature. For example, while energy is conserved when heat is transferred through a wall, exergy is actually lost in the course of this heat transfer.

Exergy analysis is also useful for comparing different energy carriers (e.g. electricity and heat) on an equivalent basis. For example, if a compression refrigeration cycle is used to keep a room (e.g. a cold store) at a constant temperature, electricity will be required to drive this cycle. The room exergy losses can be linked to the thermal exergy output required from the refrigeration cycle; this in turn can be linked to the electricity required to drive the cycle. Because the thermal exergy required at the room has a lower value than the electricity supplied to the refrigeration cycle, one kW thermal exergy lost through the room enclosure (e.g., due to heat transmission) is likely to require more than one kW electricity input to drive the cycle. By the same token, exergy savings at room level may well translate into larger exergy savings further up in the chain (e.g. substantially smaller electricity input to drive the refrigeration cycle).

Section 4.2 of this chapter discusses the exergy of heat and cold near environmental temperature, and uses the exergy factor to express a general relationship between thermal exergy and temperature. It shows an exergy factor plot to quantify the intuitive notion that heat or cold is more valuable when further from the environmental temperature. Such a notion is typified by e.g. the underground ice stores built before the advent of mechanical refrigeration, in order to preserve winter ice for warmer months.

Section 4.3 addresses exergy needs at the core and at the periphery of processes by means of two examples. Being valid for the specific situations addressed, these examples are not meant to spawn general conclusions. Nevertheless, they indicate that

exergy losses at the core of a process (e.g., distillation; room heating) can build up to substantially larger exergy losses at the periphery (e.g. heat and power utilities).

Section 4.4 reviews simplified expressions for calculating exergy losses due to heat transfer through the enclosure of a cold store, and section 4.5 presents a calculation example using these expressions. Section 4.6 closes this chapter.

4.2 Exergy of heat and cold near ambient conditions

Exergy (or availability) can be defined [2] as the maximum theoretical work that would be delivered by a system as it comes into equilibrium with the environment. This implies that this exergy can only be used ‘down to’ the level that is given by the environment. Hence, the definition is not complete until the environment is specified.

Usually average values of the earth are selected: the reference temperature may be specified as $T_o = 298.15 \text{ K}$ (25°C) [5] and the reference pressure p_o as 1 atm. However, the earth is not in equilibrium; temperature, pressure and composition vary from place to place around the globe. When a system, e.g. a steam turbine at 650 K (ca. 380°C), operates relatively far from environmental temperature, the effect of selecting 288.15 K (15°C) or 298.15 K (25°C) as the reference temperature T_o is relatively minor. However, when considering systems at near-environmental temperature, e.g. refrigeration or space heating / cooling, it is recommendable to account for variations of T_o in both place and time. A similar point can be made for differences in pressure and chemical potential, when applicable.

4.2.1 Exergy factor

Exergy often appears as heat and cold, i.e. a temperature difference. Chapter 7 (section 7.3) examines the exergy associated with heat transfer in more detail. The present section defines an exergy factor for simple systems at constant temperature, and uses it to express a general relationship between thermal exergy and temperature.

When comparing between the energy and exergy of a system, it is convenient to use the exergy factor [5], i.e. the ratio Ex/Φ between the exergy Ex and the energy of the transferred heat Φ . This ratio indicates the fraction of the total heat content (Φ) with a potential to be useful as thermal exergy (Ex). This usefulness depends on the temperatures involved.

Assuming the heat is transferred between a system (e.g. food storage room) at temperature T and an environment at temperature T_o (both T and T_o being constant), the exergy factor becomes

$$\frac{Ex}{\Phi} = \left| \frac{1 - T_o}{T} \right| \quad (4.1)$$

The right term of this equation is also known as the Carnot factor, which is further discussed in Chapter 7, section 7.3.

In case the system temperature T changes as a result of heat transfer to the environment at T_0 (e.g., as in a flow of ventilation air), the energy and exergy equations undergo integration to reflect this change. For relatively small temperature changes, the specific heat c_p can be assumed to remain constant, and hence can be cancelled out in the Ex/Φ quotient. The exergy factor thus becomes [5]:

$$\frac{Ex}{\Phi} = \left| 1 - \frac{T_0}{T - T_0} \ln \frac{T}{T_0} \right| \quad (4.2)$$

One special case of Eq. (4.2) is that of system a at temperature T_a exchanging heat with system b at T_b , both T_a and T_b being different from T_0 . Such an example could be air from docking station b at e.g. 5 °C infiltrating into cold store a at e.g. -20 °C. The exergy factor then is

$$\frac{Ex}{\Phi} = \left| 1 - \frac{T_0}{T_b - T_a} \ln \frac{T_b}{T_a} \right| \quad (4.3)$$

4.2.2 Thermal exergy and temperature

The exergy factors plotted in Fig. 4.1 express a general relationship between thermal exergy and temperature, based on Eq. (4.1). They indicate a theoretical need for thermal exergy (as a fraction of the thermal energy needed), in order to keep the system temperature T_{in} constant in an environment of a different temperature T_0 . This definition of the exergy factor is only concerned with the minimum thermal exergy needs for compensating the spontaneous heat transfer across the system boundary (e.g., the walls of a room or vessel). It neglects the thermal exergy losses likely to arise from heat transport and temperature differences between the room and the refrigeration cycle. It also does not account for heat gains from air infiltration or internal heat loads (e.g. cool-down of stored product in cold stores).

The vertical axis of Fig. 4.1 displays the exergy factor Ex/Φ , and the horizontal axis shows the ratio T_{in} / T_0 of the temperature T_{in} inside a given system (e.g. cold store, oven, furnace) to the outdoor temperature T_0 . The use of these dimensionless ratios enables the exergy factor for different temperature combinations (T_{in} and T_0) to be expressed in one single curve. Fig. 4.1 shows that the exergy factor Ex/Φ is higher the further the temperature of a system is from the environment temperature.

Since Kelvin temperatures are used ($0\text{ °C} = 273.15\text{ K}$), many temperatures commonly found in food-processing fall within the relatively narrow range of T_{in} / T_0 between 0.9 and 1.7. For example, in The Netherlands we may assume a common outdoor temperature to be $T_0 = 283\text{ K}$ (ca. 10 °C). If we take a cold store at an indoor temperature T_{in} of 253 K (ca. -20 °C), the ratio T_{in} / T_0 is 0.9. For an oven at $T_{in} = 473\text{ K}$ (ca. 200 °C), T_{in} / T_0 is 1.73. By way of comparison, in a high-temperature furnace (e.g. for baking bricks), temperatures may reach 1473 K (ca. 1200 °C), whereby the ratio T_{in} / T_0 becomes 5.2.

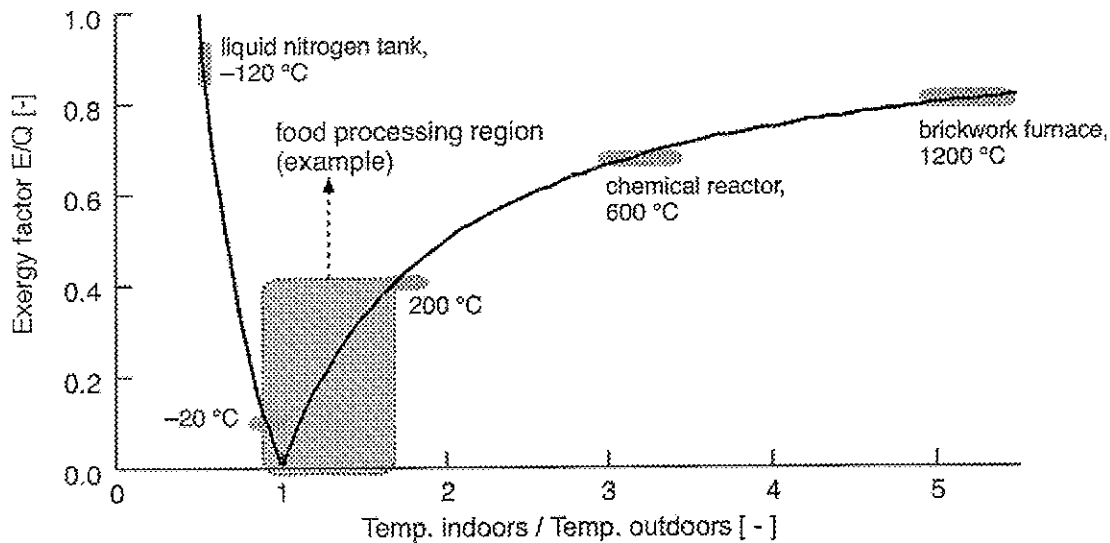


Figure 4.1 Exergy factor of heat and cold as a function of temperature, based on Eq. (4.1)

One can define the cooling need (or refrigeration load) as the rate at which heat has to be removed from the cold storage room to compensate for heat gains through its walls, in order to keep T_{in} constant. An exergy factor of 0.12 indicates that the exergy need accounts for only 12% of the cooling energy need.

Although this exergy factor may be low compared to that of cryogenic systems such as liquid nitrogen storage tanks, it is shown in section 4.3 that changes in thermal exergy needs at user level (e.g. cold storage room) may result in greater changes at utility level (e.g. refrigeration cycle).

4.2.3 Outdoor temperatures and exergy needs

The previous section first defined the exergy factor Ex/Φ , and then used it to express the relation between temperature levels and the minimum thermal exergy needed to keep enclosed systems (e.g. rooms or vessels) at constant temperature. The present section illustrates how these exergy needs can vary with outdoor temperature.

Figure 4.2 shows exergy factors, based on Eq. (4.1). The figure indicates the theoretical need for thermal exergy to compensate for heat gains through the walls, so as to keep constant the indoor temperature T_{in} of indoor spaces at 20 °C (e.g. an office) and -20 °C (e.g. a cold storage room). The environment temperature T_o is varied as shown in Table 4.1.

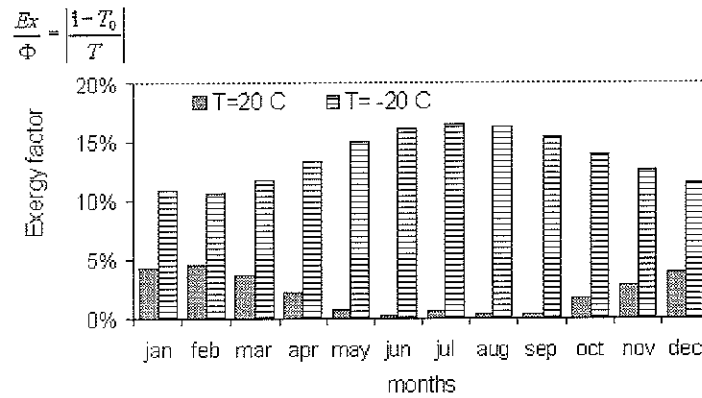


Figure 4.2 Exergy factor for rooms at $-20\text{ }^{\circ}\text{C}$ and $+20\text{ }^{\circ}\text{C}$, based on monthly outdoor temperatures (from Eq. (4.1))

Table 4.1 Monthly average outdoor temperatures for The Netherlands (de Bilt, 1961-1970)

Months	jan	feb	mar	apr	may	jun	jul	aug	sep	oct	nov	dec
$T_o\text{ [}^{\circ}\text{C]}$	7.3	6.9	9.4	13.7	17.9	20.7	21.8	20.9	18.9	15.2	11.9	8.9

For the room at $20\text{ }^{\circ}\text{C}$, the monthly variation in exergy factors Ex/Φ in Fig.4.2 quantifies the intuitive perception that heat is scarcer (or harder to obtain) in the colder months, when it is mostly needed. For the cold store at $-20\text{ }^{\circ}\text{C}$, on the other hand, the Ex/Φ ratios are higher in the warmer months. They do remain above 10% throughout the year, reflecting the intuitive notion that refrigeration temperatures are harder to maintain than room temperatures.

Figure 4.2 also illustrates the intuitive notion that heat tends to be more valuable in winter and cold in summer, when they are scarcer (and thus harder to obtain or keep). This notion would not be apparent from a first-law energy analysis.

4.2.4 The exergy of cold

When $T_{in} < T_o$, there is a lack of energy in the system, i.e., coldness, which is sometimes expressed as negative energy. Exergy, however, is always positive (also by definition, if one considers that work is positive at all times) [6]. Shukuya [4] refers to the ‘cool exergy’ which flows through the building envelope system (exterior walls), when the indoor temperature T_{in} is lower than the outdoor temperature T_o .

Although the direction of energy flow changes depending on whether $T_{in} > T_o$ or $T_{in} < T_o$, the exergy flow through a building envelope is always from indoors to outdoors. What changes, according to Shukuya, is only whether this outward flow is of ‘warm’ exergy ($T_{in} > T_o$) or ‘cool’ exergy ($T_{in} < T_o$).

4.3 Exergy needs from core to periphery

Chapter 7 discusses the physical principles underlying exergy losses from heat distribution, heat storage and heat transfer. It can be argued that reducing the exergy needs at the core of a process can help to avoid amplified exergy needs at the peripheral levels. This line of reasoning presupposes that core processes fulfil the user's primary needs (e.g. to keep a room or industrial process at constant temperature), and that the periphery (e.g. energy utilities) is intended to supply thermal or electric exergy to the core processes. Because heat transfer, heat distribution and energy conversion usually entail exergy losses, it is likely that the exergy input required by peripheral processes will be larger than the actual exergy needed by the core processes.

Since the definition of core and utilities depends on the nature of the process being examined, caution is required when formulating and extrapolating conclusions. Analysis results for a specific system are likely not to be directly applicable to other systems with different needs or composition. Although the results themselves should not be extrapolated, an approach from core to periphery can provide insight on a case-by-case basis.

Two different examples of a core-to-periphery approach are presented below. What is intended as a reference is not necessarily specific conclusions drawn for those particular cases, but rather the authors' approach to their analyses.

4.3.1 The 'shell model' approach

De Wit [7] uses a 'shell' model (*schillenmodel*) to analyze a distillation process. The core consists of reactors and separation processes. Heat delivery and heat recovery shells surround the core, transferring heat to or recovering heat from the core process. The use of heat pumps allows heat to be transferred from low to high temperature (i.e., 'against' the usual temperature gradient).

Figure 4.3 is a simplified version of a figure shown by De Wit in [7]. The inner shell is a distillation process with small exergy losses (schematically shown by a thin vertical arrow to the left of the core process). A heat pump (HP, black rectangle) exchanges heat with the core process. Part of this heat is valuable (exergy, shown as white horizontal arrows); the unusable portion is designated 'anergy'. The exergy losses from the heat pump are shown to be larger than those from the core process (heat pump and refrigeration cycle exergy losses are analyzed in chapter 5). A combined heat and power (CHP, black rectangle) cycle burns natural gas, and supplies electricity to the heat pump (white horizontal arrow) and steam to the core process (oblique arrow). The exergy losses from the CHP (including combustion) are shown to be larger than those in the heat pump.

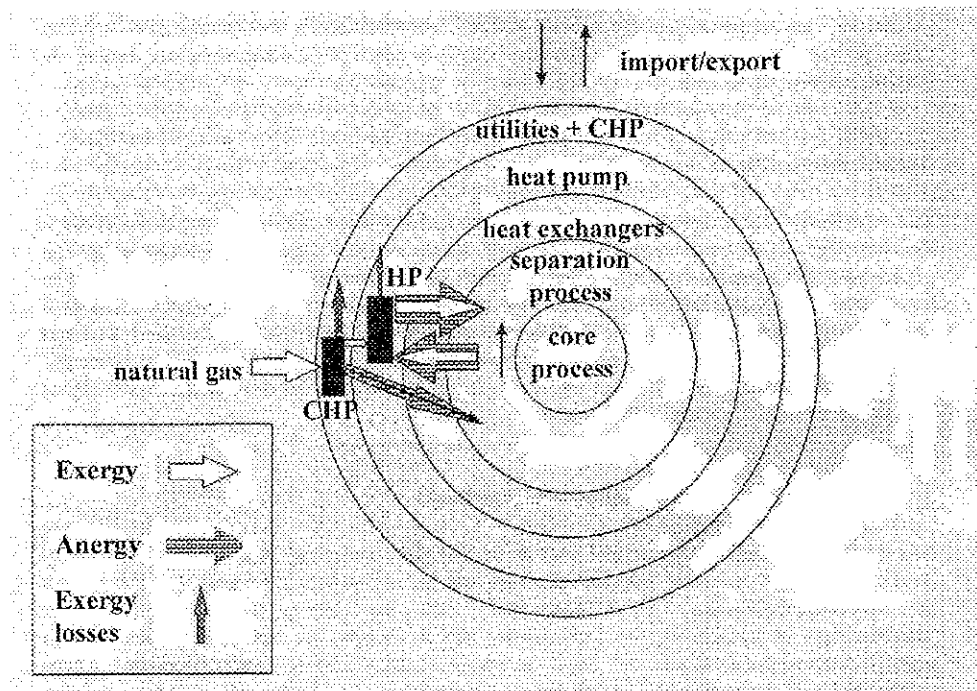


Figure 4.3 Schematic of the 'magnifying lens' effect of exergy losses from core to outer shell (adapted from de Wit [71])

These results are specific for a given set of conditions, and presuppose a core process requiring large amounts of heat at relatively low temperature (i.e., low exergy needs). Due to the low exergy needs at the core level, a heat pump can be driven with a relatively small amount of electric exergy. De Wit performed a similar analysis for a process with high exergy needs at the core (e.g. higher temperatures) and noted that the increased need for electricity to drive the heat pump would result in substantially higher needs for natural gas to drive the CHP cycle.

De Wit notes that small exergy losses at the core cause larger exergy losses in the outer shells of the model. Because exergy losses are caused by every heat transfer and heat transport step, reducing exergy losses at the inner core will lead to large reductions in fuel and electricity input at the outer shell. The 'shell' model works as a magnifying glass for the exergy losses. Hence, reducing exergy losses at the core often works more effectively; if a peripheral improvement is to have the same effect as a core improvement, then this improvement will have to be much larger.

When considering these remarks, it is important to keep in mind that reducing thermal exergy losses does not imply eliminating them: when the temperature of a system differs from that of the environment, this temperature difference will cause exergy losses due to heat transfer.

4.3.2 The 'exergy consumption' approach

Shukuya [3] uses the term 'exergy consumption' to articulate the idea that something is consumed when energy is degraded, and to explore the benefits of using natural energy sources (e.g. solar) to meet the exergy needs of buildings. Even more than food

production processes, buildings are characterized by a need for large amounts of thermal energy at levels very close to the environment.

One specific example of thermal exergy for core processes in buildings refers to space heating systems. Shukuya presents a calculation example for a system consisting of a gas-fired boiler, a room heater (e.g. a fan coil) and a room (indoor air plus building envelope – external walls). The room can be regarded as the core, which needs thermal exergy to be kept at a constant temperature.

Figure 4.4 presents a schematic of the system, and Table 4.2 lists the assumptions.

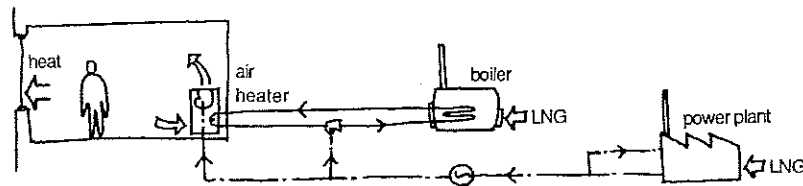


Figure 4.4 Schematic of the space heating system analyzed by Shukuya [3]

Table 4.2 Assumptions for the space heating system analysis [3]

Case	Overall heat transfer coefficient (U value) averaged for walls and window [$\text{W}/(\text{m}^2 \cdot \text{K})$]	Boiler thermal efficiency [%]
1	3.00	80
2	1.59	80
3	1.59	95
4	3.00	95

A 6.0m x 6.0m x 3.0m room with one exterior wall having a 1.5m x 6m glazed window is assumed. The exterior-window and -wall U values are 6.2 and 2.67 $\text{W}/(\text{m}^2 \cdot \text{K})$ for Cases 1 and 4; 3.6 and 1.14 for Cases 2 and 3. The infiltration air change rate is 0.8 h^{-1} for Case 1; and 0.4 h^{-1} for Cases 2 and 3. The room air temperature kept constant at 293 K (20°C) in all cases while the outdoor air temperature is assumed to be constant at 273 K (0 °C). Outlet air temperature, inlet and outlet water temperatures of the heat exchanger are assumed to be 303 K (30°C), 343 K (70°C), and 333 K (60°C), respectively, for all Cases. Electric power is supplied to a fan and a pump at a rate of 30 W and 23 W in Case 1; 16 W and 17 W in Cases 2 and 3. The ratio of the chemical exergy to the higher heating value of liquidified natural gas (LNG) is 0.94. The thermal efficiency of the power plant, that is, the ratio of produced electricity to the higher heating value of LNG supplied is 0.35.

The calculation example consists of a baseline case (case 1, with a poorly insulated building envelope and a boiler of 80% thermal efficiency) and three improved cases. In

case 2, building envelope insulation is improved, which leads to reduced thermal exergy needs at the core. In case 4, the boiler efficiency is improved to 95% (reduced exergy needs at the periphery), and in case 3 both wall insulation and boiler efficiency are improved. The largest exergy loss shown in Fig. 4.5 takes place at the boiler, and is due to the degradation of chemical exergy in the natural liquefied gas (LNG, used in Japan) into heat, via a combustion process. Thereafter, hot water from the boiler exchanges heat with room air in the heating unit, leading to subsequent exergy losses. Heated air then mixes with room air, leading to a further exergy loss. Finally, the thermal exergy of the indoor air is dissipated to the outdoor environment, through the building envelope (external wall and window).

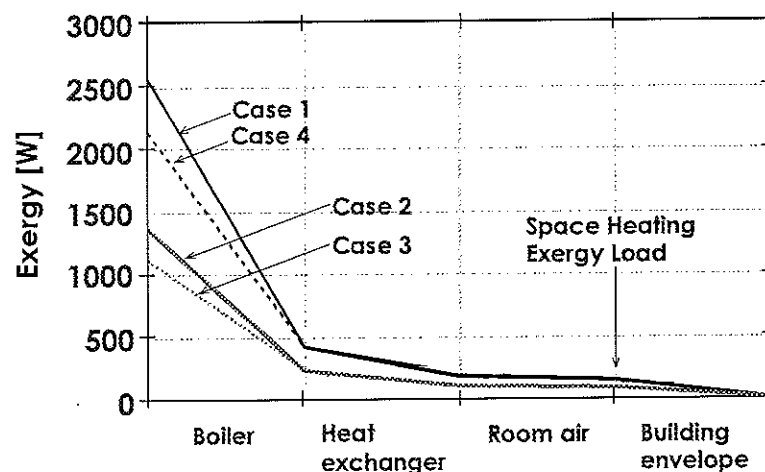


Figure 4.5 Exergy losses from periphery to core in a space heating system [3]

The thermal exergy required for the ‘core’ process of keeping the room temperature constant accounts for only ca. 6 ~ 7% of the chemical exergy input into the boiler. This relatively small need of exergy at room level may suggest that there is little to be gained by improving the building envelope. In this example, however, the calculations indicate that insulating the building envelope (from 3 to 1.6 W/m² K) is more effective than improving the boiler thermal efficiency from 80% to 95%.

These two examples illustrated two different approaches to the analysis of exergy flows in multi-component systems, where thermal exergy is needed to sustain a core process (e.g., distillation, room heating) and supplied by peripheral utilities (e.g., heat pump, boiler).

Section 4.4 presents simple expressions for estimating the thermal exergy needed to keep a room (e.g. cold store) at constant temperature, and section 4.5 shows a calculation example illustrating how this exergy needed to keep the room cool can be magnified at the refrigeration cycle.

4.4 Exergy loads for refrigeration

The terms ‘thermal load’ and ‘refrigeration load’ are often used in air-conditioning and refrigeration engineering to designate the rate at which heat is being gained or lost by a

system (often a room). If the system (e.g. a room) is to be kept at constant temperature, these heat gains (or losses) have to be compensated for by the removal (or supply) of heat at the same rate. This heat removal (or supply) is often carried out by mechanical means.

Typically, refrigeration load consists of various segments: transmission (heat transferred into the refrigerated space through walls, floors and other surfaces); product (heat removed from and produced by refrigerated products); internal (heat produced by people, lighting, motors); air infiltration (air entering the refrigerated space); and equipment (heat gains created by refrigerating equipment).

This section focuses on the transmission and air infiltration loads of refrigerated spaces (such as cold stores). These loads refer to the transmission and infiltration heat gains – and resulting exergy losses – that have to be compensated for in order to keep the spaces at a constant temperature. It should be kept in mind (as discussed in section 4.2.4) that ‘cold’ exergy will flow outwards (i.e. will be lost from a cooled room to the outdoor environment), even though the energy flows will be in the inward direction (i.e. heat will be gained from the outside environment).

The exergy load for refrigerating a room Ex_{load_room} can thus be defined as the outward flow of exergy from a room due to heat transmission through the building envelope Ex_{trns} and air infiltration Ex_{inf} .

$$Ex_{load_room} = Ex_{trns} + Ex_{inf} \quad (4.4)$$

Section 4.2.1 dealt with the exergy factors Ex/Φ for systems kept at constant temperature (e.g. a room) or undergoing a temperature change (e.g. air entering the room and cooling down to room temperature). When the corresponding energy flows Φ are known, exergy flows Ex can be obtained simply by multiplication. When directly calculating thermal exergy flows, it is convenient to use Eqs. (4.5) to (4.7) below.

4.4.1 Transmission loads

The thermal exergy Ex_{trns} lost through the building envelope (surface area A , overall heat transfer coefficient U) from a space at a constant indoor temperature T_{in} , owing to a temperature difference with an outdoor environment at T_o , can be expressed as:

$$Ex_{trns} = \Phi_{trns} \times \left| \frac{T_{in} - T_o}{T_{in}} \right| = \frac{U \times A \times (T_{in} - T_o)^2}{T_{in}} \quad (4.5)$$

The indoor air temperature T_{in} is presumed to be constant and uniform throughout the room, and temperature differences are assumed to be small enough to justify calculation with an overall heat transfer coefficient [5].

4.4.2 Air infiltration loads

The exergy loss Ex_{inf} accompanying heat transfer Φ_{inf} to a steady mass flow (Φ_m) of air (isobaric specific heat c_p) infiltrating a colder space can be stated in simplified form as [5]:

$$Ex_{inf} = \Phi_{inf} \times \left(1 - \frac{T_o}{T_{in} - T_o} \times \ln \frac{T_{in}}{T_o}\right) = \Phi_{m_inf} \times c_p \times \left|T_{in} - T_o - T_o \times \ln \frac{T_{in}}{T_o}\right| \quad (4.6)$$

The air is regarded as a dry, ideal gas with specific heat 1.0 kJ/kg K. When the infiltrating air temperature T_{inf} differs from the outdoor temperature T_o , the exergy loss from air infiltration becomes:

$$Ex_{inf} = \Phi_{inf} \times \left(1 - \frac{T_o}{T_{in} - T_{inf}} \times \ln \frac{T_{in}}{T_{inf}}\right) = \Phi_{m_inf} \times c_p \times \left|T_{in} - T_{inf} - T_o \times \ln \frac{T_{in}}{T_{inf}}\right| \quad (4.7)$$

4.4.3 Exergy losses at refrigeration cycle and heat source / sink

Chapter 5 discusses the exergy of refrigeration cycles in detail, and provides the equations used in the present chapter to calculate the exergy losses at the individual components: condenser, evaporator, compressor and throttling device.

In Chapter 5, the exergy input is defined as the compressor shaft power. The output, or useful exergy effect of the refrigeration cycle $Ex_{rise_source_flow}$, is defined as the exergy flow change of the heat source medium (e.g. outside air). The present chapter looks into the further exergy losses which may take place when the exergy $Ex_{rise_source_flow}$ gained by the heat source medium is used to keep a room (e.g. cold store) at constant temperature.

The first loss occurs when the heat source medium (e.g. air) enters the evaporator heat exchanger at temperature $T_{source_in} = T_{in}$, exits at a lower temperature T_{source_out} and then mixes with room air at temperature T_{in} . This mixing causes an exergy loss at the heat source, $Ex_{loss_mix_source}$. In simplified form, this loss can be expressed as:

$$Ex_{loss_mix_source} = Ex_{rise_source_flow} - Ex_{load_room} \quad (4.8)$$

Subsequently, the remaining cold exergy Ex_{load_room} is lost via the building envelope through heat transmission Ex_{trns} and air infiltration Ex_{inf} (see Eq. 4.4).

4.5 Exergy loss example for a cold store with refrigeration cycle

This section presents a calculation example for a cold store (enclosed in a docking station) kept at constant temperature by a refrigeration cycle.

4.5.1 Calculation example

A simplified example of thermal exergy losses for a cold store and a refrigeration cycle in steady-state conditions is presented. The system is assumed to consist solely of a cold store (enclosed in a docking station) and a refrigerating machine providing cooling capacity to the cold store. Heat transfer and pressure losses from heat distribution are neglected. Temperatures are assumed to be constant at $T_{in} = -20\text{ °C}$ inside the cold store, $T_o = 10\text{ °C}$ outdoors and $T_{inf} = 5\text{ °C}$ in the docking station surrounding the cold store. The cold store is assumed to lose cold exergy as a result of transmission heat gains, to the docking station (through walls and ceiling) and to the ground (through the floor). Cold exergy losses through the door via infiltration from the docking station are also accounted for. Transmission heat gains through the door are neglected.

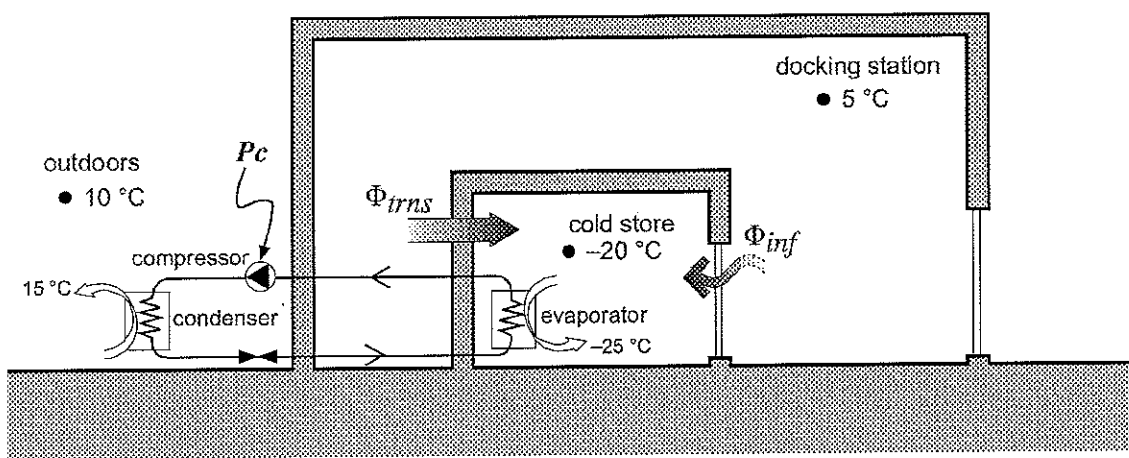


Figure 4.6 Cold store and refrigeration cycle schematic

An air-cooled direct expansion refrigeration cycle is assumed to provide the cooling capacity $\Phi_{in} = 10\text{ kW}$ required to keep the cold store temperature constant at $T_{in} = -20\text{ °C}$ (the exergy of refrigeration cycles is discussed in chapter 5). The cycle is assumed to use Refrigerant R-134a and the isentropic compressor efficiency is taken as $\eta_c = 0.65$. For a basic cycle (single stage, no superheating or subcooling) with evaporation and condensation temperatures constant at -27 °C and 18 °C , these assumptions yield a cooling COP (COP_R) of 2.9 and a compressor power input of $P_c = 3.43\text{ kW}$ [2]. Chapter 5 discusses the exergy of refrigeration cycles in more depth.

From an energy viewpoint, these numbers say that 3.43 kW electric energy are needed for the refrigeration cycle to remove 10 kW thermal energy from the cold store, in order to keep its temperature constant at $T_{in} = -20\text{ °C}$. An exergy analysis, on the other hand, shows losses which are not immediately apparent when examining the process from an energy viewpoint.

Table 4.3 shows an overview of the conditions assumed for the cold store and refrigeration cycle.

Table 4.3 Overview of cold store and refrigeration cycle conditions

Subsystem	Variable	Value
Environment	T_o	10 °C
Refrigeration cycle	refrigerant	R134a
	$T_{\text{condensation}}$	18 °C
	$T_{\text{evaporation}}$	-27 °C
	Φ_{in}	10 kW
	COP_R	2.9
	η_c	65%
Heat source / sink	medium	air
	$T_{\text{sink_in}}$	10 °C
	$T_{\text{sink_out}}$	15 °C
	$\Phi_{\text{m_sink}}$	2.68 kg/s
	$T_{\text{source_in}}$	-20 °C
	$T_{\text{source_out}}$	-25 °C
	$\Phi_{\text{m_source}}$	1.99 kg/s
Docking station	$T_{\text{inf_into_cold_store}}$	5 °C
	$\Phi_{\text{m_inf}}$	0.166 kg/s
Cold store	$T_{\text{in_cold_store}}$	-20 °C
	$U_{\text{wall}} = U_{\text{ceiling}}$	0.29 W/m ² K
	A_{wall}	350 m ²
	$A_{\text{ceiling}} = A_{\text{floor}}$	300 m ²
	U_{floor}	0.17 W/m ² K
	$V_{\text{cold_store}}$	1500 m ³

4.5.2 Calculation results

The results were obtained with a spreadsheet model, using Eqs. (4.4) to (4.8) for the cold store and the equations presented in Chapter 5 for the refrigeration cycle.

Table 4.4 Exergy input and losses, from refrigeration cycle to cold store

Subsystem	Variable	Exergy loss [kW]	Exergy loss [%]
Refrigeration cycle	$Ex_{\text{loss_cond}}$	0.33	9.6
	$Ex_{\text{loss_exp}}$	0.36	10.5
	$Ex_{\text{loss_evap}}$	0.25	7.2
	$Ex_{\text{loss_comp}}$	1.08	31.5
Heat source / sink	$Ex_{\text{rise_sink_flow}}$	0.12	3.4
	$Ex_{\text{rise_source_flow}}$	1.30	37.8
	$Ex_{\text{input}} = P_c$	3.43	100
Cold store	$Ex_{\text{loss_mix_cold-room_air}}$	0.40	30.5
	$Ex_{\text{loss_infiltration}}$	0.28	21.5
	$Ex_{\text{loss_transmission_walls}}$	0.47	36.4
	$Ex_{\text{loss_transmission_floor}}$	0.15	11.6
	$Ex_{\text{input}} = Ex_{\text{rise_source_flow}}$	1.30	100

Table 4.4 shows exergy input and losses for the components analyzed. Figure 4.7 shows a plot of these results, where the exergy losses at the various components are visualized as line segments of varying slope. Since all line segments have the same (arbitrary) length, the slopes visually indicate the relative exergy losses corresponding to each of the components listed under the horizontal axis. These components are grouped into three subsystems, namely: refrigeration cycle; heat source/sink; and room.

The graph is built as follows: the height of the first point (plotted on the vertical axis) corresponds to the electricity input $Ex_{input} = 3.43$ kW. The first line segment corresponds to the condenser exergy losses Ex_{loss_cond} , and is plotted by subtracting $Ex_{loss_cond} = 0.33$ kW from Ex_{input} , plotting the resulting point at 3.10 kW, and drawing a sloped line between both points. Exergy losses for the other components are plotted in a similar manner.

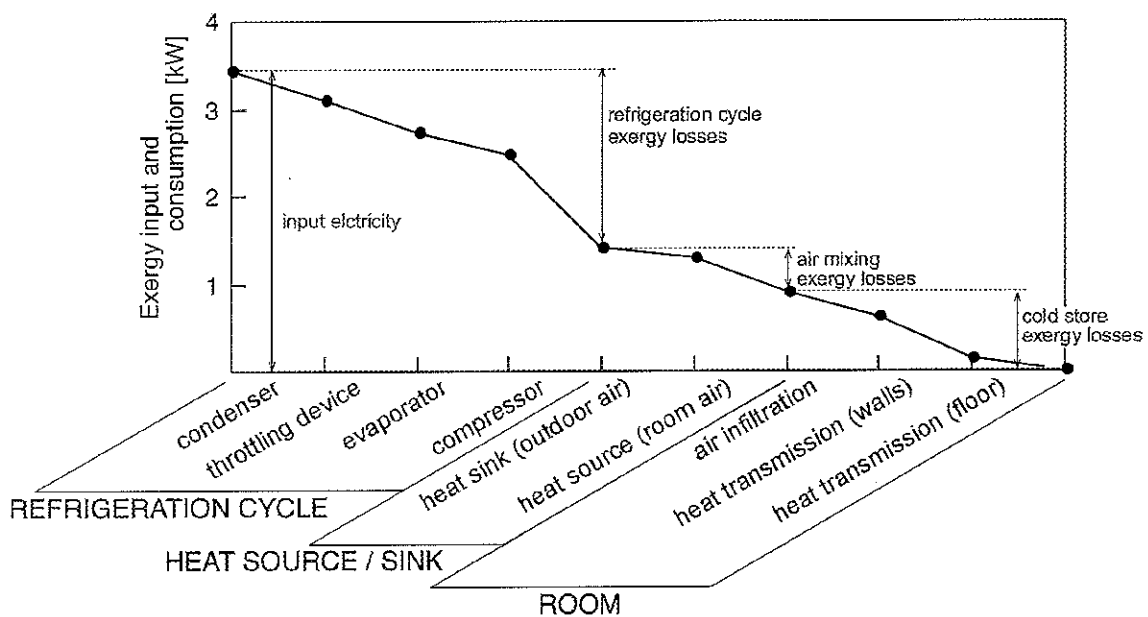


Figure 4.7 Visualization of exergy input and losses, from refrigeration cycle to cold store

For the refrigeration cycle, the largest losses take place at the compressor. Considering the relatively small refrigeration capacity of this cycle, 65% isentropic efficiency was assumed for the calculations. Chapter 5 shows that improving the compressor efficiency can significantly reduce exergy losses.

Exergy losses also take place at the heat sink and heat source, as pointed out in section 4.4.3. Finally exergy is lost in the cold store, due to: cold air supply below room temperature; heat transmission Φ_{trns} via walls, ceiling and floor; and heat gains Φ_{inf} due to air infiltration from the docking station. For the conditions analyzed, the largest exergy losses through the cold store enclosure are due to heat transmission through walls and floor. These losses could be reduced by means of improved thermal insulation (thicker and/or more insulating material) and reduced air infiltration rates (shorter door opening times and/or improved air-tightness). Lowered the docking station temperature can also reduce cold store exergy losses from air infiltration, but will likely increase exergy losses from the docking station to the outdoor environment.

4.6 Conclusions

Food processing plants require relatively large amounts of thermal energy at near-environmental temperatures. The closer these temperatures are to the environment, the lower the exergy content of the required thermal energy.

The exergy factor (the ratio between the exergy Ex and energy Φ of heat) was used to express the exergy content of thermal energy, and its relation to temperature levels. Considering that many food processing temperatures lie in the region between $-20\text{ }^{\circ}\text{C}$ and $200\text{ }^{\circ}\text{C}$, and assuming an outdoor temperature of $10\text{ }^{\circ}\text{C}$ (frequently occurring in The Netherlands), the exergy factors of heat used in food processing fall in the region between 0.1 and 0.4. In other words, only 10% to 40% of the heat or cold delivered by heating or cooling equipment consists of thermal exergy.

Also, because food processing temperatures are relatively close to environmental temperature T_o , the reference value T_o selected for calculation is likely to influence the results, as illustrated in Fig. 4.2. For simple steady-state calculations, monthly or daytime average temperatures may be used, but more detailed dynamic calculations may require hourly temperature values.

This chapter presented simple expressions for quantifying exergy losses in a cold store (enclosed in a docking station) as a result of steady-state transmission and infiltration heat gains through its enclosure. A set of specific conditions (temperatures, insulation values and air infiltration rates), was assumed for the cold store, and its cooling needs were coupled to the output of a refrigerating cycle. Calculations were then performed using a spreadsheet model, in order to show the relative magnitude of exergy losses from refrigerating cycle to cold store.

For the specific conditions analyzed, the largest exergy losses took place in the refrigeration cycle (at the compressor), followed by the cold store enclosure, (heat transmission through walls and ceiling). For these specific conditions, exergy losses due to air infiltration were smaller at 8%.

The basic model presented in this chapter allows the effect of varying temperatures, thermal insulation and air infiltration rates to be examined for a room (e.g. cold store) for steady-state conditions. This variation is left as an exercise. The model can also be coupled to a refrigeration cycle model, in order to assess the relative magnitude and impact of losses in both the cold store and the refrigeration cycle. Chapter 5 discusses the exergy of refrigeration cycles in more detail.

4.7 Acknowledgements

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5 Refrigeration and heat pumping in food processing plants

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5.1 Introduction

In food processing plants refrigeration is often an essential utility to guarantee the product quality. Sometimes it is even more than a utility since the whole production process depends on the availability and performance of the refrigeration plant, as for instance in ice manufacturing plants.

This chapter introduces and discusses evaluation criteria of vapour compressor cycles based on analysis with the first and second laws of Thermodynamics. Although other cycles can be applied for attaining refrigeration effects, vapour compressor cycles are most common. The most simple vapour compressor cycle will be discussed here as an example. Food processing plants will mostly require complex types of vapour compressor cycles. Similar analysis can be developed for such complex cycles. Sahin and Kodal [8,9] and Chen [2] present thermoeconomic optimization studies of two stage refrigeration systems including the effect of the major irreversibilities and economic parameters.

Figure 5.1 shows schematically a vapour compression refrigeration / heat pump cycle. The cycle operates counter clock wise. A refrigerant undergoes a number of processes in a closed loop. Starting from point 1, low pressure refrigerant vapour is compressed to high pressure state 2. This requires the compressor shaft power P_c . In the condenser heat is rejected to a heat sink. In food processing plants the heat sink is mostly the surrounding air, often in combination with water evaporative cooling. Sometimes the heat can be rejected to another process step where heating is needed. In that case the cycle has a dual function: refrigerating machine and heat pump. Due to heat rejection, the refrigerant first cools down to saturated vapour conditions, then it condenses until all refrigerant becomes a liquid at saturated conditions. Eventually the refrigerant is slightly subcooled as it leaves the condenser with state 3. Generally a throttling device substitutes the turbine shown in the figure. In that case no power is recovered from the expansion process, i.e. $P_t = 0$. After the throttling device the refrigerant is again at low pressure and has a state in the two-phase region. In the evaporator heat is absorbed from the heat source until the refrigerant attains vapour saturated conditions. Eventually the vapour is superheated until state 1.

The rate of heat removal from the heat source, heat rejection to the heat sink and the power input to the compressor are determined from their definitions:

$$\Phi_{in} = \dot{\phi}_{m_ref} * (h_1 - h_4) \quad (5.1)$$

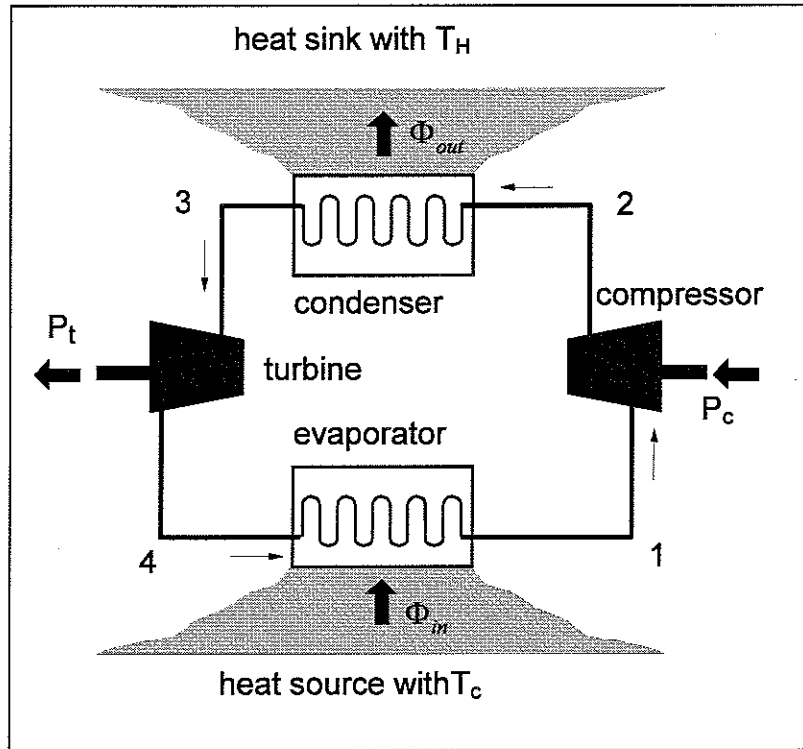


Figure 5.1 Vapour compression refrigeration / heat pump

$$\Phi_{out} = \phi_{m_ref} * (h_2 - h_3) \quad (5.2)$$

$$P_c = \phi_{m_ref} * (h_2 - h_1) \quad (5.3)$$

The enthalpy of state 2 follows from the definition of isentropic compressor efficiency, if this efficiency is known:

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (5.4)$$

The (first law) performance of refrigeration cycles and heat pumps is expressed in terms of the coefficient of performance (COP), which is defined as

$$COP = \frac{\text{objective_of_the_cycle}}{\text{required_input}} \quad (5.5)$$

The objective of a refrigeration cycle is to remove heat from the low temperature heat source:

$$COP_R = \frac{\Phi_{in}}{P_c - P_t} \quad (5.6)$$

while the objective of a heat pump is to deliver heat to the high temperature heat sink:

$$COP_{HP} = \frac{\Phi_{out}}{P_c - P_t} \quad (5.7)$$

The coefficient of performance will be a function of the operating conditions of the system and of the refrigerant used in the cycle. Fig. 5.2 shows for a number of refrigerants the single stage vapour compression cycle COP for a condensation temperature of 35°C. The COP has been calculated by filling Eqs. (5.1) and (5.3) in Eq. (5.6) and calculating the different enthalpy values with REFPROP [7]. All further presented refrigerant data have been calculated with this program. As the evaporating temperature reduces, the COP significantly decreases but is mostly above 1. Ammonia (NH₃) and n-pentane show the largest values for the COP, followed by a number of refrigerants with similar performance: isobutane (R600a), propane (R290), R134a and R410A. The in the Netherlands frequently used low temperature mixtures R404A and R507 show significantly lower COP's. Carbon dioxide (CO₂) as a single stage cycle shows unacceptable low COP values. By using complex system designs it is possible to attain higher COP's at lower temperatures.

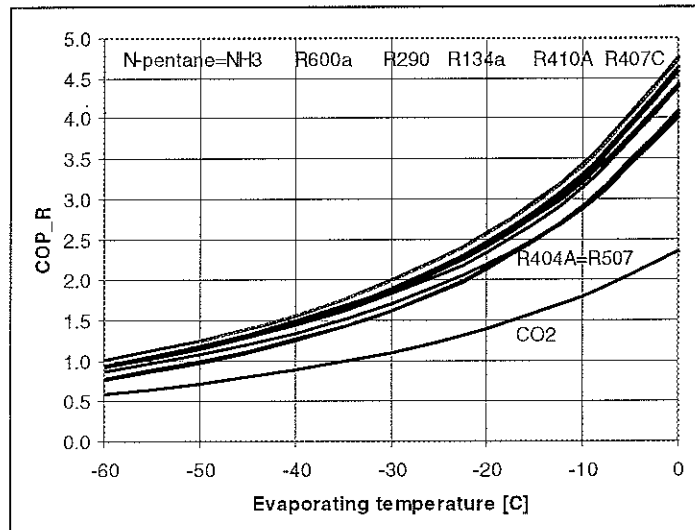


Figure 5.2 Coefficient of performance of refrigerating cycle as a function of the evaporating temperature for a condensation temperature of 35°C, no subcooling, no superheating and compressor isentropic efficiency of 70%

The ideal refrigeration / heat pump cycle is the reversed Carnot cycle. This cycle consists of two reversible isothermal and two isentropic processes. This cycle is totally reversible and as such is sustainable and has the largest performance for a given set of sink and source conditions. It can be derived that

$$COP_{R,Carnot} = \frac{T_c}{T_H - T_c} \quad (5.8)$$

and

$$COP_{HP,Carnot} = \frac{T_H}{T_H - T_C} \quad (5.9)$$

These performance values serve as a (second law) standard against which actual cycles can be compared resulting in the second law efficiency:

$$\eta_R = \frac{COP_R}{COP_{R,Carnot}} \quad (5.10)$$

and

$$\eta_{HP} = \frac{COP_{HP}}{COP_{HP,Carnot}} \quad (5.11)$$

Since the Carnot refrigeration cycle would be sustainable, these values also indicate the sustainability level of the system. Generally, practical refrigeration systems show second law efficiencies below 40%. Figure 5.3 shows, for the conditions of Fig. 5.2, values of the second law efficiency for single stage refrigeration cycles with different refrigerants. Ammonia (NH₃) appears again as the refrigerant with the largest potential to reach high second law efficiencies. Media as R507 are clearly less efficient refrigerants. For larger systems higher efficiencies can be expected due to higher compressor isentropic efficiencies. A global indication of the quality of a refrigeration plant (or heat pump) can be obtained by calculating its second law efficiency. If this efficiency shows a value below 40%, the cycle performance is poor but the cause of malfunction cannot be identified. Only a second law analysis of the cycle will allow for this identification. This will be the topic of section 5.3.

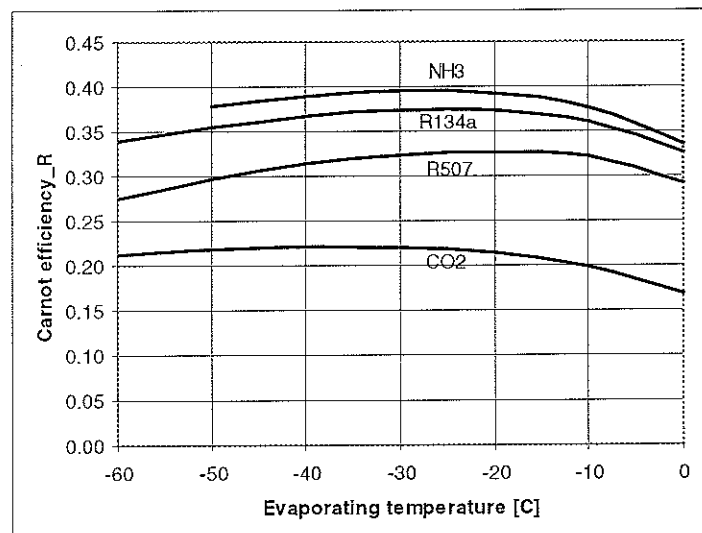


Figure 5.3 Second law (Carnot) efficiency of a single stage refrigeration cycle operating under the conditions of Fig. 5.2 with temperature driving forces of 7.5 K for both sink and source heat exchangers

When evaluating the sustainability of a refrigeration / heat pump cycle, not only its energetic performance plays a role but aspects as contribution to ozone layer depletion, global warming potential, safety and toxicity may determine the sustainability of the cycle. In recent years the total equivalent warming impact (TEWI) of the medium used in the refrigeration / heat pump plant has been adopted to evaluate its sustainability. It includes the direct global warming impact due to emissions of the refrigerant and the indirect global warming impact due to CO₂ emission during the production of the electrical energy needed to drive the cycle. In this way it also includes the energetic performance of the plant.

$$TEWI = GWP * \text{refrigerant_content} * (\text{leak_rate} * i + \text{end_life_leak_percentage}) + \frac{\Phi_{in}}{COP_R} * \text{number_operating_hours} * \frac{CO_2_emission}{kWh} \quad (5.12)$$

In Eq. (5.12) GWP represents the global warming potential of the refrigerant in CO₂ equivalents, *i* is the expected number of operating years, and the number of operating hours includes all operating hours during the life of the system. For an industrial refrigeration plant operating under the conditions of Figs. 5.2 and 5.3, Fig. 5.4 shows the total equivalent warming impact as a function of evaporating temperature. The assumptions under which the figure applies are given in the figure caption. Ammonia is again the best refrigerant with the lowest environmental impact. Refrigerant mixtures R404A and R507 have a 50 to 80% larger environmental impact.

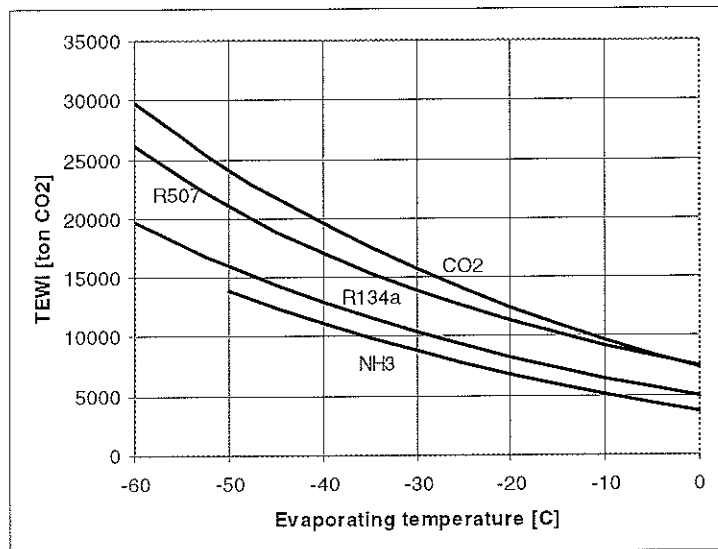


Figure 5.4 Total equivalent warming impact (TEWI) of refrigerating systems with different refrigerants. Assumed refrigerant content 1000 kg, leak rate 5%, end life leak percentage 10%, operational life time 15 years, number of operating hours 2000 hours/year, energy conversion factor 0.58 kg CO₂/kWh

Again this evaluation method allows comparison of alternatives but does not allow for an identification of the reason why a plant shows too high TEWI values. A second law analysis of the cycle is needed if this identification is required. This will be the topic of section 5.3.

5.2 Second law analysis

In the derivation of Eqs. (5.8) and (5.9) it is assumed that the heat sink and heat source temperatures are constant, even though heat is added or removed. This implies sink and heat sources of infinite size. In practical systems the sink and source consist of a flow of a fluid that undergoes a temperature change as heat is exchanged. In refrigeration systems the heat sink is mostly an air flow, eventually indirectly through a water flow. The heat source is also frequently an air flow that undergoes a temperature drop. In food processing plants flows of brines are also common heat sources. Consider, for instance, a continuous freezer where a calcium chloride solution flow is cooled down from -29 to -31°C by evaporating ammonia at -36°C . This is illustrated in Fig. 5.5. Since heat is removed from the calcium chloride solution, its temperature is lower at the outlet T_{out} of the heat exchanger than at the inlet, T_{in} . As it will be shown later, also the refrigerant temperature may change as the refrigerant flow passes the heat exchangers. Here this is less relevant since Eqs. (5.8) and (5.9) refer to the ideal reversible reference process based on the external temperatures imposed to the cycle.

The inlet and outlet calcium chloride temperatures of the continuous freezer are imposed by the food processing process. If a different (complex) cycle, refrigerant or heat exchanger arrangement are used the requirement for the source conditions remains.

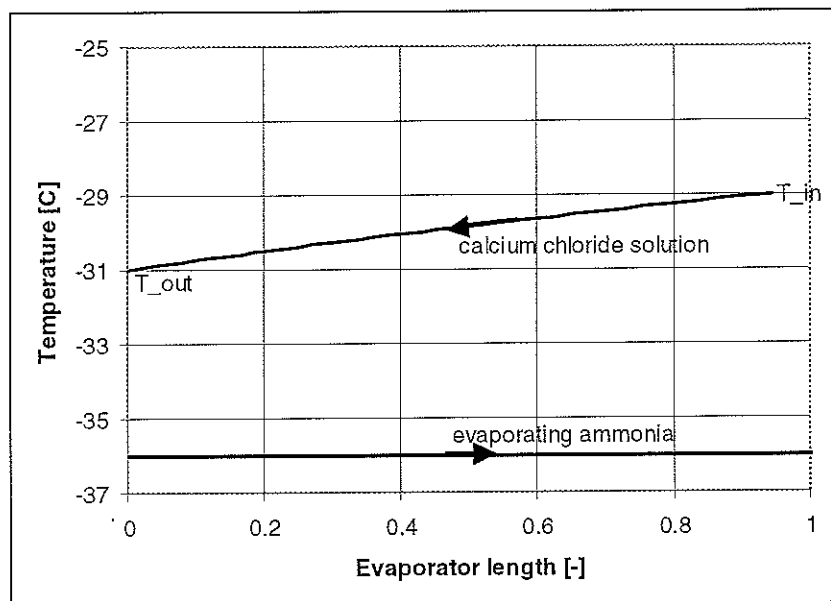


Figure 5.5 Heat source (calcium chloride solution) with a gliding temperature

Similar considerations can be drawn for the heat rejection side. Heat sink and heat source conditions give a constant basis for the comparison of options but their conditions have to consider their gliding character. Toubert [10] derives that in case of gliding sink and source temperatures, the COP of the ideal reversible vapour compression cycle is obtained when the thermodynamic averaged sink and source temperatures are filled in Eqs. (5.8) and (5.9). These temperatures cause the same exergy loss in the condenser and evaporator as the gliding temperatures. The thermodynamic averaged sink and source temperatures are given by:

$$\bar{T}_H = \frac{\Phi_{out}}{\phi_{m_sink} c_{p_sink} \ln(T_{sink_out} / T_{sink_in})} = \frac{T_{sink_out} - T_{sink_in}}{\ln(T_{sink_out} / T_{sink_in})} \quad (5.13)$$

and

$$\bar{T}_C = \frac{\Phi_{in}}{\phi_{m_source} c_{p_source} \ln(T_{source_out} / T_{source_in})} = \frac{T_{source_out} - T_{source_in}}{\ln(T_{source_out} / T_{source_in})} \quad (5.14)$$

Exergy or specific flow availability (Moran & Shapiro [6]) is the work that would be delivered by a reversible process that would bring the flow in equilibrium with the environment conditions. This process consists of an isentropic expansion to environment pressure and an isothermal expansion to the entropy state of the environment. The exergy of a flow with state n can be calculated from:

$$ex_n = (h_n - h_0) - T_0(s_n - s_0) + \frac{c_n^2}{2} + gz_n \quad (5.15)$$

In vapour compression cycles the kinetic and potential energy terms are generally negligibly small in comparison with the other two terms so that Eq. (5.15) simplifies to

$$ex_n = (h_n - h_0) - T_0(s_n - s_0) \quad (5.16)$$

When analyzing vapour compression refrigeration cycles, the environment conditions must be taken as the thermodynamic averaged sink temperature, $\bar{T}_H = T_0$. Generally the environment pressure is the atmospheric pressure, $p_0 = 101.3$ kPa. The environment enthalpy, h_0 and entropy, s_0 must be calculated for the medium under consideration. For the refrigerant state points this implies that

$$h_0 = f(T_0, p_0) \quad (5.17)$$

and

$$s_0 = f(T_0, p_0) \quad (5.18)$$

must be calculated from the (superheated vapour) data tables of the refrigerant. Assuming constant specific heat values for the heat sink and heat source media and neglecting pressure differences between air and surrounding, the exergy values for the sink and source media can be approached with

$$ex_{sink} = c_{p_sink} (T_{sink} - T_0) - T_0 c_{p_sink} \ln(T_{sink} / T_0) \quad (5.19)$$

5.3 Exergy analysis of a refrigeration cycle

The rate of availability destruction or exergy loss, Ex_{loss} , for each of the components of a vapour compression refrigeration cycle allows for an identification of the sources of malfunction within the cycle. When the specific exergy values for each state are known, a simple exergy balance will allow for the calculation of the exergy loss of a component. An analysis of the critical design components becomes then possible. In the following sections each main component will be analyzed separately, following the state numbering given in Fig. 5.1.

5.3.1 Condenser

Fig. 5.6 shows the conditions around the condenser. The condenser is considered externally adiabatic, so that there is no heat transfer with the surroundings. The refrigerant coming from the compressor with state 2 is desuperheated and condensed to state 3 at the condenser outlet. In countercurrent flow, the heat sink medium (water or air) is heated from $T_{\text{sink_in}}$ to $T_{\text{sink_out}}$. Between the two media, the heat flow Φ_{out} is exchanged. The exergy loss in the condenser is obtained from

$$Ex_{\text{loss_cond}} = \phi_{m_ref} (ex_2 - ex_3) + \phi_{m_sink} (ex_{\text{sink_in}} - ex_{\text{sink_out}}) \quad (5.20)$$

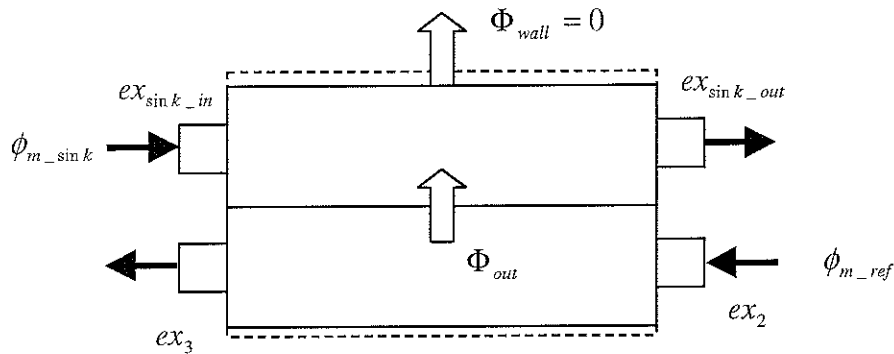


Figure 5.6 Conditions around the condenser

5.3.2 Evaporator

Similar considerations can be made for the evaporator. The situation is schematically shown in Fig. 5.7.

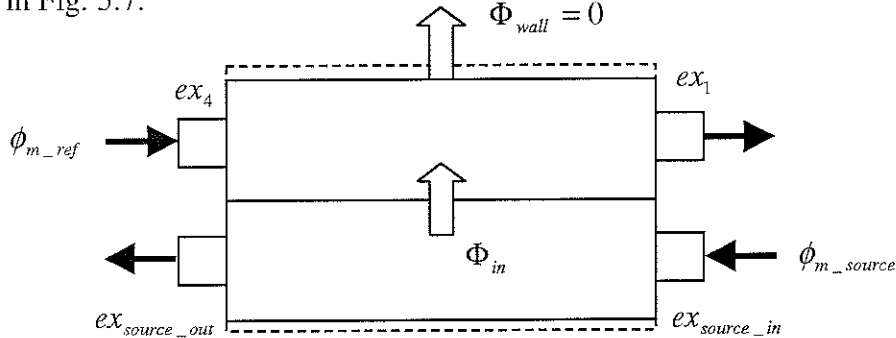


Figure 5.7 Conditions around the evaporator

An analysis of Fig. 5.7 gives for the exergy loss in the evaporator

$$Ex_{\text{loss_evap}} = \phi_{m_ref} (ex_4 - ex_1) + \phi_{m_source} (ex_{\text{source_in}} - ex_{\text{source_out}}) \quad (5.21)$$

The useful exergy effect of the cycle is the exergy flow change of the heat source medium:

$$Ex_{\text{rise_source_flow}} = \phi_{m_source} (ex_{\text{source_out}} - ex_{\text{source_in}}) \quad (5.22)$$

5.3.3 Compressor

Again the compressor is considered to be externally adiabatic, what is a reasonable assumption for the larger compressors without heat removal. Screw compressors for refrigeration applications are generally oil injected. In the analysis of such compressors, the exergy flows associated with the oil flow must also be included in the analysis.

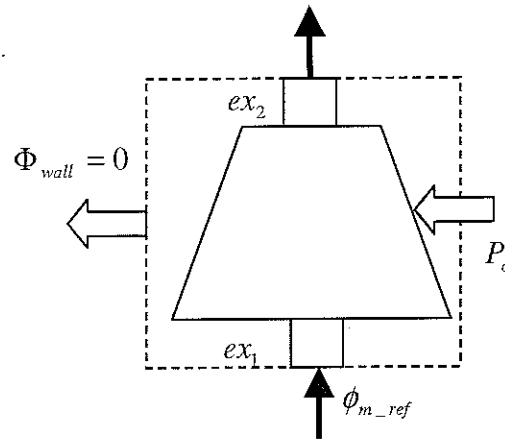


Figure 5.8 Conditions around the compressor

The exergy loss in the compressor becomes:

$$Ex_{loss_comp} = \phi_{m_ref} (ex_1 - ex_2) + P_c \quad (5.23)$$

5.3.4 Throttling device

Similarly for the expansion device:

$$Ex_{loss_exp} = \phi_{m_r} (ex_3 - ex_4) - P_t \quad (5.24)$$

Generally there will be no work delivery to the surroundings so that $P_t = 0$.

5.3.5 Exergy efficiency of the cycle

The exergy efficiency of a vapour compression refrigeration cycle is defined as the ratio between objective of the cycle (useful exergy effect in evaporator) and exergy input to the process:

$$\eta_{R_2nd_law} = \frac{Ex_{rise_source_flow}}{P_c - P_t} \quad (5.25)$$

The difference between the exergy input ($P_c - P_t$) and the exergy rise of the source flow is equal to the sum of exergy losses per component. This allows for the construction of Sankey diagrams that visualize the contribution of the different components to the total exergy losses of the system. This is illustrated in Fig. 5.9.

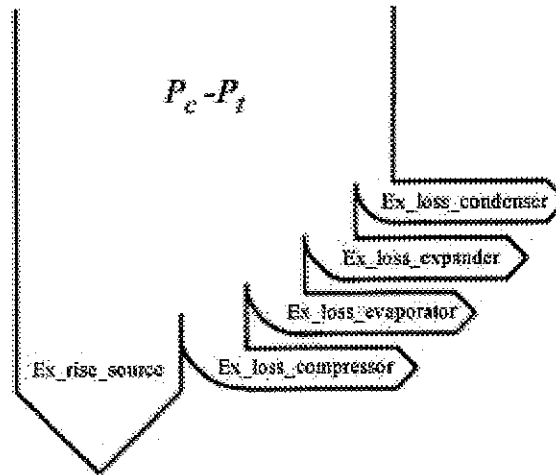


Figure 5.9 Sankey diagram illustrating how the different losses are visualized

5.3.6 Application example

A vapour compression refrigeration system, using R134a as the working fluid, operates under the conditions listed in Table 5.1. The condenser is a water cooled condenser in which Φ_{out} is rejected to a water flow. The heat source is dry air circulated within a refrigerated room. All latent heat effects associated with the air flow should be neglected. The refrigerant passes the condenser with negligible pressure drop but undergoes a significant pressure drop as it passes the evaporator. The data have been taken from a real plant. Consider all components and refrigerant lines to be externally adiabatic and kinetic and potential energy effects to be negligible. Make an exergy analysis of the plant and compare this analysis with the results of a first law analysis.

Table 5.1 Operating conditions of example refrigeration cycle

Medium	Variable	Value
Sink	Φ_{out}	5.804 kW
	T_{sink_in}	22.1°C
	ϕ_{m_sink}	0.283 kg/s
	c_{p_sink}	4.187 kJ/KgK
Source	Φ_{in}	4.6 kW
	T_{source_in}	3.3°C
	ϕ_{m_source}	1.61 kg/s
	c_{p_source}	1.006 kJ/kgK
Refrigerant	$T_{condensation}$	26.0°C
	$T_{condenser_out}$	24.9°C
	$T_{evaporator_in}$	-16.0°C
	$T_{evaporator_out}$ (saturation)	-17.6°C
	$T_{evaporator_out}$	0.3°C
	$T_{compressor_out}$	60.1°C

The first step in the analysis is locating the principal operating states of the refrigerant 134a cycle in T-s and p-h diagrams. Pressure – enthalpy diagrams are most widely used in the refrigeration field but for convenience also the T-s diagram will also be drawn here. For this analysis REFPROP [7], a refrigerant properties calculation program that includes a variety of refrigerants, has been used. Alternatively saturated and superheated vapour tables for R134a may be used. Data for a confined number of refrigerants can be found in Moran & Shapiro [6]. An extensive number of refrigerants can be found in ASHRAE [1].

The pressure at the high pressure side of the cycle is the saturation pressure corresponding to the condensation temperature (26°C), or $p_2 = p_3 = 685.3$ kPa. State 2 is fixed by p_2 and $T_2 = T_{compressor_out} = 60.1^\circ\text{C}$: $h_2 = 446.9$ kJ/kg and $s_2 = 1.824$ kJ/kgK. State 3 is fixed by p_2 and $T_3 = T_{condenser_out} = 24.9^\circ\text{C}$: $h_3 = 234.3$ kJ/kg and $s_3 = 1.119$ kJ/kgK. The refrigerant state change in de condenser from stage 2 to stage 3 is given in Fig. 5.10.

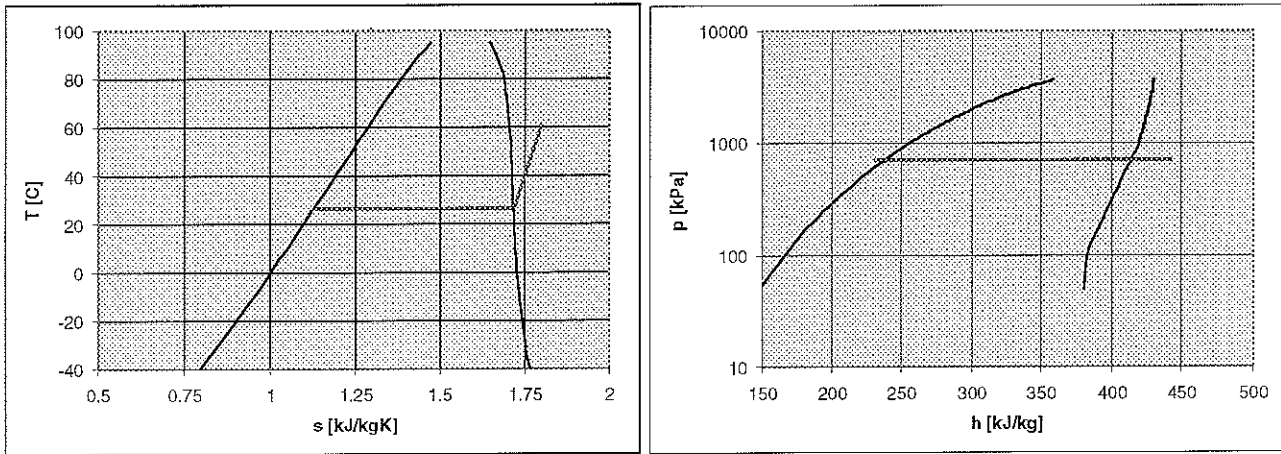


Figure 5.10 State change in condenser in T - s diagram (left) and p - h diagram (right). The line represents part of the 685.3 kPa isobar in both diagrams.

The pressure at the low pressure side of the cycle changes through the evaporator and is, at the inlet of the evaporator, the saturation pressure corresponding to the inlet saturation temperature (-16.0°C), or $p_4 = 157.2$ kPa and, at the outlet of the evaporator, the saturation pressure corresponding to the outlet saturation temperature (-17.6°C), or $p_1 = 147.0$ kPa. State 1 is fixed by p_1 and $T_1 = T_{\text{evaporator_out}} = 0.3^{\circ}\text{C}$: $h_1 = 402.8$ kJ/kg and $s_1 = 1.795$ kJ/kgK. The expansion through the valve is a throttling process implying that $h_4 = h_3 = 234.3$ kJ/kg. The entropy of state 4 follows from the local quality:

$$s_4 = s_{L_{157.2}} + \frac{h_3 - h_{L_{157.2}}}{h_{V_{157.2}} - h_{L_{157.2}}} (s_{V_{157.2}} - s_{L_{157.2}}) \quad (5.26)$$

with the index V indicating saturated vapour conditions and L saturated liquid conditions. From Eq. (5.26) $s_4 = 1.136$ kJ/kgK. The refrigerant state change in the evaporator from stage 4 to stage 1 is given in Fig. 5.11.

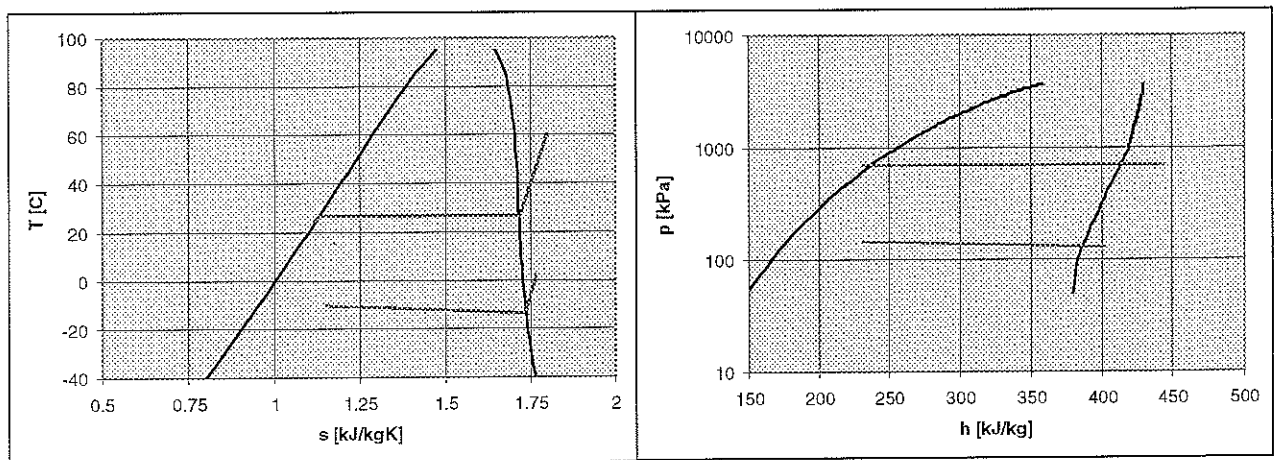


Figure 5.11 The line represents the process in the evaporator in both diagrams, considering a pressure drop from $p_4 = 157.2$ kPa to $p_1 = 147.0$ kPa

The processes in expansion valve and compressor can now be obtained by connecting state 3 to state 4 and state 1 to state 2, respectively. This is shown in Fig. 5.12.

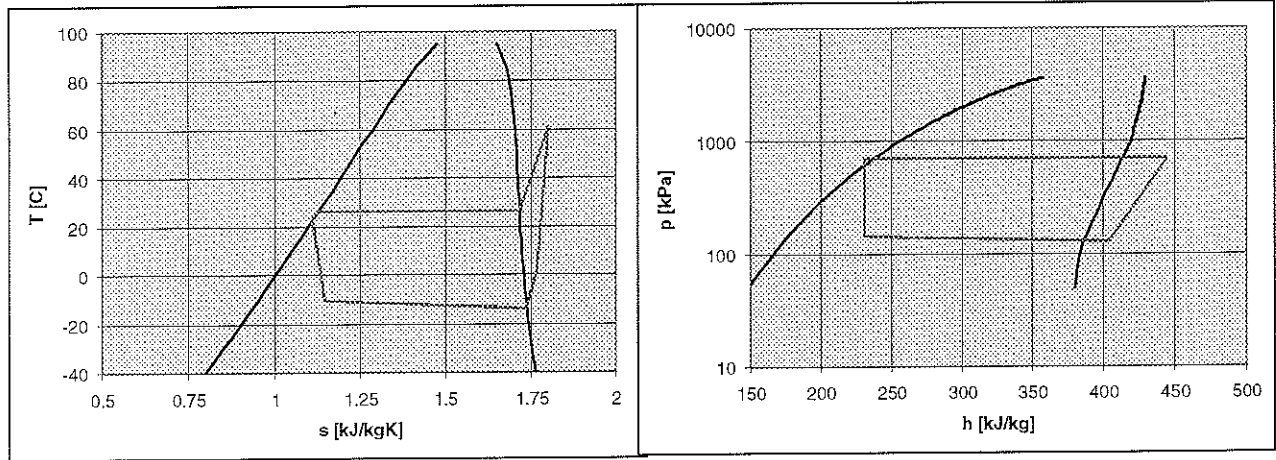


Figure 5.12 State change in cycle in T - s diagram (left) and p - h diagram (right)

The sink medium outlet temperature can be obtained from

$$T_{\text{sink_out}} = T_{\text{sink_in}} + \frac{\Phi_{\text{out}}}{\dot{\phi}_{m_ \text{sink}} * c_{p_ \text{sink}}} \quad (5.27)$$

With the data of Table 5.1, $T_{\text{sink_in}} = 22.1 + 273.15 = 295.25$ K and $T_{\text{sink_out}} = 300.14$ K. The thermodynamic averaged sink temperature follows from Eq. (5.13): $\bar{T}_H = 297.69$ K. Similarly, the source medium outlet temperature can be obtained from

$$T_{\text{source_out}} = T_{\text{source_in}} - \frac{\Phi_{\text{in}}}{\dot{\phi}_{m_ \text{source}} * c_{p_ \text{source}}} \quad (5.28)$$

With the data of Table 5.1, $T_{\text{source_in}} = 0.3 + 273.15 = 273.45$ K and $T_{\text{source_out}} = 273.61$ K. The thermodynamic averaged source temperature follows from Eq. (5.14): $\bar{T}_C = 275.03$ K. The first law of thermodynamics gives

$$P_c = \Phi_{\text{out}} - \Phi_{\text{in}} \quad (5.29)$$

Equations (5.6) and (5.8) allow for calculation of the COP of the refrigeration cycle $COP_R = 3.82$ and of the reversible refrigeration cycle operating under similar external conditions $COP_{R_Carnot} = 12.14$, respectively. The second law efficiency is, with eq. (5.10), 31%.

In case of isentropic compression the discharge conditions are fixed by p_2 and $s_{2s} = s_1 = 1.795$ kJ/kgK: $h_{2s} = 437.4$ kJ/kg. The isentropic efficiency of the compressor follows then from Eq. (5.4): $\eta_c = 0.785$, what is a quite reasonable value for such a relatively small compressor.

Application of Eq. (5.16) for the refrigerant states and Eq. (5.19) for sink and source media gives the results listed in Table 5.2.

Table 5.2 Overview of state conditions and calculated exergy values.

State	T [K]	h [kJ/kg]	s [kJ/kgK]	ex [kJ/kg]
1		402.8	1.795	9.56
2s		437.4	1.795	44.16
2		446.9	1.824	45.03
3		234.3	1.119	42.30
4		234.3	1.136	37.24
sink_in	295.25			0.042
sink_out	300.14			0.042
source_in	276.45			0.801
source_out	273.61			1.036
0	297.69	424.2	1.899	

Equations (5.20) through (5.24) can now be used to evaluate the exergy losses in each component. The mass flow of refrigerant can be evaluated from, for example, Eq. (5.1). The results are shown in Table 5.3.

Table 5.3 Overview of exergy losses in application example components.

	Exergy loss [kW]	Exergy loss [%]
Ex_{loss_cond}	0.075	6.2
Ex_{loss_exp}	0.138	11.5
Ex_{loss_evap}	0.377	31.3
Ex_{loss_comp}	0.236	19.6
$Ex_{rise_source_flow}$	0.379	31.5
$Ex_{input} = P_c$	1.204	100.0

These results are illustrated in the pie diagram of Fig. 5.13.

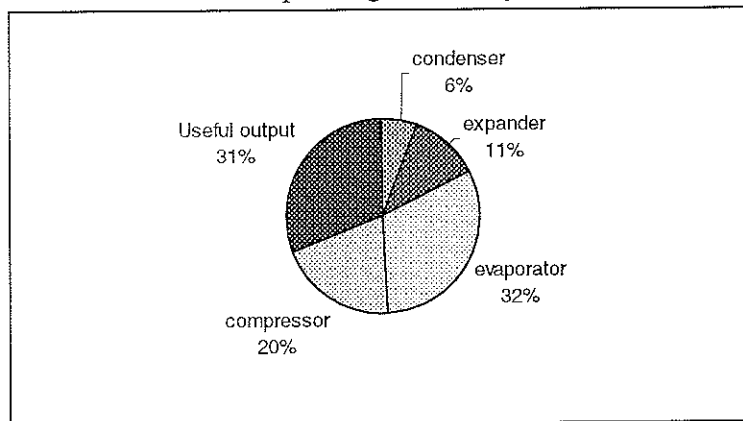


Figure 5.13 Distribution of the exergy losses through the different components of the refrigeration system of the application example. The evaporator gives the largest contribution.

When Eq. (5.25) is filled in, it appears that the exergy efficiency of the cycle corresponds to the second law efficiency calculated before: 31%. Table 5.3 and Fig. 5.13 show that, in addition to the global quality of the performance of the cycle given by the second law efficiency, an exergy analysis allows for identification of the components that give the largest contribution to deviation from ideal behavior. In this case the evaporator followed by the compressor. In the next section the causes of irreversibility per component will be discussed per component.

5.3.7 Impact of malfunction corrections

The exergy losses in heat exchangers will be discussed by in Chapter 7. The condenser operates around environmental conditions so that heat transfer to the surroundings is negligible. The evaporator is installed in a refrigerated room and its purpose is to exchange heat with the surrounding air. The refrigerant side pressure drop in the evaporator is non-negligible but is already accounted for in the conditions of state 1, at the outlet of the evaporator. The main cause of irreversibility in the heat exchangers is heat transfer between sink and condensing medium and between source and evaporating medium.

Figure 5.14 illustrates the conditions around the evaporator as listed in Table 5.1.

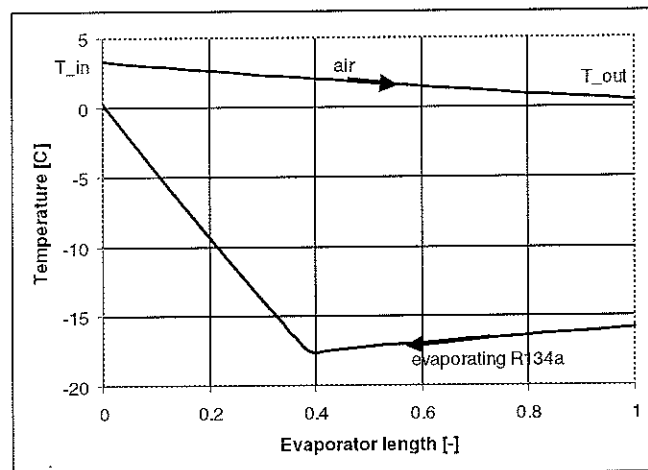


Figure 5.14 Temperature change of refrigerant and source medium (air) through the evaporator.

It appears that the temperature driving force for heat transfer is rather large. In this case the malfunction of the superheating control associated with a wrong design of the refrigerant distribution is leading to this large temperature driving forces. Assuming that the maintenance personnel can fix this malfunction the situation schematically illustrated in Fig. 5.15 could result. Now states 4 and 1 have different conditions. The pressure at the low pressure side of the cycle is, at the inlet of the evaporator, the saturation pressure corresponding to the inlet saturation temperature (-4.0°C), or $p_4 = 252.6 \text{ kPa}$ and, at the outlet of the evaporator, the saturation pressure corresponding to the outlet saturation temperature (-5.6°C), or $p_1 = 237.8 \text{ kPa}$. State 1 is fixed by p_1 and $T_1 = T_{\text{evaporator_out}} = 0.3^{\circ}\text{C}$: $h_1 = 400.5 \text{ kJ/kg}$ and $s_1 = 1.749 \text{ kJ/kgK}$. The expansion through the valve is a throttling process implying that $h_4 = h_3 = 234.3 \text{ kJ/kg}$. The

entropy of state 4 follows from the local quality (eq. (5.26)): $s_4 = 1.128$ kJ/kgK. Assuming that the isentropic efficiency remains the same ($\eta_c = 0.785$), also the state at the compressor discharge changes: $h_2 = 429.0$ kJ/kg and $s_2 = 1.769$ kJ/kgK.

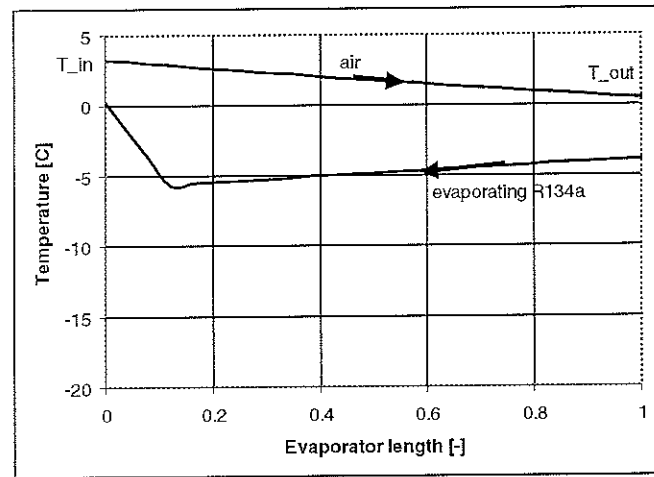


Figure 5.15 Temperature change of refrigerant and source medium (air) through the evaporator after correction of refrigerant distribution and control malfunction.

The effects of the correction of this malfunction are given in Table 5.4.

Table 5.4 Overview of exergy losses in application example components after correction of the malfunction.

	Exergy loss [kW]	Exergy loss [%]
Ex_{loss_cond}	0.033	4.2
Ex_{loss_exp}	0.074	9.4
Ex_{loss_evap}	0.138	17.4
Ex_{loss_comp}	0.165	20.9
$Ex_{rise_source_flow}$	0.379	48.0
$Ex_{input} = P_c$	0.789	100.0

The effect of the correction is quite clear both for first and second law performance. The COP increases from 3.82 to 5.83. The second law efficiency from 31% to 48%.

After this modification of the system, the compressor appears now to be the main cause of non-ideal system behavior. The compressor in this plant has already a reasonable isentropic efficiency. Nevertheless, it would be possible to substitute the compressor by a compressor with a slightly higher performance of 85%. If this compressor would be installed than an exergy analysis of the resulting system gives the results listed in Table 5.5.

Table 5.5 Overview of exergy losses in application example components after correction of the malfunction and substitution of compressor by one with 85% isentropic efficiency.

	Exergy loss [kW]	Exergy loss [%]
Ex_{loss_cond}	0.032	4.3
Ex_{loss_exp}	0.074	10.2
Ex_{loss_evap}	0.138	18.9
Ex_{loss_comp}	0.107	14.7
$Ex_{rise_source_flow}$	0.379	52.0
$Ex_{input} = P_c$	0.729	100.0

A relatively small step in isentropic efficiency of the compressor has a significant impact in the system performance: the COP goes from 5.83 to 6.31 (8% improvement), the second law efficiency from 48% to 52% (also 8% improvement).

5.4 Exergy analysis of refrigeration plant

In the previous sections the exergy losses in refrigeration cycles have been considered. The exergy entering the cycle in the evaporator is negligible in comparison with the exergy entering the system in the form of compressor shaft power requirement.

When the refrigeration plant is considered, globally there are three main exergy in-flows: electric energy input for the compressor drive, the condenser fans and / or pump(s) and the evaporator fans and / or pump(s). The useful effect of the process corresponds with the exergy increase of the air flow passing the evaporator. An exergy increase which corresponds with the exergy reduction of the same air flow due to the heat load to the storage room. The exergy input to the condenser and evaporator fans is lost. Part of the exergy input is lost directly due to electric motor losses. The rest is lost due to mixing and friction with the storage room air (evaporator) or surrounding air (condenser). A small amount of the exergy input to the compressor drive is usefully applied to remove the heat load. The rest is lost in the different components of the refrigeration plant.

Mostly plants are designed to handle extreme operating conditions. The result is that most of the time plants operate under part load conditions. If the plant switches on and off then a dynamic analysis of the plant is needed to be able to quantify the impact of the different exergy inputs as a function of time. An example of possible exergy loss distribution is given in Fig. 5.16, taken from Infante Ferreira et al [5]. The figure applies for the integrated exergy flows for a period of 24 hours. The integration is needed to quantify the effect of frost formation and continuous operation of the evaporator fans. The continuous operation of the evaporator fans is needed to guarantee the product quality. The useful effect is only 9% of the total exergy input. The largest losses take place in: Air cooler fans (41%); Electric motor of compressor drive (16%); Condenser (14%); Compressor (7%); Condenser fans (5%).

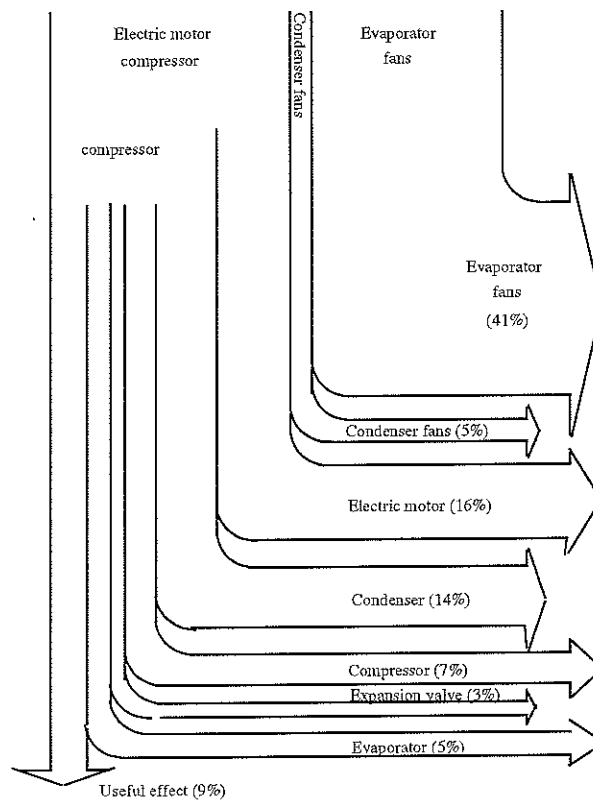


Figure 5.16 Exergy loss distribution in fruit storage (Infante Ferreira et al [2]).

Some researchers propose that the required input in Eq. (5.5) should include the auxiliary energy inputs to the system: to drive evaporator and condenser fans and / or pumps and required control systems. A so-called system COP, COSP, is then obtained. In that case, the results obtained with Eq. (5.10) are significantly lower. If the power usage is not simultaneous, the ratios of operating hours must be considered. The example of Fig. 5.16 illustrates that when studying the exergy losses, an integrated approach is needed to allow for the identification of the main causes of exergy losses. For this plant application of two-speed evaporator fans, that operate at high speed as the refrigeration cycle is in operation and at low speed if only air circulation is required to guarantee an acceptable product quality, could lead to significant exergy (and energy) performance enhancement of the system.

5.5 Reducing exergy losses

5.5.1 Cycle level

In Section 5.1 it has been illustrated that the choice of the operating medium has to some extent effect on the level of the second law efficiency that can be attained. In Fig. 5.3 only single stage operation under a determined set of external conditions is shown,

so that other specific conditions will give different results. Ammonia and n-pentane show the best performance. Then a group of refrigerants show similar performance: isobutene (R600a), propane (R290), R134a and R410A. The refrigerants R404A and R507, quite popular in the Netherlands, definitely are thermodynamically less efficient and should be avoided. When used at low temperatures in combination (cascade or secondary fluid arrangement) with ammonia at the high temperature side, carbon dioxide attains performances comparable with the R134a group of refrigerants (Infante Ferreira [3]).

Although it has not been discussed here, when the temperature lift between source and sink is large, multi-stage operation should be considered.

Subcooling has always a positive and large effect on the efficiency of the vapour compression cycle. This is illustrated in Fig. 5.17 for a number of refrigerants. For every refrigeration system the possibility of creating extra subcooling while the condensing temperature is maintained should be investigated.

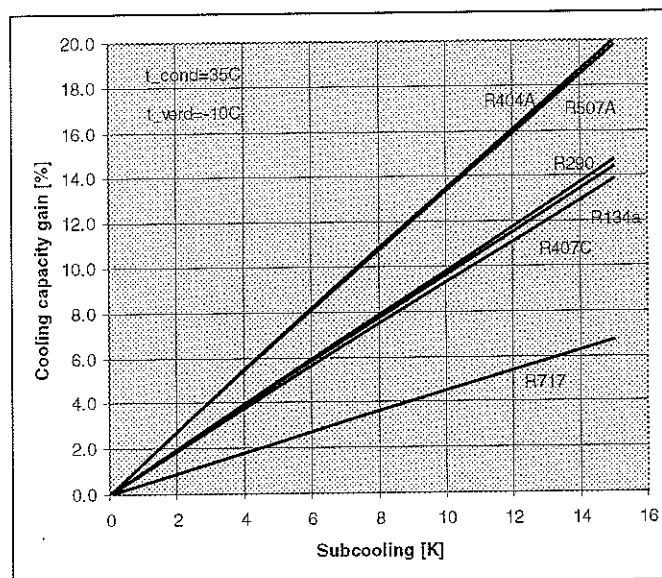


Figure 5.17 Effect of subcooling on cooling capacity (and COP) for a condensation temperature of 35°C and evaporating temperature of -10°C.

5.5.2 Condenser

The main cause of exergy loss in the condenser is the heat transfer at finite temperature difference between refrigerant and sink medium. This if the exergy loss associated with sink-fluid forced convection is not considered. These exergy losses may also be substantial: in the case of Fig. 5.16 5% of the total exergy losses of the plant compared with 14% due to heat transfer.

The temperature difference between condensing refrigerant and sink medium will always be a trade off between energy consumption (and associated exergy losses) and investment costs.

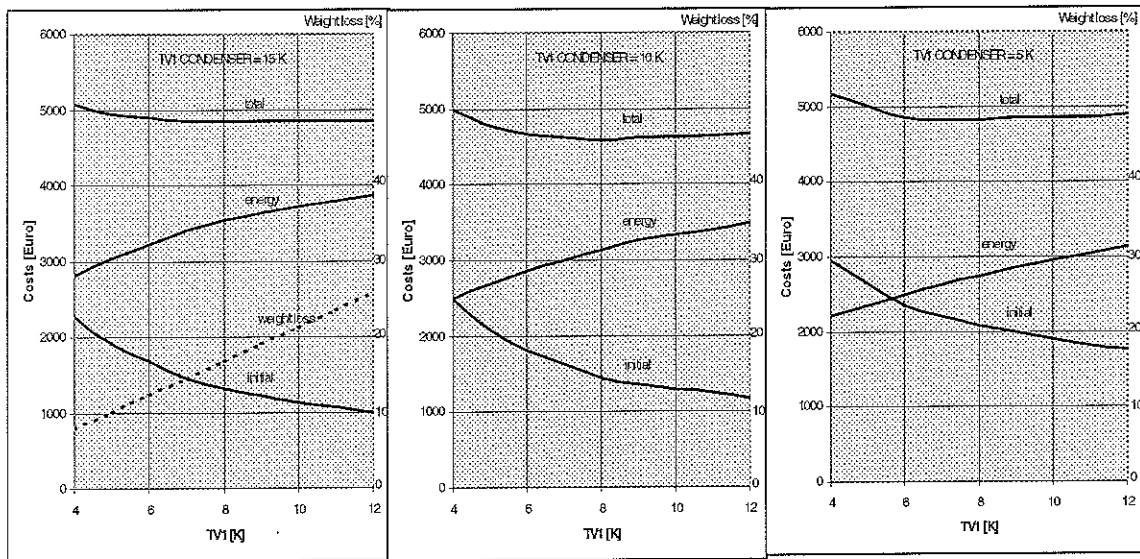


Figure 5.18 Temperature driving force and trade-off between energy and investment.

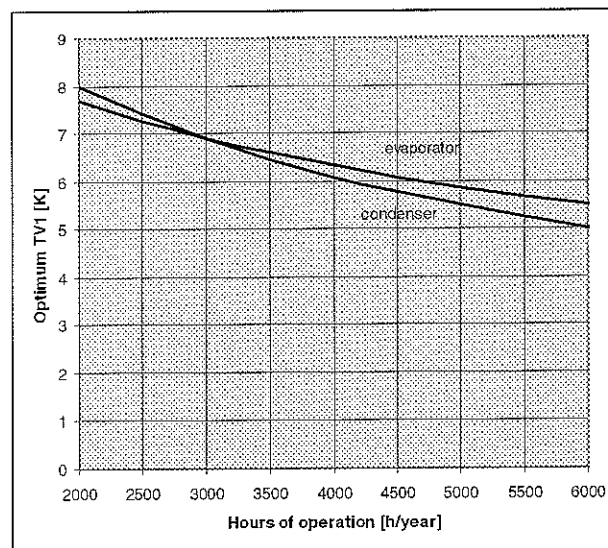


Figure 5.19 Optimum temperature driving force for condenser and evaporator for a specific application.

Fig. 5.17 illustrates, for specific conditions of a plant using an air cooled condenser, that if the temperature difference is chosen too large (15K between air inlet and condensation temperature) or too low (5K between air inlet and condensation temperature) the total costs of energy + investment are larger than for an intermediate case (10K between air inlet and condensation temperature). In the first case the energy costs are too large while in the second case the investment costs are too large. It should be noted that the common practice in the Netherlands is designing condensers with 15K between air inlet and condensation temperature. Fig. 5.18 illustrates that the number of operating hours of the refrigeration plant plays an important role in the optimum temperature driving force for both condenser and evaporator.

5.5.3 Evaporator

Similar considerations can be applied to the evaporator as for the condenser. Also here the trade off between energy and investment costs play a role. Figs. 5.17 and 5.18 illustrate also the impact of evaporator driving force. However, since the temperature driving force also plays a role in the product quality, the impact of temperature driving force on product quality must also be considered.

The quantity of water vapour removed by the evaporator from a refrigerated / freezing room depends on the temperature driving force. With temperature differences of 5-6 K between air inlet and evaporating medium there will be almost no water vapour removal while for temperature driving forces of 10 K 20 to 35% of the heat removal will be associated with latent heat effects. Products sensible to dry out may then loose weight and quality. The impact of evaporating temperature driving force on weight loss, for long term fruit storage applications is also shown in Fig. 5.17.

In other cases in freezing processes a large temperature driving force is used to guarantee a high frozen product quality. This is for instance the case for consumption ice manufacturing, where extreme large temperature driving forces are applied. The need for these extreme driving forces should be investigated: what is the impact of driving force on crystal size and what the impact of crystal size on ice taste.

5.5.4 Compressor

There are several causes for the irreversibilities in compressors: internal flow losses, internal heat transfer and heat conduction, mechanical friction, internal leakage, mixing and external convection and radiation. Depending on the compressor type and size the contribution of the different effects may be quite different. The food processing plant designer and owner is confronted with existing compressor designs so that the exergy losses of the compressor are associated with the compressor selection. Infante Ferreira and Toubert [4] give an overview of the isentropic efficiency of the refrigerating compressors available in the Dutch market. In some application ranges there is large spread in compressor quality. This is illustrated in Table 5.6.

The average values have been printed bold. The other two values indicate the minimum and maximum values and show the isentropic efficiency range within the same compressor size group. The isentropic efficiency of semi-hermetic and hermetic compressors includes the electric motor losses. Against the expectations, the larger compressors do not show higher isentropic efficiencies than the smaller models.

Table 5.6 *Isentropic efficiency as a function of refrigerant volume flow (R22 evaporating at -10°C).*

Volume flow [m ³ /h]	Reciprocating			Screw		Scroll
	Open	Semi-hermetic	Hermetic	Open	Semi-hermetic	Hermetic
10-20	0.88 0.88 0.88	0.63 0.70 0.79	0.60 0.68 0.72			0.63 0.67 0.71
20-50	0.77 0.81 0.88	0.68 0.73 0.82	0.57 0.59 0.61			
50-100	0.81 0.85 0.83	0.73 0.77 0.81		0.71 0.71 0.71		
100-200	0.70 0.77 0.81	0.70 0.74 0.79		0.71 0.73 0.75	0.63 0.65 0.66	
200-300	0.70 0.75 0.80	0.70 0.74 0.76		0.75 0.75 0.75	0.66 0.66 0.66	
300-500	0.71 0.71 0.71			0.64 0.68 0.70		
500-1000	0.72 0.72 0.72			0.70 0.71 0.73		

5.5.5 Expansion device

In recent years two-phase expanders have been the topic of a number of researches. In some applications two-phase expanders appear to decrease significantly the exergy losses and so the performance of the vapour compression refrigeration / heat pump cycle. Fig. 5.20 shows for the some refrigerants the effect of expander isentropic efficiency on the COP for a heat pump cycle.

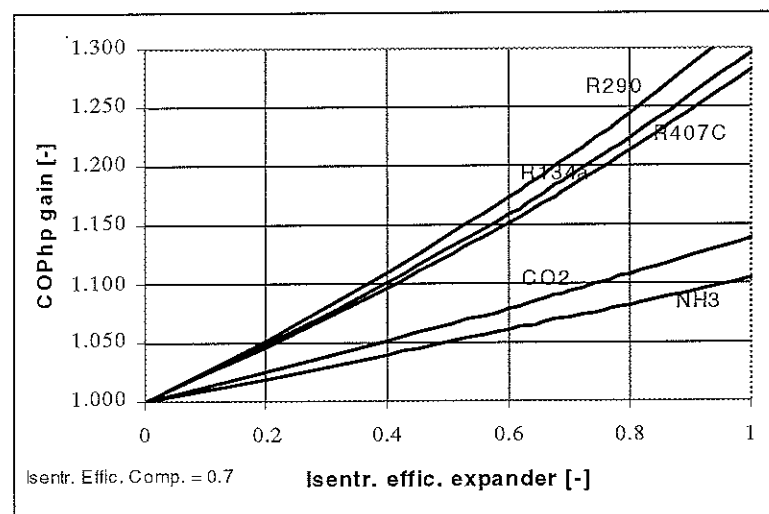


Figure 5.20 *COP gain for a heat pump when an expander is used to recover (part of) the expansion work*

If isentropic efficiencies above 60% can be attained with practical expanders then significant cycle performance gains can be obtained for the refrigerants in the R134a and propane (R290) group.

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6 Value diagrams and exergy efficiencies

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6.1 Introduction

The exergy analysis of plants, for the conversion of energy or the production of chemical substances, requires in general the determination of all exergy flows that are transferred between the distinguished apparatuses or unit operations of such a plant. The resulting exergy losses can provide useful information with regard to the overall performance of the considered plant. However it is in general difficult to judge thermodynamic losses without further reference. An evaluation of plant performance will usually require a comparison of the thermodynamic performance of specific apparatuses or unit operations with available data from previously built plants. Therefore exergy efficiencies have appeared to be very useful. However exergy efficiencies are not only useful at the level of apparatuses or unit operations but also at higher system levels e.g. for the comparison of overall plant performance (total system) or the performance of subsystems. The combination of exergy efficiencies with exergy flow diagrams (Grassmann diagrams) can briefly provide an overview of the most important information with regard to the thermodynamic performance of even complex plants.

6.2 The value diagram for heat transfer

Q, T -diagrams are frequently used to present heat transfer processes. In Fig. 6.1a the temperature of both flows in a heat exchanger is given as a function of the heat transferred to the primary flow, the flow that is the heated flow. In Fig. 6.1b the temperature curves of the same flows are shown in the value diagram. In this diagram the temperature of the flows are also given as a function of the heat transferred to the primary flow; but the temperature at the vertical axis is replaced here by the term $(1 - T_0/T)$. As this axis begins at $T = T_0$ and goes up to $T = \infty$, the values on this axis can go from 0 to 1. If it is assumed that an infinitesimal small amount of heat dQ is transferred from the secondary flow, the flow that is cooled down in the heat exchanger, the resulting decrease in temperature dT_s may be neglected. For the exergy of this amount of heat can be written:

$$dEx_s = \left(1 - \frac{T_0}{T_s}\right) \cdot dQ \quad (6.1)$$

In the value diagram the area 1-3-4-6-1 equals the amount of heat dQ , whereas the area 1-2'-5'-6-1 equals the exergy dEx_s of this amount of heat. The term $(1 - T_0/T_s)$ indicates which part of the considered heat can be converted into work and can be seen as the exergy fraction of this amount of heat.

The total exergy transferred from the secondary flow can be determined by integrating Eq. (6.1) from the inlet temperature $T_{s,in}$ to the outlet temperature $T_{s,out}$:

$$Ex_s = \int_{T_{s,in}}^{T_{s,out}} \left(1 - \frac{T_0}{T_s}\right) \cdot dQ \quad (6.2)$$

This amount of exergy equals the whole area below the temperature curve in the value diagram. Within the heat exchanger the heat dQ is transferred to the primary flow. The exergy of the heat supplied to this flow equals:

$$dEx_s = \left(1 - \frac{T_0}{T_s}\right) \cdot dQ \quad (6.3)$$

In the value diagram this exergy is represented by the area 1-2-5-6-1. This amount of exergy is smaller than the exergy transferred from the secondary flow. The difference, the area 2-2'-5'-5-2, is exergy that is lost due to the temperature difference necessary to transfer heat from the secondary to the primary flow. The total exergy absorbed by the primary flow can be calculated by integrating from inlet to outlet:

$$Ex_p = \int_{T_{p,in}}^{T_{p,out}} \left(1 - \frac{T_0}{T_p}\right) \cdot dQ \quad (6.4)$$

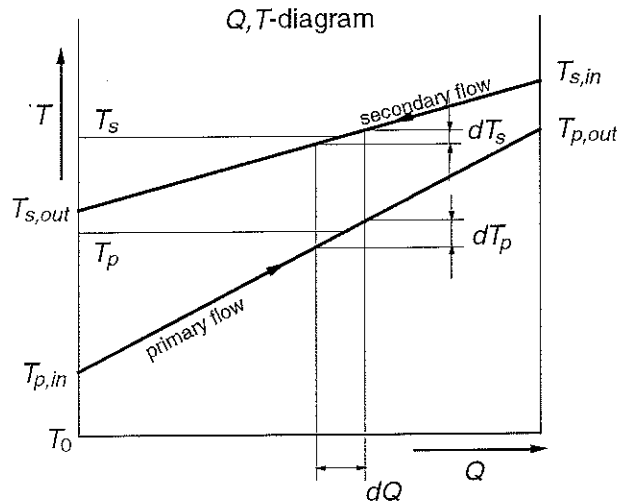


Figure 6.1a Heat transfer process represented in a Q, T -diagram

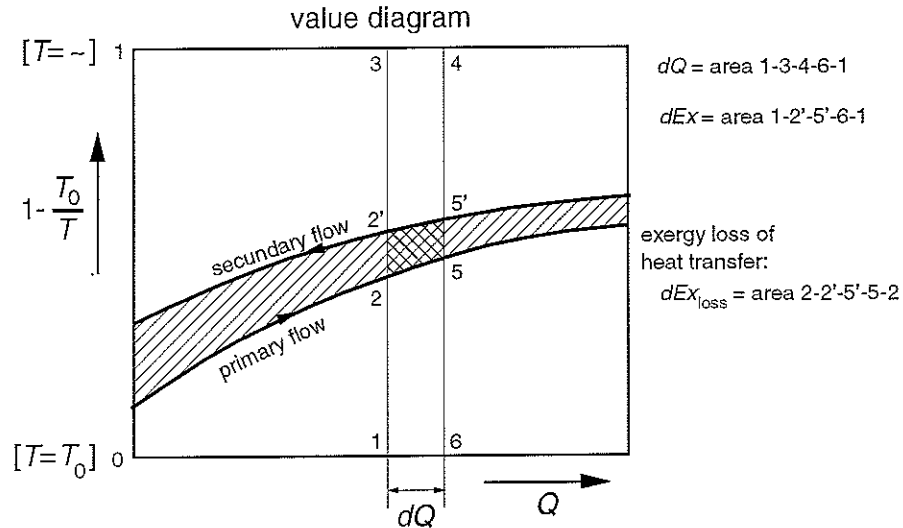


Figure 6.1b Heat transfer process represented in a value diagram

The absorbed exergy by the primary flow equals the area below the temperature curve for this flow in the value diagram. The total exergy loss due to heat transfer, $Ex_s - Ex_p$, is represented by the area between the two temperature curves. Therefore value diagrams easily present amounts of exergy and exergy losses during heat transfer processes. From this diagram it will also be clear that an increase in temperature difference between the flows that exchange heat will increase the exergy loss due to heat transfer. The temperature difference can be seen as the driving force for the heat transfer process; increasing the driving force will accelerate the process, so that the size of the heat exchanger can be reduced, but causes higher exergy losses.

Like Q, T -diagrams, value diagrams can also be used to present the heat transfer to a flow when this heat transfer takes place in more heat exchangers. Figure 6.2 shows the Q, T -diagram for the heat transfer in the feedwater train of a large steam turbine plant as an example. In this feedwater train feedwater is preheated in a series of eight extraction preheaters. The Q, T -diagram shows the temperature curves of both the secondary and the primary fluids. The applied temperatures and extraction pressures are optimized to minimize overall exergy loss. The diagram clearly shows the applied temperature differences for heat transfer. In particular temperature differences in the low temperature heat exchangers should be limited.

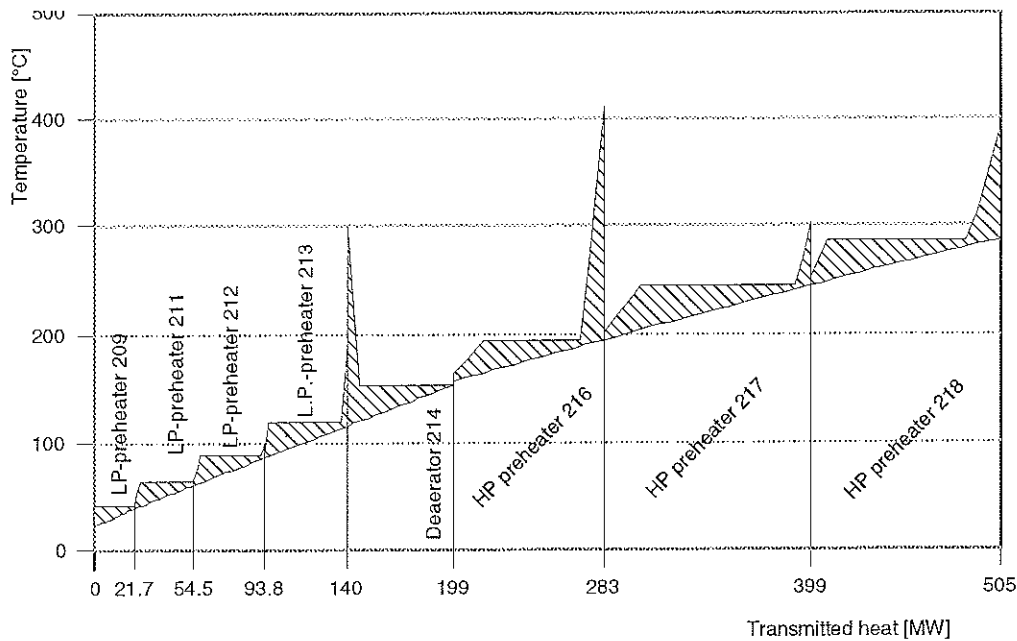


Figure 6.2 *Q,T-diagram for the heat transfer in a feedwater train*

The value diagram for this feedwater train is shown in Fig. 6.3. The shaded areas in this diagram represent the exergy loss caused by the heat transfer in the feedwater train. From this diagram it becomes clear why the temperature differences for heat transfer must be limited particularly in the low pressure (LP) preheaters. Higher temperature differences will seriously raise the exergy loss in these heaters because of the rather steep temperature curve of the feedwater in the low temperature region.

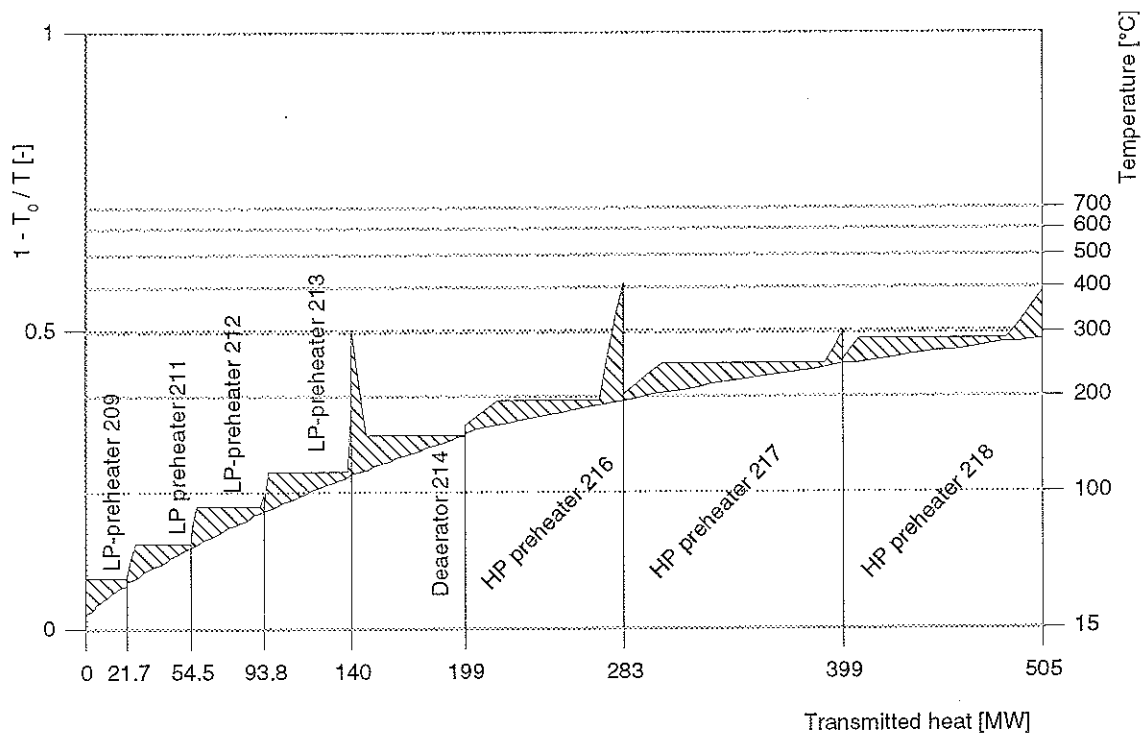


Figure 6.3 *Value diagram of the same feedwater train as in Fig. 6.2*

The heat transfer in a large steam boiler is presented as another example. The Q,T -diagram is shown in Fig. 6.4. The diagram shows the theoretical flue gas temperature from the temperature achieved during adiabatic combustion till ambient temperature. Due to simultaneous combustion and heat transfer in the boiler furnace, high flue temperatures over approximately 1500 °C do not occur in actual boilers. But the theoretical flue gas temperatures will provide the correct basis for evaluating exergy losses. The Q,T -diagram in Fig. 6.4 also shows the temperature curves of the heated fluids. The evaporator is located in the part with the hottest flue gas. After passing the successive superheaters and reheaters the flue gas is finally cooled in the economizer and air preheater. With exception of the economizer and the air preheater temperature differences between the primary fluid (the heated fluid) and the flue gas are very high, thus limiting the overall heat transfer area. A better insight into the exergy losses due to heat transfer in the boiler can be derived from the corresponding value diagram, as shown in Fig. 6.5. The value diagram confirms the high exergy losses in the high temperature part of the boiler, the furnace with evaporator and the first superheater. But it also appears that exergy losses in the low temperature heat exchangers are certainly not negligible.

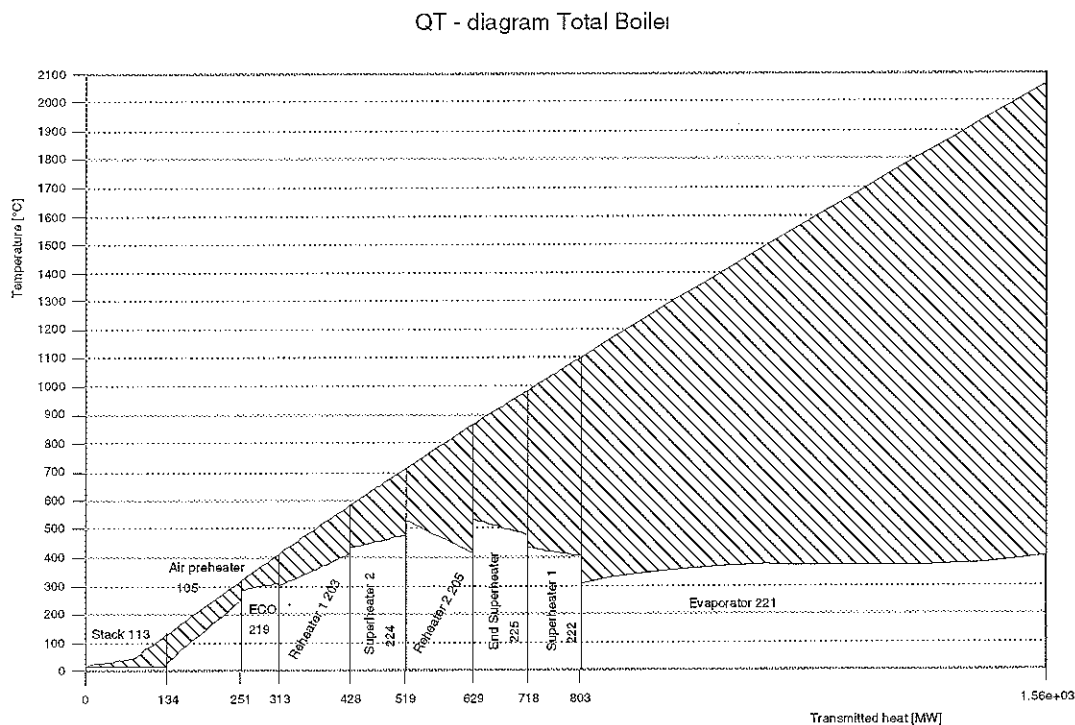


Figure 6.4 Q,T -diagram of heat transfer in a large steam boiler

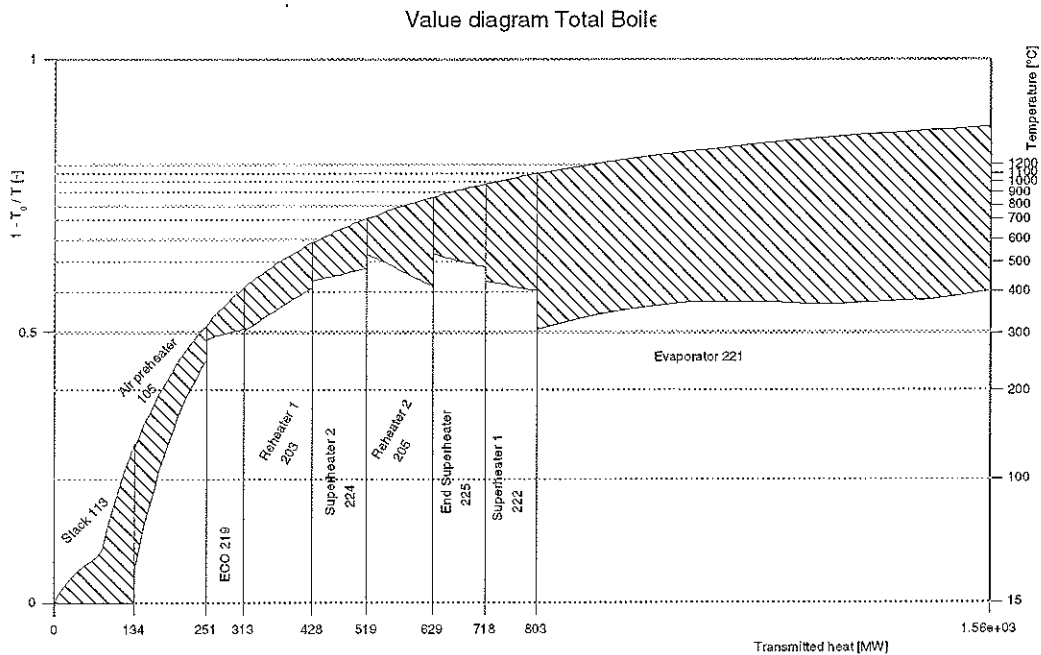


Figure 6.5 Value diagram of heat transfer in a large steam boiler

6.3 The value diagram for thermal power plants

The value diagram is primarily developed for the evaluation of heat transfer processes. But with some ingenuity it can also be used for the evaluation of thermal power plants: large power stations as well as combined heat and power plants. Starting point for the exergy analysis of power plants is the fuel exergy that is supplied to the plant. In general the fuel exergy does not equal the heating value of the fuel, however the differences between the exergy value and the lower or higher heating value of fossil fuels are limited. The difference between exergy and lower heating value can be established by the exergy factor of the fuel:

$$f_{ex,F} = \frac{ex_F}{LHV_F} \quad (6.5)$$

In Table 6.1 this ratio is roughly indicated for different types of fossil fuel. The ratios for coal and oil are estimated values, derived from [Baehr, 1979] and [Baehr, 1987], the ratio for natural gas is a value calculated for natural gas (Slochteren quality).

Table 6.1 Ratio between the exergy and lower heating values for different types of fossil fuel

fuel	$f_{ex,F}$
hard coal	1.02 - 1.03
fuel oil	0.98 - 1.01
natural gas	1.04

The width of the value diagram represents an amount of heat; this can be for instance the heating value (here the lower heating value will be applied) of 1 kg of fuel. Instead of the lower heating value, one can take the exergy of the fuel as the width of the diagram, as is done in Fig. 6.6. As the vertical scale of the diagram starts at $T = T_0$ and ends at $T = \infty$, the length of the vertical axis equals 1. The total area of the diagram then represents the exergy of the fuel. Such a diagram can be very useful for evaluating thermal plants.

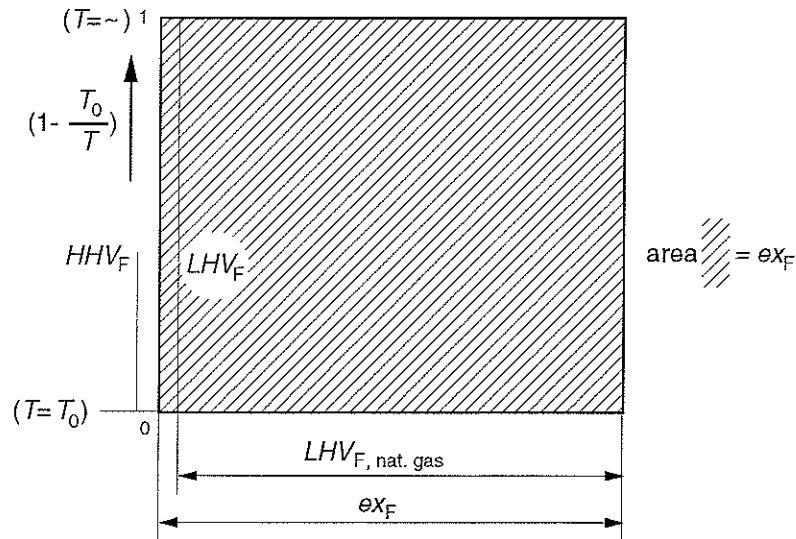


Figure 6.6 Value diagram based on the exergy of the fuel

After adiabatic combustion of a fuel, the flue gasses will have a high temperature. As fuel and oxidizer are supposed to enter the combustion chamber at ambient temperature, the heat that can be transferred from the flue gas by cooling down till ambient temperature will equal the lower heating if it is assumed that water vapour in the flue gas will not condense. In Fig. 6.7 the temperature curve of the flue gas during cooling down is represented, assuming that the exergy factor $f_{ex,F}$ is higher than 1. At the right border of the diagram the temperature of the gas is the adiabatic combustion temperature. The area below the temperature curve then equals the exergy of the heat that can be derived from the flue gas. After combustion only the exergy of the heat is available for use. This means that the exergy represented by the shaded area above the temperature curve together with the area at the left side of the diagram representing the difference between the exergy and the lower heating value has been lost. Therefore this area is considered to be the exergy loss due to combustion.

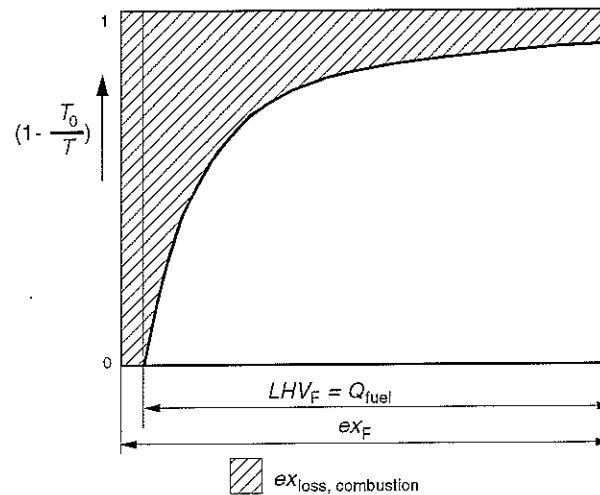


Figure 6.7 Value diagram with the temperature curve of flue gas during cooling from adiabatic combustion till ambient temperature

In a steam boiler heat from the flue gas is transferred to the steam cycle. The feedwater entering the boiler is heated till approximately saturation temperature, evaporated and superheated respectively. In the value diagram shown in Fig. 6.8 the temperature curve of the steam is also added. Flue gasses leave the boiler above ambient temperature and are supposed to be discharged at boiler outlet temperature via the stack to the atmosphere. Therefore the available heat in the flue gas is partially lost. The corresponding exergy loss in Fig. 6.8 is indicated as the exergy loss by the stack. The area below the temperature curve of water/steam represents the exergy transferred to the steam cycle. Then the area between the temperature curves of flue gas and water/steam represents the exergy loss due to heat transfer in the boiler.

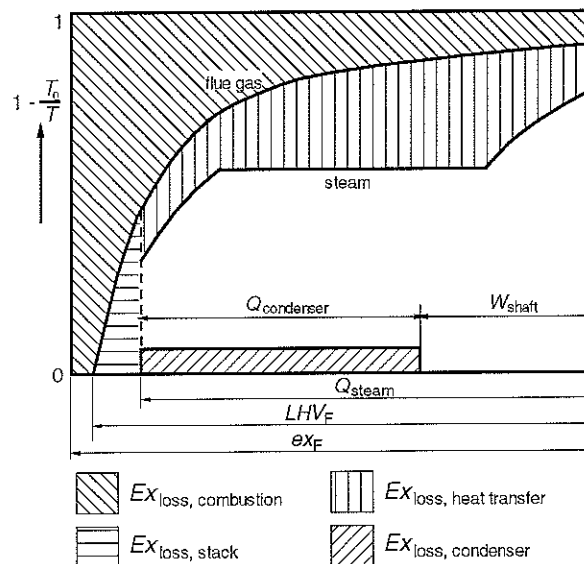


Figure 6.8 Value diagram of a conventional steam cycle power station

The generated steam by the boiler is expanded in a steam turbine; there the largest part of the exergy transferred to steam cycle is converted into mechanical energy (= 100 % exergy). After expansion the steam is condensed; in the condenser heat is transferred to

the environment. Water or air is generally used as cooling fluid. The difference between condensing temperature and ambient temperature is determined by the temperature increase of the cooling fluid in the condenser and the temperature difference between the primary and secondary fluid necessary to enable heat transfer. Consequently condenser temperature is 10 K or more, higher than the ambient temperature. The exergy loss resulting from this temperature difference is represented in Fig. 6.8 as the exergy loss of the condenser. The remaining (not shaded) area below the water/steam temperature curve represents (roughly) the generated mechanical (or electrical) work. It must be noted that Fig. 6.8 mainly shows losses occurring outside the steam cycle; including the internal exergy losses in the same value diagram can not easily be done.

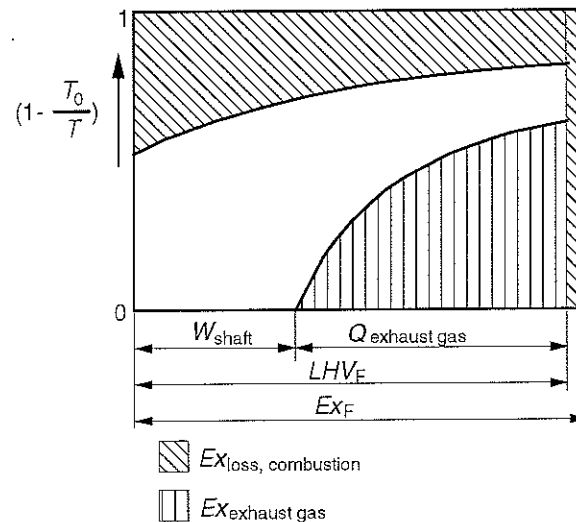


Figure 6.9 Value diagram of a gas turbine cycle

Today gas turbines are often used for electricity generation, in case of large-scale electricity production in combination with a bottoming steam cycle in so called combined cycle plants. A gas turbine cycle can also be represented in a value diagram as shown by Fig. 6.9: Just like the value diagram of the steam cycle, the exergy losses within the gas turbine cycle, like losses due to friction in compressor and turbine, are neglected in this diagram. In an open cycle gas turbine heat is transferred to the cycle in the combustion chamber. Air enters the combustion chamber at elevated temperature due to the previous compression; in the combustion chamber a further increase of temperature is obtained. The upper temperature curve in Fig. 6.9 represents the temperature increase in the combustion chamber; here heat is transferred to the gas turbine cycle. The area above this curve, together with the strip at the right side of the diagram due to the difference between the heating value and the exergy of the fuel, is the exergy loss due to combustion. The area below this curve represents the exergy that is transferred to the cycle.

After expansion in the turbine the temperature of the flue gasses is rather high (around 500 °C). The lower curve in Fig. 6.9 represents the temperature of the flue gas while cooling down till ambient temperature. The area below this curve represents the exergy that is still available in the hot exhaust gas of the turbine. When the flue gas is directly discharged to the environment, this exergy will be lost. In stationary applications however the residual heat of the turbine exhaust gasses is generally used to generate

steam in a heat recovery steam generator. In industrial applications mainly saturated steam is generated; in case of large scale power plants (combined cycle plants) superheated steam is generated for the bottoming steam turbine cycle. Figure 6.10 indicates which part of the flue gas exergy can be recovered under these circumstances.

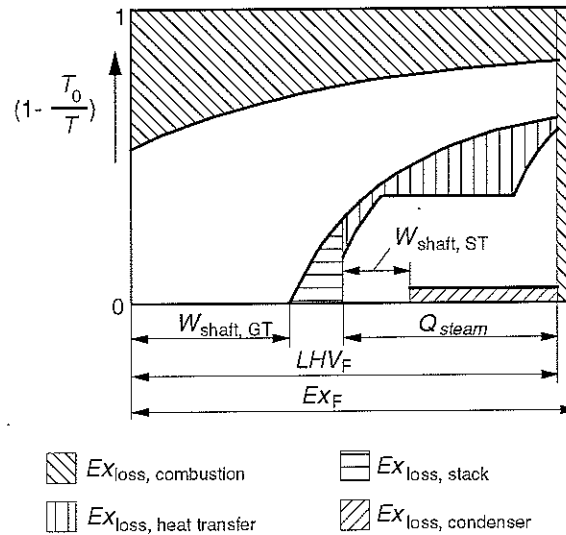


Figure 6.10 Value diagram of combined cycle plant

6.4 Exergy efficiencies

6.4.1 Introduction

Exergy efficiencies can be used for varying purposes. An obvious application is to use them for assessing, analyzing and optimizing processes and systems. We can think here of processes and systems for converting substances in chemical process plants, and of processes and systems for energy conversion. Below, we will specifically deal with their application in energy conversion systems.

Exergy efficiencies have only minor importance for a rough evaluation of electricity production units: the generated electricity equals the produced exergy, while usually the exergy of common fuels for electricity production differs only a few percentages from the LHV . For that reason, a power plant's exergy efficiency will differ only slightly from its thermal efficiency and will, therefore, not really provide additional information. The situation is completely different for combined heat and power (CHP) plants. The exergy efficiency there also visualizes the thermodynamic significance of the generated heat. Since the exergy of heat depends on the temperature, and is for finite temperatures always smaller than the energy quantity, exergy efficiencies of combined heat and power plants will usually be lower than comparable thermal efficiencies. Specifically the temperature level of the generated heat determines the difference. The exergy efficiency may provide additional information about the quality of the conversion in the CHP plant; the interpretation of exergy efficiencies, however, is still ambiguous.

Exergy efficiencies are particularly valuable in analyzing and optimizing systems. An exergy analysis usually includes a detailed calculation of the exergy values of process flows and the exergy losses in the system. Such a calculation shows the places in the system where losses occur and the extent of these losses. In the analysis the question has to be answered how the exergy losses can be avoided or limited. Based on the absolute value of exergy loss, it is usually difficult to assess whether an exergy loss in an apparatus is unnecessarily large. An exergy efficiency in which the exergy loss is compared with the added or transferred exergy gives a better picture of the quality of the processes in the apparatus, and thus also gives a better impression of whether exergy losses can be reduced. Exergy efficiencies of apparatuses (or parts of plants) can also be valuable in checking process calculations. Unrealistic input data or incorrect assumptions can result in unusual efficiencies. Obviously, only frequently applied apparatuses are eligible for such a checking.

6.4.2 General definition of exergy efficiency

Generally, efficiencies (and thus also exergy efficiencies) for practical use must meet a number of conditions.

1. The sensitivity for changes in the system involved must be large. Efficiencies must be defined in such a way that all values between 0 and 1 are possible, and no other values.
2. Preferably, the definition of efficiency must be applicable in practice. This means that the definition, without additions, must be practicable to a large number of different systems.
3. It must be possible to calculate efficiency values quickly, using available data. Preferably, one should avoid the necessity of making very detailed additional calculations.
4. Efficiencies are a measure for a system's quality. Such a standard is only reliable if it is based on data that take into account the influence of all relevant parameters. The quality of the process calculation performed determines whether this condition is satisfied.

As stated above, an efficiency definition not only concerns a theoretically sound choice, but has to consider also the calculation of exergy values needed for these efficiencies. The preference for a specific efficiency definition is thus also determined by the way in which exergy values are available.

Several authors have provided definitions for exergy efficiencies, e.g. [Kotas, T.J.], [Tsatsaronis, G.], [Brodyansky, V.M., Sorin, M.V., Le Goff, P.]. From an evaluation of published definitions it appears not to be possible to satisfy all mentioned conditions simultaneously. Depending on the significance of the various conditions, it is possible to define exergy efficiencies in various ways. Figure 6.11 distinguishes two different definitions of efficiencies, i.e. *universal efficiency* and *functional efficiency*¹⁾. The universal efficiency is based on a generally workable definition for exergy efficiency.

¹⁾ The names as applied here for the exergy efficiencies are not derived from literature. In the literature stated, a name for the universal efficiency was not found; various names, however, are used for functional efficiency: rational efficiency, efficiency with transiting exergy, etc.

Due to the relative insensitivity to changes in the system, it is rejected as insufficient in the publications stated. The functional efficiency is preferred, but requires further specification, depending on the type of system. For certain systems, relevant specifications are difficult or completely inconceivable. Both definitions of efficiency are introduced here in order to emphasize the significance of functional efficiency and to show the impossibility of specifying functional efficiency for certain systems.

The *universal efficiency* is defined as follows:

$$\eta_{Ex,u} = \frac{\sum Ex_{out}}{\sum Ex_{in}} \quad (6.6)$$

In which:

$\sum Ex_{out}$ is the exergy of the energy flows leaving the system
 $\sum Ex_{in}$ is the exergy of energy flows entering the system

The difference in exergy between the ingoing and outgoing energy flows equals the exergy loss, i.e.

$$\sum Ex_{in} = \sum Ex_{out} + \sum Ex_{loss} \quad (6.7)$$

For (6.6), we can also write:

$$\eta_{ex,u} = \frac{\sum Ex_{in} - \sum Ex_{loss}}{\sum Ex_{in}} \quad (6.8)$$

The universal efficiency offers a clear definition for a variety of systems. A disadvantage of this definition, however, is that the efficiency values obtained can be insensitive to changes in the system. This occurs, for example, when only part of the flows undergo a change or when the flows undergo only minor changes. The exergy loss is then small compared to the exergy of the ingoing energy flows. In that the exergy flows contain large “ballast flows”: exergy flows that are actually fed to the process, but not directly involved in the intended conversion. As a result of these ballast flows, the universal efficiency may be insensitive to changes in exergy loss.

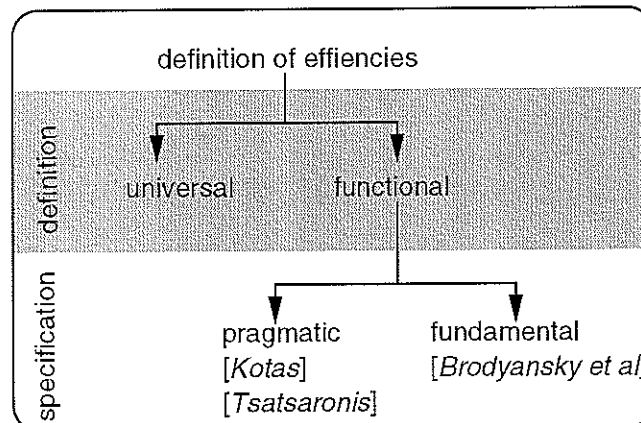


Figure 6.11 Possible division of exergy efficiencies

This is the reason for defining functional efficiency, in addition to universal efficiency. With functional efficiency, the influence of ballast flows is eliminated as much as possible in order to achieve the highest possible sensitivity to changes in the system. A general definition of *functional efficiency* is:

$$\eta_{ex,f} = \frac{\sum Ex_{product}}{\sum Ex_{source}} \quad (6.9)$$

In which:

$\sum Ex_{product}$ is the exergy of that part of the outgoing energy flows that can be considered to be the product of the system;

$\sum Ex_{source}$ is the exergy of that part of the ingoing energy flows that can be considered necessary for making the product in the present process.

Basically, $\sum Ex_{source}$ must be identical to the supplied exergy minus the exergy of the ballast flows, or:

$$\sum Ex_{source} = \sum Ex_{in} - \sum Ex_{ballast} \quad (6.10)$$

Similarly, it can be written for $\sum Ex_{product}$:

$$\sum Ex_{product} = \sum Ex_{out} - \sum Ex_{ballast} \quad (6.11)$$

Since by definition, the exergy of ingoing and outgoing ballast flows is the same, the difference in exergy value between "source" and "product" must also be identical to the sum of the exergy losses in the system, i.e.:

$$\sum Ex_{source} = \sum Ex_{product} + \sum Ex_{loss} \quad (6.12)$$

The Eqs. (6.9) through (6.12) are insufficiently clear about how to calculate the functional efficiency of a specific system. First, it must be established which flows (or sub-flows) are part of $Ex_{product}$, Ex_{source} or $Ex_{ballast}$. In addition to the definition given, a more detailed elaboration/specification (see also Fig. 6.11) of the functional efficiency is needed. It appears, however, that it is not possible to provide a generally valid specification of $Ex_{product}$, Ex_{source} or $Ex_{ballast}$. Thus the exergy of product and source has to be specified for each individual system.

The various authors do not use uniform methods and names to specify functional efficiencies. The fundamental approach, as supported by [Brodyansky], is preferred, but produces large practical problems since it requires a drastic breakdown of exergy values. For this reason, a more pragmatic approach has been chosen that is very similar to the proposals made by [Kotas] and [Tsatsaronis]. Therefore the breakdown of exergy values of flows of matter can be limited to the breakdown into the thermo-mechanical exergy (Ex_{tm}) and the chemical exergy (Ex_{ch}).

In the appendix the functional efficiency of a large number of apparatuses has been specified. Considering the limited breakdown of exergy values required here, this specification is generally workable. Exergy values – and thus exergy efficiencies – depend on the environment definition chosen. Strictly speaking, mutual comparison of exergy efficiencies is only possible if the same environment definition is assumed.

6.4.3 Explanation of efficiency definitions

The difference between universal efficiency and functional efficiency can be illustrated with a simple example: the heat transfer process in a heat exchanger.

The purpose of a heat exchanger is to heat a process flow – called primary flow – by withdrawing heat from another process flow – called secondary flow. The exergy of the primary flow will increase as a result of the absorbed heat, and the exergy of the secondary flow will decrease. Figure 6.12 visualizes the exergy change in the process

flows. The universal exergy efficiency for this heat exchanger can be obtained from Eq. (6.6):

$$\eta_{ex,u(\text{heat exchanger})} = \frac{Ex_{s,out} + Ex_{p,out}}{Ex_{s,in} + Ex_{p,in}} \quad (6.13)$$

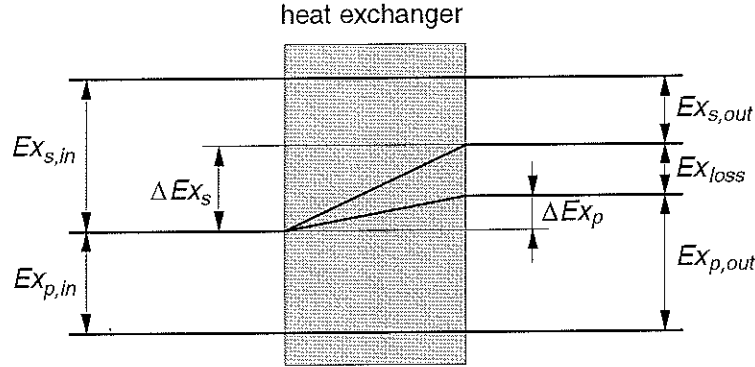


Figure 6.12 Change in exergy quantities during heat transfer

Considering the exergy quantities involved in the process, the exergy quantities $Ex_{p,in}$ and $Ex_{s,out}$ flow through the process without undergoing any change. These flows can be regarded as ballast flows that are not part of the process and thus need not be considered in the process assessment. If possible, the specification of a functional efficiency should consider only changes in exergy quantities. It is assumed that the purpose of the heat exchanger is to heat the primary flow. The exergy change ΔEx_p of the primary flow can be regarded as the desired product of the process, $Ex_{product}$ from Eq. (6.9). The secondary flow provides the necessary exergy to obtain this product. The exergy change ΔEx_s of the secondary flow can thus be regarded as the exergy source, Ex_{source} . Then, for the functional exergy efficiency of the heat transfer process can be written:

$$\eta_{ex,f(\text{heat exchanger})} = \frac{\Delta Ex_p}{\Delta Ex_s} = \frac{Ex_{p,out} - Ex_{p,in}}{Ex_{s,in} - Ex_{s,out}} \quad (6.14)$$

In this equation, the exergy flows seen as ballast are not taken into consideration. Actually, in comparison (6.13) the exergy loss is related to the total exergy supplied ($Ex_{s,in} + Ex_{p,in}$), while in Eq. (6.14) the exergy loss is related to the exergy change in the secondary flow (ΔEx_s). Since ΔEx_s is always smaller than ($Ex_{s,in} + Ex_{p,in}$), the functional efficiency (6.14) is more sensitive to changes in exergy loss than the universal efficiency (6.13).

6.5 References

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7 Fluid flow and heat transfer exergy loss

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7.1 Introduction

Separate or combined effects of fluid flow and heat transfer are encountered in many technical installations and with many chemical processes. Fluid flow originates from work or potential energy that is converted into the kinetic energy of flowing mass. Such energy conversions are always accompanied by losses. Part of the added energy is converted into heat by internal friction of the fluid. In thermodynamics this is expressed as: Irreversible processes cause production of entropy and loss of exergy. For equal mass flow the exergy loss in pipe flow can be diminished by increasing the pipe diameter. In that way the driving force needed to keep the mass flowing is reduced.

Pipes conducting fluid having a temperature different from surrounding temperature can exchange heat with the surroundings by conduction, convection and radiation. Such a heat transfer over a finite temperature interval always means exergy loss. This loss can be reduced with a smaller heat exchanging area, thus smaller diameter of the pipe. So, when fluid flow and heat transfer act together both phenomena have to be taken into account in a joint treatment for an assessment and optimization of exergy loss. Transport phenomena like fluid flow and heat transfer are described in many standard textbooks, e.g. by Van den Akker and Mudde [1], Beek and Muttzall [2], Bird, Stewart and Lightfoot [3], Coulson and Richardson [4] and Kreith and Bohn [5]. With the basic knowledge from these books and the laws of thermodynamics the characteristic phenomena can be described and exergy loss rated. A comprehensive overview concerning the production of entropy and its minimization in technical processes is given by Bejan [6,7]. In the following only Newtonian fluids will be considered. This means fluids for which viscosity η [kg/m s] is single valued, that is not depending on flow conditions or flow history. Further only hydrodynamically and thermally fully developed flows will be considered. These are flows for which the velocity and temperature profiles are independent of axial position and time.

In many thermodynamic installations and processes the wanted production of e.g. heat, cold, electricity, liquid helium, etc. is directly proportional to mass flow. Moreover the mass flow is a constant in the whole or large parts of the plant or installation. As a consequence, in this contribution, the mass flow will be used often instead of mean velocity or volume flow.

7.2 Pipe flow

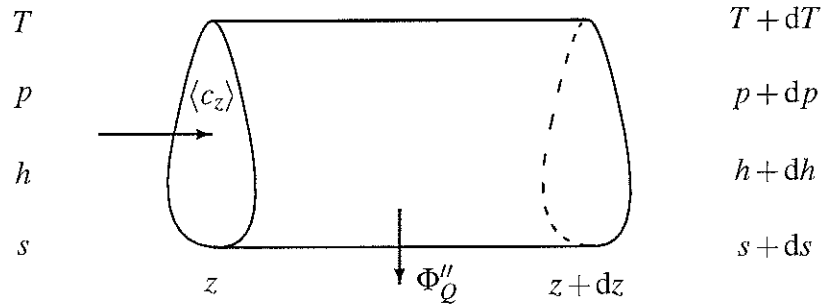


Figure 7.1 Length element dz of pipe with state properties temperature T , pressure p , enthalpy h , entropy s , mean velocity $\langle c_z \rangle$ and heat flux Φ_Q''

In pipe flow often the notion hydraulic diameter is encountered. If the cross section is non circular the hydraulic diameter D_h equals four times the ratio of cross section of the fluid A and the wet perimeter F

$$D_h = \frac{4A}{F} \quad (7.1)$$

For a pipe with circular cross section the hydraulic diameter is equal to the diameter. The relation between mean axial fluid velocity $\langle c_z \rangle$, mass density ρ , mass flow rate Φ_m and volume flow rate Φ_V is given by:

$$\Phi_m = \rho \Phi_V = \rho A \langle c_z \rangle \quad (7.2)$$

The pressure drop over a small length element of the pipe dz , see Fig. 7.2, is given by the Fanning equation:

$$-\frac{dp}{dz} = \frac{4f \frac{1}{2} \rho \langle c_z \rangle^2}{D_h} = \frac{4f \Phi_m^2}{2D_h \rho A^2} \quad (7.3)$$

f is called the friction factor. The so-called friction factor f is a function of the Reynolds number

$$Re = \frac{\rho \langle c_z \rangle D_h}{\eta} = \frac{4 \Phi_m}{\eta F} \quad (7.4)$$

and the roughness ϵ of the wall of the tube. The dissipation per unit of length is given by:

$$\frac{dP}{dz} = - \frac{dp}{dz} \Phi_V \quad (7.5)$$

The necessary pumping power can be found by integrating the dissipation per unit of length. The dissipated mechanical energy per kg of mass flowing is given by:

$$dw_t = - \frac{1}{\rho} \frac{dp}{dz} dz \quad (7.6)$$

For a pipe system that is kept on surrounding temperature, the dissipated mechanical energy is equal to the exergy loss. Mechanical work has been transformed into heat at surrounding temperature, which can not be further used. For the determination of the exergy loss, information is needed about the friction factor. This is the subject of the following part.

7.2.1 Laminar pipe flow

For laminar flow, also known as Poiseuille flow, in a horizontal pipe with circular cross section it is found:

$$4f = \frac{64}{Re}, \quad 0 < Re < 2000 \quad (7.7)$$

If the cross section is non circular, the friction factor is still proportional to $1/Re$, but with a different coefficient of proportionality. For laminar flow between two infinite parallel plates it is found $4f=96/Re$. Bird et al. [3] present an analytical solution for the flow in between two concentric cylinders, the so-called annulus. Then for the friction factor a value is found in between the values for flow in a pipe with circular cross section and for flow between parallel plates. Other examples are given by Kreith en Bohn [5].

Laminar flow dissipation of vapour and liquid in pipe with circular cross section

Condensation and evaporation processes are found in electrical power stations, refrigerators and in liquefaction. Here the example is chosen of liquid water flowing to an evaporator at 100 °C and the steam that is formed is ducted away. Both flows are laminar and are in horizontal pipes with circular cross section. In other examples, the same situation will be considered under different flow conditions. What is the ratio between the dissipation when both tubes having the same diameter and how should the ratio between diameters be chosen to have equal dissipation per unit pipe length?

If the Fanning Eq. (7.3) together with Eq. (7.7) is substituted into the expression for dissipation per unit mass flow (7.6) it is found:

$$dw_t = \frac{128}{\pi} \frac{\eta \Phi_m}{\rho^2 D^4} dz \quad (7.8)$$

Table 7.1 *Some properties of water and steam at 100 °C and 101325 Pa.*

		Water	Steam	
Mass density	ρ	958,38	0,5977	kg/m ³
Viscosity	η	$0,2818 \cdot 10^{-3}$	$0,1255 \cdot 10^{-4}$	kg/m s
Thermal conductivity	λ	0,682	0.0249	W/m K
Specific heat capacity	c_p	4211.	2034.	J/kg K
Prandtl number	Pr	1.74	1.02	-

With equal diameter and equal mass flow, the ratio of dissipation of vapour 1 (steam at 100 °C and standard pressure) and liquid 2 (water at 100 °C and standard pressure) according to Eq. (7.8) is given by:

$$\frac{\eta_1 \rho_2^2}{\eta_2 \rho_1^2} = \frac{0.01255}{0.2818} \left(\frac{958.38}{0.5977} \right)^2 = 1.145 \cdot 10^5.$$

For equal dissipation the fourth root from this value has to be taken to find: $D_1/D_2 = 18.4$. For equal diameter the ratio between Reynolds numbers: $Re_1/Re_2 = \eta_2/\eta_1 = 22.5$. For the equal dissipation case it is found: $Re_1/Re_2 = \eta_2 D_2/\eta_1 D_1 = 1.22$. These ratios are well within bounds, so that both flows indeed can be laminar.

7.2.2 Turbulent pipe flow

For a model of turbulent flow in a smooth pipe the equation of Blasius gives the friction factor as function of Reynolds number:

$$4f = 0.316 Re^{-1/4}, \quad 2500 < Re < 10^5 \quad (7.9)$$

This equation has been expressed with a dashed line in Fig. 7.2. Also other empirical relations are known for this case. Be aware of the range of validity that always should be given with such equations.

The relation of the friction factor with the Reynolds number and roughness of the pipe surface has been expressed in the solid curves given in Fig. 7.2. Such a figure is often called Moody diagram. These curves have been calculated with the equation given by Coulson and Richardson [4]:

$$\begin{aligned} (f/2)^{-0.5} &= 2.5 \ln \left(0.27 \varepsilon + 0.885 Re^{-1} (f/2)^{-0.5} \right), \\ 2500 &< Re < 10^7 \end{aligned} \quad (7.10)$$

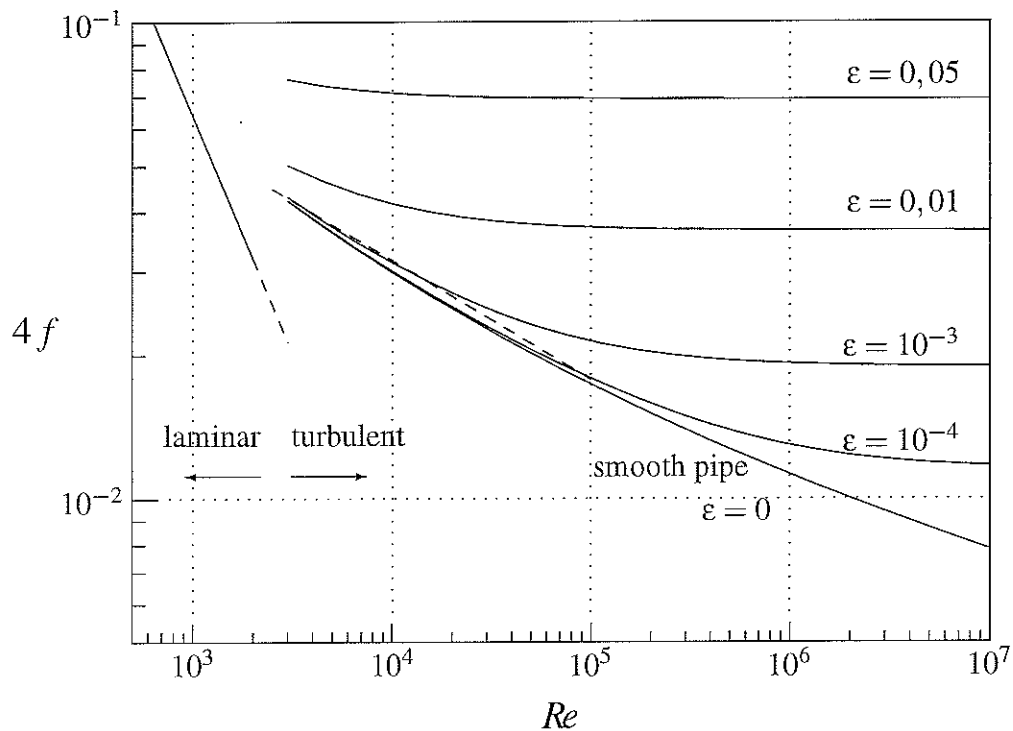


Figure 7.2 Friction factor $4f$ as function of Reynolds number Re and relative roughness ε of inner pipe surface.

Here ε is the relative roughness, the ratio of mean height of the roughness and pipe diameter. If relative roughness becomes large the friction factor is almost independent of Re . It also can be noticed that the results for smooth pipes obtained with this equation do not coincide with the Blasius equation. Differences up to 10 % can be seen in the value of friction factor. This gives an impression of the accuracy that can be reached with this kind of equations. For completeness, left in Fig. 7.2 the friction factor for laminar flow in a round tube as known from Eq. (7.7) is presented.

Turbulent flow dissipation in smooth pipe

Suppose now that the friction factor is given by the Blasius equation and repeat then the example before. Now dissipation is given by:

$$dw_t = \frac{1.7876}{\pi^{1.75}} \frac{\eta^{0.25} \Phi_m^{1.75}}{\rho^2 D^{4.75}} dz \quad (7.11)$$

A 10 % larger diameter decreases friction losses with more than 50 %. The ratio between dissipations in case of equal diameter and equal mass flow is given by:

$$\left(\frac{\eta_1}{\eta_2}\right)^{0.25} \frac{\rho_2^2}{\rho_1^2} = \left(\frac{0.01255}{0.2818}\right)^{0.25} \left(\frac{958.38}{0.5977}\right)^2 = 1.181 \cdot 10^6.$$

For the case of equal dissipation it is found for the ratio of diameters: $D_1/D_2 = (1.181 \times 10^6)^{1/4.75} = 19.0$.

Turbulent flow dissipation in very rough pipe with constant friction factor f_0

Again the same questions are asked as in the preceding examples. The dissipation caused by internal friction now becomes:

$$dw_t = 4f_0 \frac{8}{\pi^2} \frac{\Phi_m^2}{\rho^2 D^5} dz.$$

For the ratio of dissipations with equal diameter it is found:

$$\frac{\rho_2^2}{\rho_1^2} = \left(\frac{958.38}{0.5977} \right)^2 = 2.571 \cdot 10^6$$

and the ratio of diameters for equal dissipation is given by: $D_1/D_2 = (2.571 \times 10^6)^{1/5} = 19.1$. It is remarkable how little the ratios of diameter at equal dissipation differ in the three examples given.

7.2.3 Other systems

Examples of other systems that need a different treatment from the flow in long pipes are: Long and narrow bends in pipes, diffuser, venturi, throttling valve, tap, metering flange, jump in cross section, porous medium, arrangement of catalysts, packed and fluidized beds, shell and tube heat exchanger, etc.. The flow in this kind of pipe fittings or apparatus will not be fully developed. Yet it can be treated in a similar way as pipe flow, after introduction of a friction loss factor or friction coefficient that is a measure for flow resistance. Friction loss factor K_f is directly related to the dissipation per unit mass flowing:

$$w_f = K_f \frac{1}{2} \langle c_z \rangle^2 \quad (7.12)$$

Velocity c_z here is taken downstream of the apparatus. In the literature earlier mentioned [1,2,3] values for the resistance number are given. If in certain cases values are missing, then measurement of pressure drop and volume flow can give the answer. For flow in a pipe with length L the friction loss factor is given by: $K_f = 4fL/D_h$.

7.3 Heat transfer

When in thermodynamics two thermally interacting systems at different temperatures exist, heat transfer will occur until temperatures become equal. The temperature difference can be seen as driving force for the heat flow. In general, heat transfer processes are irreversible. Only in the limiting cases when losses approaches zero a reversible process is possible. This can be reached by an adiabatic process or infinitely well insulated process. Then heat flow becomes zero. The other limit is an isothermal process. Then the heat remains at the same temperature and no possibility to generate work is lost. Any deviation from the ideal isothermal or ideal adiabatic process means irreversibilities, entropy production and exergy loss.

First two infinitely large heat reservoirs at temperatures T_a respectively T_b are considered. It is supposed that $T_a > T_b$ and an amount heat dQ is transferred from reservoir a to reservoir b. It is supposed that no heat is lost in this process or that interaction with the surroundings does not exist. This is an irreversible process with production of entropy. This entropy production is positive and equals:

$$dS_{\text{irrev}} = dQ \left(\frac{1}{T_b} - \frac{1}{T_a} \right) \quad (7.13)$$

Then exergy loss is simply found with the Gouy-Stodola theorem:

$$dEx = T_0 dS_{\text{irrev}} = dQ T_0 \left(\frac{1}{T_b} - \frac{1}{T_a} \right) = dQ T_0 \left(\frac{T_a - T_b}{T_a T_b} \right) \quad (7.14)$$

With this equation it can be seen directly that exergy loss becomes nil in an adiabatic process ($dQ = 0$) and with an isothermal process ($T_a = T_b$). At temperature levels far above surrounding temperature the exergy loss by heat transfer is relatively small. At temperature levels far below surrounding temperature the exergy loss can be considerable. In the production of liquefied air or more pronounced with the production of liquid helium this becomes important. When in helium production the surrounding temperature $T_0 = 300\text{K}$, and further somewhere in a heat exchanger $T_a = 12\text{K}$ en $T_b = 4\text{K}$ the exergy loss amounts to $50dQ$. To prevent this type of exergy loss it is important to keep the temperature difference $T_a - T_b$ as small as possible. In a heat exchanger this can be approached with an as large as possible heat exchanging area. This has technical, e.g. heat exchange with surroundings, as well as economical limits.

A principle way to prevent the exergy loss is to couple heat exchange with production of work as indicated in Fig. 7.3.

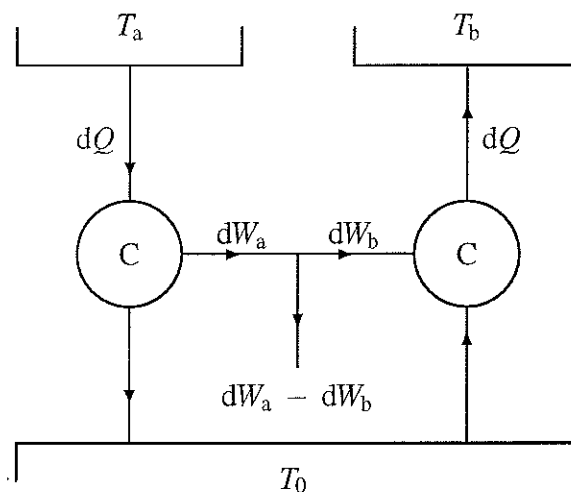


Figure 7.3

Reversible heat exchange between reservoirs a and b with help of a reversible engine and a reversible heat pump e.g. Carnot processes.

A reversible engine (Carnot engine) receiving an amount of heat dQ at temperature T_a and rejecting heat at surrounding temperature produces an amount work:

$$dW_a = dQ \left(1 - \frac{T_0}{T_a} \right) \quad (7.15)$$

A reversible heat pump (Carnot heat pump) takes heat from the surroundings and delivers an amount of heat dQ at temperature T_b . The heat pump asks for a work input

$$dW_b = dQ \left(1 - \frac{T_0}{T_b} \right) \quad (7.16)$$

The net work to be gained then equals:

$$dW_a - dW_b = dQ T_0 \left(\frac{1}{T_b} - \frac{1}{T_a} \right) \quad (7.17)$$

As expected, this is equal to the exergy loss calculated in Eq. 7.14.

Maintaining of a constant temperature

A dwelling has to be kept internally at 20°C, while outside temperature equals 0°C. If heating occurs with electrical resistance heating in total 5 kW electrical energy is needed. However, when this loss is compensated with an ideally operating Carnot heat pump between internal and external temperature, power is needed as input for the heat pump:

$$5 \cdot 10^3 \left(1 - \frac{273.15}{293.15} \right) = 341 \text{ W}.$$

This is the minimal rate of work needed to maintain the internal temperature in the given situation.

Now suppose it is tried to reach this situation with help of an electrically driven Rankine cycle operating as heat pump. This is the same cycle as often applied in the household refrigerator. Because of irreversibilities this practical heat pump needs 1.5 kW to maintain the internal temperature. It is supposed that the electrical power station has an efficiency of 40 % based on natural gas heat content. The remaining question now is if space heating with such an electrically driven heat pump is favorable in the use of primary energy in comparison to a high efficiency burner with 90 % efficiency on natural gas heat content.

The high efficiency burner needs as natural gas power input $5.0/0.9 = 5.556$ kW, whereas the practical heat pump needs as natural gas power input $1.5/0.4 = 3.75$ kW. With the properties given here, the electrically driven heat pump is energetic and exergetic better than the high efficiency burner. In practice also aspects like investment and maintenance costs, price policy of suppliers and tax policy of government will play

their role and can lead to different conclusions. Anyhow we can see that both practical cases are still far away from the theoretical minimum power and that a favorable thermodynamic solution of the heating problem probably will look quite different. A possible solution can be found in the combined production of heat and power. This can be at large scale when district heating is offered by the electrical power station or on small scale at the end user. For this last option fuel cells are possible, but still many technical and economical problems have to be overcome. Here comparison was simple because in the two cases the same primary fuel was used. In case of different fuels a comparison on the exergy of the fuel is necessary.

Pinch analysis

From the preceding example it has become clear that to fulfil a given demand on heat at a prescribed temperature level in practice not only the exergy related to this demand has to be supplied, but that due to irreversibilities in combustion, energy conversions, heat transfer, flow, etc., a lot of exergy is still lost. Much can be saved with better mutual tuning of the supply and demand of heat and cold within an apparatus or industrial plant. Pinch analysis is a development tool to minimize the demand on heat and cold within a system by the application of e.g. heat exchangers. Compared to the analysis based on exergy, the pinch analysis has the drawback that it is, in its basic form, restricted to heat transfer. Then a number of possible savings, originating from the second law of thermodynamics, is neglected, because as driving force only temperature differences are considered.

7.4 Combined effect of flow and heat transfer

7.4.1 Pipe flow with heat transfer to the surroundings

Consider a small length element of a pipe dz that exchanges heat with the surroundings via a layer of insulation material and through which a fluid flows. This small part of a pipe, see Fig. 7.1, acts as a thermodynamic open system. The total exergy loss consists of contributions from internal mechanical dissipation and heat transfer to the surroundings.

Heat transfer

With the heat transfer via the insulation layer, per unit mass flowing, an amount of heat dq is transferred from temperature level T to surrounding temperature T_0 without production of any useful work. Consequently this means per unit mass flowing, an exergy loss equal to:

$$dex_1 = dq \left(1 - \frac{T_0}{T} \right) \quad (7.18)$$

This expression is also the minimal work needed to compensate for the heat loss to the surroundings. When the temperature of the fluid is lower than surrounding temperature both the sign of dq and $1-T_0/T$ change and loss remains loss.

Mechanical dissipation

The dissipated mechanical energy is converted into heat at temperature level T . If this temperature is higher than surrounding temperature in principle some work can be gained back with a reversible process. This leads to a net loss of exergy:

$$dex_2 = dw_t - dw_t \left(1 - \frac{T_0}{T}\right) = dw_t \frac{T_0}{T} \quad (7.19)$$

This expression, the product of mechanical dissipation with a loss factor T_0/T , is also in use for the description of friction losses in turbines and compressors. From this expression it also can be seen that when $T < T_0$, extra work is needed to compensate for the dissipated mechanical energy.

Total exergy loss

The total exergy loss of the open system per unit mass flowing now becomes:

$$dex = dex_1 + dex_2 \quad (7.20)$$

Both the dissipated mechanical energy and the exchange of heat with the surrounding still depend on flow characteristics. For the case of turbulent flow in a smooth pipe and the Blasius equation being valid the mechanical dissipation can be found in Eq. 7.11. For the heat transfer with the surroundings of the length element of the pipe the general expression is:

$$dq = \frac{\Phi'' \pi D dz}{\Phi_m} = \frac{(T - T_0) \pi D dz}{\Phi_m \left(\frac{1}{\alpha} + \frac{1}{\alpha_{iso}}\right)} \quad (7.21)$$

Here α is the heat transfer coefficient within the fluid and α_{iso} the heat transfer coefficient of the wall of the pipe and the layer of insulation material. For simplicity it is supposed that the pipe is that well insulated that the heat transfer process in the fluid can be neglected. Further it is assumed the heat transfer coefficient α_{iso} is independent of the diameter of the pipe. Then the total loss of exergy of the length element of pipe can be written:

$$dex = \frac{1.7876}{\pi^{1.75}} \frac{\eta^{0.25} \Phi_m^{1.75} T_0}{\rho^2 D^{4.75} T} dz + \frac{\alpha_{iso} (T - T_0)^2 \pi D}{\Phi_m T} dz \quad (7.22)$$

From this equation it is clear that the exergy loss from heat transfer (2nd term) increases with the diameter and the exergy loss from friction (1st term) at the same time decreases

In a situation with complete freedom, the diameter can be chosen such that the contribution of friction is negligible and additionally the insulation thickness is adapted to the desired level. When the level of insulation is fixed, e.g. because of costs, there exists an optimal diameter D_0 for which the exergy loss is minimal. This situation can be found by differentiation.

$$D_0^{5.75} = 0.365 \frac{\eta^{0.25} \Phi_m^{2.75} T_0}{\rho^2 \alpha_{\text{iso}} (T - T_0)^2} \quad (7.23)$$

Of course this equation is only valid as long as the assumptions made still hold. However, the character of the solution given is general. The exergy loss from friction decreases with the diameter and the exergy loss from heat transfer increases with the diameter.

Ratio of diameters at minimum exergy loss

Again the evaporator at standard pressure is considered, where water with temperature 100 °C enters and steam at temperature 100 °C exits. Both flows are in smooth pipes and are turbulent. The pipes are both well insulated with the same α_{iso} . In this case the ratio between diameters is given by:

$$\frac{D_1}{D_2} = \left(\left(\frac{\eta_1}{\eta_2} \right)^{0.25} \frac{\rho_2^2}{\rho_1^2} \right)^{1/5.75} = 11.4 .$$

If it is further supposed that $\Phi_m=0.4$ kg/s, $T_0=298.15$ K and $\alpha_{\text{iso}}=1.0$ W/m² K, then it is found for the diameter of the water flowing pipe in case of minimal exergy loss: $D_2=0.013$ m.

7.4.2 Heat exchangers

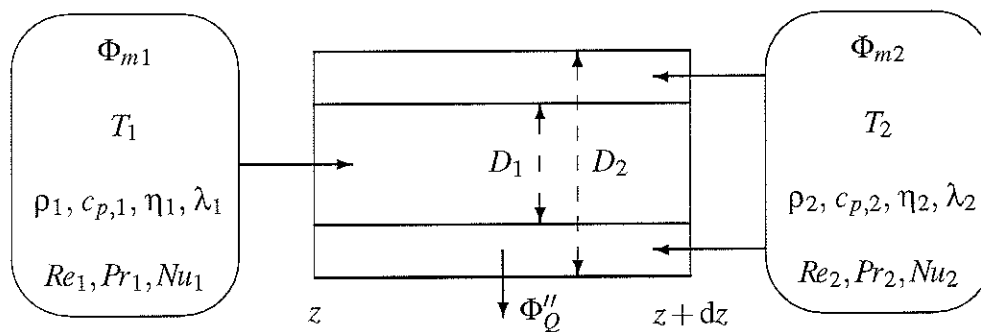


Figure 7.4 Cross section of an element dz of a simple heat exchanger consisting of concentric tubes.

In a heat exchanger the ultimate goal is the perfect heat transfer between two flows. Exergetically perfect means that the temperature differences between the two flows become negligible small. It can then be seen immediately that in this respect a co-current heat exchanger can never beat a counter-current heat exchanger. The exergy loss of a co-current heat exchanger is of the same order as with mixing of two flows with different temperature. With a counter-current heat exchanger, in case there is no heat exchange with the surroundings and no mechanical dissipation, the two flows can approach equal temperature and the exergy loss then becomes nil. This situation can only be reached when the two flows are thermally balanced, or:

$$\Phi_{m1} c_{p,1} = \Phi_{m2} c_{p,2} \quad (7.24)$$

For this ideal situation, the simple counter-current heat exchanger, that exists of two concentric round tubes and as depicted in Fig. 7.4 will be considered. The overall heat transfer coefficient between the two fluids is α_{tot} . The influence of the total length of the heat exchanger L on the exergy loss caused by heat transfer will be investigated. With help of a heat balance and the entering temperatures $T_1(0)$ and $T_2(L)$, it is found for the temperatures as function of axial distance z :

$$T_1(z) = T_1(0) - \frac{z}{L} (T_1(0) - T_2(L) - \Delta T) \quad (7.25)$$

$$T_2(z) = T_2(L) + \left(1 - \frac{z}{L}\right) (T_1(0) - T_2(L) - \Delta T) \quad (7.26)$$

The temperature difference between the two flows is constant.

$$\Delta T = \frac{T_1(0) - T_2(L)}{1 + \alpha_{\text{tot}} L \pi D_1 / \Phi_{m1} c_{p,1}} \quad (7.27)$$

Now two limiting cases can be found for which the exergy loss is nil. One limit is found for $L=0$. In this case there is no heat transfer and both flows keep their original temperature. As the purpose of a heat exchanger is the transfer of heat, this trivial limit will not be further considered. The other limit is for $L \rightarrow \infty$. In that case $\Delta T = 0$ and consequently the production of entropy is nil. Keep in mind that these calculations hold for the idealized counter-current heat exchanger only. Of course an infinite length is quite unpractical. With a finite length, ΔT still can be decreased with a large heat exchanging surface and a high overall heat transfer coefficient between the fluids. This kind of measures directly influences the exergy loss caused by friction or by heat exchange with the surroundings. If with ridges or other protrusions the heat exchanging area is augmented, also the roughness of the wall increases and with that the mechanical dissipation. Also when heat transfer is augmented by promoting turbulence, dissipation increases. Pressure drop, and thus dissipation, is proportional to length. With larger diameters heat transfer to the surroundings is promoted.

These counter acting aspects make, in a nutshell, the dilemma in heat exchanger design. Not yet to mention other restrictions as: limited allowed weight or volume, high pressures, safety regulations in nuclear reactors, fouling, aggressive or toxic liquids, etc..

7.4.3 Exergy loss in a heat exchanger

In a heat exchanger the two mass flows are not equal in general. That is why it is more convenient in the comparison of the different contributions to use the rate of exergy loss in stead of the specific exergy loss. The rate of exergy loss can simply be obtained from the earlier results for exergy loss per unit of mass flowing by multiplication with the appropriate mass flow rate.

Exergy loss by heat exchange with the surroundings

This subject already has been treated in section 7.4.1. The rate of exergy loss by this heat transfer is given by:

$$\dot{E}_{x0} = \frac{\alpha_{iso} (T_2 - T_0)^2 \pi D_2}{T_2} dz \quad (7.28)$$

Not all quantities are independent here, e.g. α_{iso} still can be a function of length L or diameter D_2 .

Exergy loss by dissipation in flow

For the exergy loss by mechanical dissipation in the flow, the same procedure can be followed as already applied in section 7.4.1. For the inner tube it can be followed completely. It is only necessary to multiply with Φ_{m1} to obtain the rate of exergy loss. In this way it is found:

$$\dot{E}_{x1} = \frac{1.7876}{\pi^{1.75}} \frac{\eta_1^{0.25} \Phi_{m1}^{2.75} T_0}{\rho_1^2 D_1^{4.75} T_1} dz \quad (7.29)$$

For the so-called annulus between diameters D_1 and D_2 first the friction factor has to be calculated with help of the Reynolds number based on the hydraulic diameter. For the annulus in Fig. 7.4 with inner diameter D_1 and outer diameter D_2 the hydraulic diameter equals $D_2 - D_1$. This yields Reynolds number:

$$Re_2 = \frac{4 \Phi_{m2}}{\eta_2 \pi (D_1 + D_2)} \quad (7.30)$$

Applying the same procedure as for Eq. 7.11 leads to:

$$\dot{E}_{x2} = \frac{1.7876}{\pi^{1.75}} \frac{\eta_2^{0.25} \Phi_{m2}^{2.75} T_0}{\rho_2^2 (D_2 + D_1)^{1.75} (D_2 - D_1)^3 T_2} dz \quad (7.31)$$

Exergy loss by heat transfer between the flows

For this loss for a length element of pipe dz it is found:

$$\dot{E}_{x3} = T_0 \alpha_{\text{tot}} (T_2(z) - T_1(z)) \pi D_1 \left(\frac{1}{T_1(z)} - \frac{1}{T_2(z)} \right) dz \quad (7.32)$$

Then still an expression for α_{tot} the overall heat transfer coefficient between the flows is needed. If the thermal resistance of the wall of the pipe is neglected,

$$\frac{1}{\alpha_{\text{tot}}} = \frac{1}{\alpha_1} + \frac{1}{\alpha_2} \quad (7.33)$$

is obtained, with α_1 respectively α_2 the heat transfer coefficients in both fluids. These coefficients can be found with the Dittus-Boelter-relation that is well-known from literature [5]. For heat transfer from a turbulent flow in a smooth pipe:

$$Nu = 0.023 Re^{0.8} Pr^{0.4}, \quad 2300 < Re < 10^7, \quad 0.5 < Pr < 120 \quad (7.34)$$

The Nusselt number $Nu = \alpha D_b / \lambda$, with λ the thermal conductivity of the medium and the Prandtl number Pr being the ratio between kinematic viscosity η/ρ and the thermal diffusivity $\lambda/\rho c_p$. So only material properties are needed together with a value for the Reynolds number. α_1 and α_2 now can be calculated with:

$$\alpha_1 = 0.023 \left(\frac{4}{\pi} \right)^{0.8} \frac{\lambda_1^{0.6} \Phi_{m1}^{0.8} c_{p,1}^{0.4}}{\eta_1^{0.4} D_1^{1.8}} \quad (7.35)$$

$$\alpha_2 = 0.0297 \frac{\lambda_2^{0.6} \Phi_{m2}^{0.8} c_{p,2}^{0.4}}{\eta_2^{0.4} (D_2 - D_1) (D_2 + D_1)^{0.8}} \quad (7.36)$$

Optimization

The total rate of exergy loss is found from an addition of the different contributions:

$$\dot{E}_{x_{\text{tot}}} = \dot{E}_{x0} + \dot{E}_{x1} + \dot{E}_{x2} + \dot{E}_{x3} \quad (7.37)$$

As this is for a length element dz the total loss of the heat exchanger can be found after integration. The parameters that can be independently fixed are heat exchanger length L and diameters D_1 and D_2 . In principle the optimum design can be found by partial differentiating the equation for the total exergy loss to the three parameters. Each partial derivative then is set zero, leading to three equations for the three parameters. This sounds rather straightforward but is not. Not only α_{tot} possibly depending on length L or diameter D_2 , but also the temperatures $T_1(z)$ en $T_2(z)$ are a function of the parameters.

So, it can be very handsome if computer software is available to carry out this kind of calculations. Then a suitable model of the heat exchanger is needed that can solve the

heat and momentum equations and with that can calculate exit properties from input properties. Then it is a simple step to evaluate the exergy loss. After that a strategy for optimization has to be followed. Optimization can be done on the level of the heat exchanger or on the level of the complete installation. This need not to give the same results, as exergy loss is also determined by the mutual interaction by the components of the complete installation.

For an existing heat exchanger the different contributions to the exergy loss can also be determined by measurements. Mechanical dissipation is found by measuring volume flow and pressure drop. Input and output temperatures give an impression of ΔT and thereby the heat transfer between the two flows. The measurement of the cooling down curve in a no-flow situation gives information on α_{iso} .

Without calculations or measurements it already can be concluded that for the heat exchanger discussed here, a situation will be possible with minimum exergy loss. The heat transfer to the surroundings limits length L and diameter D_2 . Very small length and diameter are not favourable, as then the temperature difference between the flows becomes too large. Somewhere in between the optimal situation can be found.

If an optimized heat exchanger is installed care should be taken that the situation of minimum exergy loss is maintained. Heat transfer can be decreased and mechanical dissipation increased by fouling. If the thermal insulation is damaged or aged, heat transfer to the surroundings will increase. With measurement of process parameters this kind of changes in operation can be detected. By cleaning and repairing these shortcomings the operation can be kept optimal.

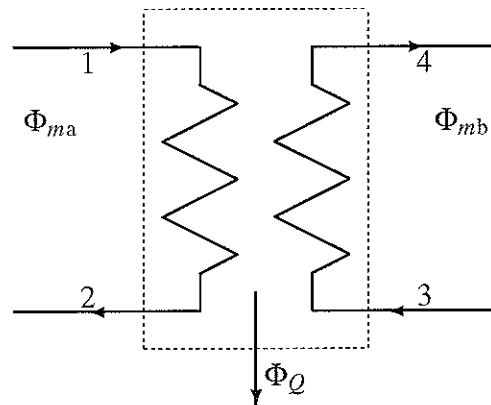


Figure 7.5 Heat exchanger

Complete heat exchanger

Now a complete heat exchanger is considered for which inflow and outflow conditions, e.g. enthalpy and entropy, are known. Both mass flow rates Φ_{ma} and Φ_{mb} are unequal. The heat flow rate to the surroundings equals Φ_0 . Pressure drops are Δp_a respectively Δp_b . Give the rate of exergy loss for the complete heat exchanger.

First notice that no useful work can be gained from the heat at surrounding temperature. The influence of pressure drop and all other irreversibilities is automatically accounted for in the state properties. So it is simply found:

$$\Delta \dot{E}x = \Phi_{ma} (h_2 - h_1 - T_0 (s_2 - s_1)) + \Phi_{mb} (h_4 - h_3 - T_0 (s_4 - s_3)) \quad (7.38)$$

This equation is in accordance with the general expression for exergy loss of an open system with boundary at surrounding temperature and can be used irrespective of which flow is heated or cooled.

7.5 References

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Symbols

Roman

	Description	Unit
A	Affinity	J/mol
A	Surface area	m ²
a	Activity	
α	Thermal diffusivity	m ² /s
c	Velocity	m/s
C_p	Heat capacity	J/K
c_p	Specific heat at constant pressure	J/kg K
$c_{p, \text{mol}}$	Molar heat at constant pressure	J/mol K
c_z	Axial velocity component	m/s
COP_{hp}	Coefficient of performance for a heat pump	
COP_{ref}	Coefficient of performance for a refrigerator	
D, D_h	(Hydraulic) diameter	m
Ex	Exergy	J
ex	Specific exergy	J/kg
ex_{mol}	Molar exergy	J/mol
Ex_{loss}	Exergy loss	J
F	Perimeter	m
f	Friction factor	
G	Gibbs-energy	J
g	Specific Gibbs energy	J/kg
g_{mol}	Molar Gibbs energy	J/mol
H	Enthalpy	J
h	Specific enthalpy	J/kg
h_{mol}	Molar enthalpy	J/mol
J	General flow	different
$l_{(x,y,z)}$	Length	m
m	Mass	kg
N	Number of particles	--
n	Amount of substance	mol
Nu	Nusselt number	
P	Power	W
p	Pressure	Pa
Pr	Prandtl number	
Q	Heat	J
q	Specific heat	J/kg
q_{mol}	Molar heat	J/mol
R	Gas constant	J/kg K
R_{mol}	Universal gas constant	J/mol K
Re	Reynolds number	
S	Entropy	J/K
s	Specific entropy	J/kg K
s_{mol}	Molar entropy	J/mol K

Symbols

T	Temperature, Celsius	°C
T	Thermodynamic temperature	K
T_0	Environmental temperature	K
U	Internal energy	J
u	Specific internal energy	J/kg
u_{mol}	Molar internal energy	J/mol
V	Voltage	V
V	Volume	m ³
v	Specific volume	m ³ /kg
v_{mol}	Molar volume	m ³ /mol
v	Reaction rate	mol/s
W	Work	J
w	Specific work	J/kg
X	Driving force	different
x	Quality	
x_i	Mass fraction	
$x_{\text{mol},i}$	Mole fraction	
z	Height	m
z	Axial cylinder coordinate	m

Greek

	Description	Unit
α	Heat transfer coefficient	W/m ² K
ε	Efficiency(second law)	
ε	Roughness	
η	Efficiency	
μ	Dynamic viscosity	kg/m s
λ	Thermal conductivity	W/mK
ν	Stoichiometric number	
ρ	(Mass) density	kg/m ³
Φ_q	Heat flow rate	W
Φ_q''	Heat flux	W/m ²
Φ_m	Mass flow rate	kg/s
Φ_v	Volume flow rate	m ³ /s

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Appendix A Functional exergy efficiencies for a variety of processes

A.1 Introduction

In general, functional exergy efficiency is defined as:

$$\eta_{ex,f} = \frac{\sum Ex_{product}}{\sum Ex_{source}} \quad (A.1)$$

In order to be able to apply this efficiency to specific systems, a further specification of $\sum Ex_{product}$ and $\sum Ex_{source}$ is required. The part of the exergy of the process flow that must be attributed to product or source has to be established for all ingoing and outgoing process flows. If only part of the exergy of a flow is attributed to a source or product, a breakdown is required into an active part (source or product) and a passive part (ballast). How to make such a breakdown depends on the desired accuracy with which exergy efficiencies have to be determined.

In practical situations, it is usually allowed to neglect the kinetic and potential energy of process flows. Also for the specification of the exergy efficiencies the kinetic and potential energy are assumed to be negligible. In a reversible process, kinetic and potential energies can be fully converted into power; if they are not negligible, they can be added to the relative exergy values.

Below, a limited breakdown of exergy values for flows is chosen into thermo-mechanical exergy and chemical exergy. *Thermo-mechanical exergy* is the power produced if the flow is brought in thermo-mechanical equilibrium with the environment in the prescribed way. The chemical composition remains unchanged in this process. *Chemical exergy* is the power generated if the flow is brought in chemical equilibrium with the environment in the prescribed way. Chemical equilibrium with the environment means that the components composing the flow are converted into environmental components and expand to the partial pressure of the respective component in the environment.

The sum of the two exergy terms equals the exergy of the flow, i.e.:

$$Ex_{flow} = Ex_{tm, flow} + Ex_{ch, flow} \quad (A.2)$$

Breaking down the exergy of a flow into a thermo-mechanical and a chemical component offers ample opportunity to distinguish between the active part (source, product) and the non-active part (ballast).

Below, functional exergy efficiencies are specified for a number of frequently applied apparatuses; these efficiencies can be used in analyzing various processes. In this specification sometimes arbitrary choices cannot be avoided; another conclusion is that that functional efficiencies cannot be stated for each apparatus. For example, the function of a condenser in a steam cycle, where the condenser transfers heat to the environment, will only become clear by considering the combination of turbine + condenser. The thermodynamic performance of the condenser can only be assessed regarding the exergy efficiency of this combination. Considering the specific character

of such combinations of apparatuses and the many possibilities, it is decided to define only efficiencies for individual apparatuses. With the given examples, the user will be able to specify functional efficiencies in an analogous manner for combinations of apparatuses closely related functions. On the following pages, in specifying the functional efficiency of each apparatus, a brief description of the process is given and the choice of exergy product and source is elucidated.

A.2 Turbine

In a turbine, a working fluid is expanded in order to deliver mechanical energy via a shaft. During the expansion process, part of the process flow will sometimes be extracted; in the case of steam turbines, for example, extraction steam is used to pre-heat boiler feedwater. In that case the turbine has several outgoing flows.

The shaft power produced is regarded as the product of the process in the turbine, so:

$$Ex_{\text{product}} = P_{\text{shaft}} \quad (\text{A.3})$$

The change in thermo-mechanical exergy of the working fluid is available as an exergy source. Since the chemical exergy of the working fluid does not change, the change in thermo-mechanical exergy is identical to the change in total exergy of the working fluid. Therefore it is supposed that:

$$Ex_{\text{source}} = Ex_{\text{tm, in}} - \sum Ex_{\text{tm, out}} = Ex_{\text{in}} - \sum Ex_{\text{out}} \quad (\text{A.4})$$

And then the turbine's functional efficiency becomes:

$$\eta_{\text{ex, f(turbine)}} = \frac{P_{\text{shaft}}}{Ex_{\text{in}} - \sum Ex_{\text{out}}} \quad (\text{A.5})$$

A.3 Heat exchanger

In a heat exchanger, a fluid (the primary flow) is generally heated by withdrawing heat from another fluid (the secondary flow). It is assumed that the purpose of the heat exchanger is to heat the primary flow. The increase in thermo-mechanical exergy of the primary flow is then the product of the heat transfer process. If the chemical composition of the working fluid remains unchanged, the increase in thermo-mechanical exergy is identical to the total exergy:

$$Ex_{\text{product}} = Ex_{\text{p, out}} - Ex_{\text{p, in}} \quad (\text{A.6})$$

The secondary flow supplies the exergy. The exergy produced by this flow is identical to the exergy change of the secondary fluid in the heat exchanger. For the secondary fluid, it also applies that the chemical composition –and thus the chemical exergy— remains unchanged.

$$Ex_{\text{source}} = Ex_{\text{s, in}} - Ex_{\text{s, out}} \quad (\text{A.7})$$

The functional exergy efficiency of the heat exchanger is then as follows:

$$\eta_{\text{ex, f(heat exchanger)}} = \frac{Ex_{\text{p, out}} - Ex_{\text{p, in}}}{Ex_{\text{s, in}} - Ex_{\text{s, out}}} \quad (\text{A.8})$$

It occurs, for example in the case of feedwater preheaters, that more than one inlet flow is present at the secondary side; then the secondary flows are mixed in the heat exchanger and leave the apparatus as one secondary flow. In this case the functional efficiency becomes:

$$\eta_{ex,f(\text{heat exchanger})} = \frac{Ex_{p,out} - Ex_{p,in}}{\sum Ex_{s,in} - Ex_{s,out}} \quad (\text{A.9})$$

Note In this efficiency heat losses are automatically taken into account, as they are included in the exergy values.

A.4 Pump

In a pump, an incompressible fluid is increased in pressure using shaft power. The exergy increase of the fluid is regarded as the product. Shaft power is available as a source for this exergy increase. The pump's functional efficiency is then:

$$\eta_{ex,f(\text{pump})} = \frac{Ex_{out} - Ex_{in}}{P_{shaft}} \quad (\text{A.10})$$

Electric motors are often used for driving pumps. If the losses of the pump drive are also taken into account, it applies:

$$\eta_{ex,f(\text{pump})} = \frac{Ex_{out} - Ex_{in}}{P_{electric}} \quad (\text{A.11})$$

A.5 Condenser

There are various ways to consider the exergy efficiency of condensers. In the case of a condenser that is intended to discharge heat to the environment, it is not possible to define a "product" and therefore it is impossible to specify a functional efficiency. However, it can occur that the exergy absorbed by the primary fluid in a condenser can be utilized elsewhere in a plant. For example, a hot condenser for the generation of hot water for district heating water or the condenser of a topping cycle that transfers heat to a bottoming cycle. Under these circumstances the same definition for functional exergy efficiency can be applied as in case of heat exchangers. Also with condensers, we may have to deal with several inlet flows at the secondary side. Then for the condenser's functional efficiency can be written:

$$\eta_{ex,f(\text{condenser})} = \frac{Ex_{p,out} - Ex_{p,in}}{\sum Ex_{s,in} - Ex_{s,out}} \quad (\text{A.12})$$

Note Usually in condensers, non-condensable gases are discharged, together with a certain vapour quantity. If this discharge is considered in the process calculation, the exergy of the discharged vapour must be added to Eq. (A.12) as part of the condenser's exergy loss.

A.6 Feedwater preheater

For feedwater preheaters, we can use the same specification as for heat exchangers and condensers.

A.7 Deaerator

In a deaerator, condensate is heated and deaerated using steam. Usually, in steam turbine cycles, several –relatively small– condensate flows are fed to the deaerator. After de-aeration, the liquid mixture is discharged to the boiler feed pump. The deaerator is a direct contact heater. The condensate flow is regarded as the primary, heat-absorbing flow. This flow is heated by mixing it with steam and condensate of higher temperatures; these hot flows can be regarded as the secondary, heat-supplying fluid flows. Analogous to the heat exchanger, for the product can be written:

$$Ex_{\text{product}} = \Phi_{m,p} \cdot (ex_{\text{out}} - ex_{p,\text{in}}) = \Phi_{m,p} \cdot ex_{\text{out}} - Ex_{p,\text{in}} \quad (\text{A.13})$$

in which $\Phi_{m,p}$ is the mass flow condensate heated in the deaerator. Similarly, for the source can be written:

$$Ex_{\text{source}} = \sum (\Phi_{m,s} \cdot ex_{s,\text{in}}) - ex_{\text{out}} \cdot \sum \Phi_{m,s} = \sum Ex_{s,\text{in}} - ex_{\text{out}} \cdot \sum \Phi_{m,s} \quad (\text{A.14})$$

The functional exergy efficiency then becomes:

$$\eta_{\text{ex},f(\text{deaerator})} = \frac{\Phi_{m,p} \cdot ex_{\text{out}} - Ex_{p,\text{in}}}{\sum Ex_{s,\text{in}} - ex_{\text{out}} \cdot \sum \Phi_{m,s}} \quad (\text{A.15})$$

Note The deaerator usually acts as the storage tank of the boiler feed pump. In steady state the storage tank has no effect on the exergy values. In the top of the de-aerator the non-condensable gases are extracted, together with a small amount of steam. If the discharge of this quantity of steam is included in the process calculation, then the exergy of this flow is part of the exergy loss of the deaerator (assuming that the discharged steam is not utilized elsewhere) and has to be subtracted from the product.

A.8 Compressor, fan

In a compressor or fan, a compressible fluid is increased in pressure by means of shaft power. The functional efficiency for compressors is specified in the same way as for pumps, i.e.:

$$\eta_{\text{ex},f(\text{compressor})} = \frac{Ex_{\text{out}} - Ex_{\text{in}}}{P_{\text{shaft}}} \quad (\text{A.16})$$

A.9 Mixer, splitter, valve

Mixers, splitters and valves are tools that are frequently used in energy conversion systems but are not used for energy conversion or energy transfer. They may cause loss of exergy, but it is not possible to define a product. Therefore, it is not possible to specify their exergy efficiency.

Note In system calculations splitters are often used to separate fluid components from a mixture. In that case, actually the splitter acts as a separator. However it is not possible to specify a meaningful functional exergy efficiency for a separator.

A.10 Drum

Circulating evaporators consist of a heat exchanger and a drum. Heating and partial evaporation of the circulating fluid occurs in the heat exchanger. In the drum, liquid and vapour are separated and the recirculated liquid is mixed with the liquid feed flow. If drum and heat exchanger are distinguished as individual apparatuses, the drum can be regarded as an apparatus in which a liquid flow is evaporated using heat withdrawn from the flows to and from the heat exchanger. The exergy absorbed during heating and evaporating the feed flow is then regarded as the product. The exergy extracted from the fluid flow circulating in the heat exchanger is the source of this product. Then the functional efficiency becomes:

$$\eta_{ex,f(\text{drum})} = \frac{Ex_{\text{steam,out}} - Ex_{\text{feedwater,in}}}{Ex_{\text{evaporator,out}} - Ex_{\text{evaporator,in}}} \quad (\text{A.17})$$

A.11 Combustion chamber

In a combustion chamber, fuel is combusted using an oxidiser. The combustion products, the flue gases, are discharged. Heat losses are neglected here and the process is supposed to be adiabatic. In modelling large boilers heat losses have to be included in the heat transfer system. The purpose of combustion is to produce a gas (flue gas) of which only the thermo-mechanical exergy is utilized. The thermo-mechanical exergy produced can thus be regarded as the product of the combustion chamber.

The fuel and oxidant are fed to the combustion chamber, often at elevated temperature and pressure; the thermo-mechanical exergy of the fuel and oxidant flows is then larger than zero. This thermo-mechanical exergy cannot be regarded as a product of the combustion process and must therefore be deducted from the thermo-mechanical exergy of the flue gas. Therefore, the increase in thermo-mechanical exergy is chosen as the product:

$$Ex_{\text{product}} = Ex_{\text{tm, flue gas}} - Ex_{\text{tm, fuel}} - Ex_{\text{tm, oxidiser}} \quad (\text{A.18})$$

As the source for the production of this exergy, the chemical exergy of the fuel and oxidiser is used. The chemical exergy of the flue gas flow is discharged to the environment and thus does not become available as a product. The net available exergy from the exergy source is:

$$Ex_{\text{source}} = Ex_{\text{ch, fuel}} + Ex_{\text{ch, oxidiser}} - Ex_{\text{ch, flue gas}} \quad (\text{A.19})$$

The functional exergy efficiency of the combustion chamber is then defined as follows:

$$\eta_{ex,f(\text{combustion chamber})} = \frac{Ex_{\text{tm, flue gas}} - Ex_{\text{tm, fuel}} - Ex_{\text{tm, oxidiser}}}{Ex_{\text{ch, fuel}} + Ex_{\text{ch, oxidiser}} - Ex_{\text{ch, flue gas}}} \quad (\text{A.20})$$

This definition does not account for the presence of non-combustible components in the fuel that are separated in the combustion chamber in the form of ash or slag. The chemical exergy of these substances is not utilized for heat production and must, therefore, be subtracted from the chemical exergy of the fuel.

In general, ash and slag are discharged from the combustion chamber at high temperature. It is possible to still utilize the thermo-mechanical exergy of these substances; the thermo-mechanical exergy of ash and slag is, therefore, regarded as a product of the combustion process. In that case, the functional efficiency becomes:

$$\eta_{ex,f(\text{combustion chamber})} = \frac{Ex_{tm, \text{flue gas}} + Ex_{tm, \text{ash}} - Ex_{tm, \text{fuel}} - Ex_{tm, \text{oxidiser}}}{Ex_{ch, \text{fuel}} + Ex_{ch, \text{oxidiser}} - Ex_{ch, \text{ash}} - Ex_{ch, \text{flue gas}}} \quad (\text{A.21})$$

Note In practice, not all of the fuel is converted in the combustion chamber. If the non-converted fuel (also in modeling) is discharged in the ash or the flue gas, the chemical exergy of this quantity of fuel is regarded as a ballast flow.

A.12 Boiler

Usually, a boiler is a complex installation, consisting of various apparatuses such as a combustion chamber, several heat exchangers, pumps and fans. In a detailed analysis of a boiler, we have to deal with each individual part and can use the exergy efficiencies for the apparatuses involved.

In some situations a rough consideration of the boiler suffices; the boiler is then regarded as just an apparatus in which a process flow absorbs heat. The quantity of fuel required can then be determined using a predefined thermal efficiency. This enables a complete system calculation without detailed boiler calculations. In such an evaluation, the flue gas flow to the stack is not considered explicitly; the heat discharged with the flue gases is included in the thermal efficiency of the boiler, in which case the exergy of the flue gas when leaving the boiler is also unknown.

The purpose of boiler and optional reheater is to supply thermo-mechanical exergy to the cycle fluid. The change in thermo-mechanical exergy of the cycle fluid can, therefore, be regarded as the product of the processes in the boiler. The chemical composition of the cycle fluid in the boiler remains unchanged; the change in thermo-mechanical exergy is identical to the change in total exergy or:

$$Ex_{\text{product}} = Ex_{tm, \text{steam}} - Ex_{tm, \text{water}} = Ex_{\text{steam}} - Ex_{\text{water}} \quad (\text{A.22})$$

Exergy is supplied to the boiler in the form of fuel. The chemical exergy of the fuel actually functions as the exergy source. In fact, the chemical exergy of the oxidant and flue gas must be considered as was done for the combustion chamber. Since these flows are not considered in the model, they are not explicitly available and must therefore be neglected in the exergy efficiency. The chemical exergy of the oxidant and flue gas, however, is small compared to the chemical exergy of fuel. Therefore it is assumed that:

$$Ex_{\text{source}} = Ex_{ch, \text{fuel}} \quad (\text{A.23})$$

For the functional efficiency of boiler or reheater can be written then:

$$\eta_{ex,f(\text{boiler})} = \frac{Ex_{\text{steam}} - Ex_{\text{water}}}{Ex_{ch, \text{fuel}}} \quad (\text{A.24})$$

Note By using the chemical exergy of the fuel as the fuel exergy, it is implicitly assumed that fuel is supplied at ambient temperature and pressure.

A.13 Moisture separator, flue gas condenser

In a moisture separator or flue gas condenser, a flue gas flow is cooled down to below the water dewpoint, causing part of the vapour present in the gas to condense. The condensate is separated from the gas flow and discharged separately. In general the gas flow will be cooled by transferring heat to another fluid flow. Therefore a moisture separator can be regarded as a heat exchanger. The only difference with a heat exchanger is that vapour is condensed from the secondary flow and discharged separately. The exergy change of the primary flow is regarded as the product. The source for this exergy change is, in principle, the exergy change of the secondary flow. The heat from the condensate can possibly still be utilized. The condensate flow is, therefore, regarded as one of the secondary outlet flows. In that case the functional efficiency becomes:

$$\eta_{ex,f(\text{moisture separator})} = \frac{Ex_{p,out} - Ex_{p,in}}{Ex_{s,in} - \sum Ex_{s,out}} \quad (\text{A.25})$$

A.14 Gasifier

In a gasifier, a solid fuel (usually entering at elevated temperature and elevated pressure) is converted into a gaseous fuel. Gasification takes place by using an oxidiser (air or oxygen) and, possibly, steam. Usually, the product gas leaves the gasifier at an elevated temperature. It may be necessary to cool the gasifier, for example, in the form of jacket cooling. Ash and slag can also be discharged at elevated temperatures. In general the thermo-mechanical exergy of these flows will be used elsewhere in the system.

Analogous to the thermal cold-gas and hot-gas efficiency, the functional exergy efficiency can be defined in two ways.

- a. The gasifier is regarded as a device that supplies only fuel. The thermal energy is not utilized elsewhere and is considered to be a loss. In this way the plant is also considered to define the cold-gas efficiency. In this case, the chemical exergy of the gas produced is regarded as the product of the gasifier. The chemical exergies of the inlet flows are supposed to be the exergy source; the chemical exergy of the discharged ash or slag is regarded as a ballast flow and must be subtracted from the chemical exergy of the fuel. The functional efficiency then becomes:

$$\eta_{ex,f(\text{gasifier})} = \frac{Ex_{ch, \text{productgas}}}{Ex_{ch, \text{solid fuel}} + Ex_{ch, \text{steam}} + Ex_{ch, \text{oxidiser}} - Ex_{ch, \text{ash}}} \quad (\text{A.26})$$

- b. The gasifier is part of a system in which also the thermo-mechanical exergy in the product gas can be utilised. The total exergy of the generated gas, together with the exergy increase of the cooling fluid, is considered to be the exergy product of the gasification process. The thermo-mechanical exergy of the supplied fluid flows: solid fuel, steam, and oxidiser, must be subtracted from this exergy product. Then have the following equation will be obtained:

$$Ex_{\text{product}} = Ex_{\text{productgas}} + Ex_{\text{tm,ash}} + (Ex_{\text{out}} - Ex_{\text{in}})_{\text{cooling}} - (Ex_{\text{tm, solid fuel}} + Ex_{\text{tm, steam}} + Ex_{\text{tm, oxidiser}}) \quad (\text{A.27})$$

Also in this case, the chemical exergy of the ingoing process flows minus the chemical exergy of the ash or slag will be the exergy source. The functional efficiency then becomes:

$$\eta_{\text{ex,f(gasifier)}} = \frac{Ex_{\text{productgas}} + Ex_{\text{tm, ash}} + (Ex_{\text{out}} - Ex_{\text{in}})_{\text{cooling}} - (Ex_{\text{tm, solid fuel}} + Ex_{\text{tm, steam}} + Ex_{\text{tm, oxidiser}})}{Ex_{\text{ch, solid fuel}} + Ex_{\text{ch, steam}} + Ex_{\text{ch, oxidiser}} - Ex_{\text{ch, ash}}} \quad (\text{A.28})$$

A.15 Reformer

A reformer can be seen as a device that should change the chemical composition of a fluid flow. The product will be a combustible gas. In order to make the conversion into the desired product gas possible, steam is added to the feed; as the fluid flow in the reformer will undergo an endothermic reaction heat transfer to this flow is required. This heat is usually obtained in a furnace where a certain amount of fuel is combusted and radiant heat as well as convection heat from the flue gases is transferred to the reformer tubes. In the calculation, the reformer and combustion chamber can be regarded as individual apparatuses. Then hot flue gas enters the reformer and the heat required for the reforming process is obtained from this flue gas flow. The change in chemical exergy of the product gas is assumed to be the product of the reforming process, i.e.:

$$Ex_{\text{product}} = Ex_{\text{ch, productgas}} - Ex_{\text{ch, steam}} - Ex_{\text{ch, feed}} \quad (\text{A29})$$

And the change in the thermo-mechanical exergy of the flue gas is considered to be the source. However, not only the chemical exergy of the product gas flow changes, but also the thermo-mechanical exergy can change. As it is assumed that this thermo-mechanical exergy can be beneficially utilized, it may be subtracted from the exergy produced by the flue gas, i.e.:

$$Ex_{\text{source}} = Ex_{\text{tm, flue gas, in}} - Ex_{\text{tm, flue gas, out}} - (Ex_{\text{tm, productgas}} - Ex_{\text{tm, steam}} - Ex_{\text{tm, feed}}) \quad (\text{A30})$$

It is assumed that the chemical composition of the flue gas during cooling in the reformer does not change. The change in thermo-mechanical exergy is then identical to the change in total exergy. The functional efficiency then becomes:

$$\eta_{\text{ex,f(reformer)}} = \frac{Ex_{\text{ch, productgas}} - Ex_{\text{ch, steam}} - Ex_{\text{ch, feed}}}{(Ex_{\text{tm, flue gas, in}} - Ex_{\text{tm, flue gas, out}}) - (Ex_{\text{tm, productgas}} - Ex_{\text{tm, steam}} - Ex_{\text{tm, feed}})} \quad (\text{A31})$$

A.16 Fuel cell

In a fuel cell, electricity is produced by an electrochemical reaction of a fuel with an oxidiser (usually air). Both flows are separately fed to the anode and the cathode of the fuel cell, respectively. In practice, fuel and oxidiser also contain components that do not take part in the reactions in the cell; these components will be discharged again together with the reaction products. Anode flow and cathode flow leave the cell separately. The fuel cell produces electricity in the form of a direct current. The electrical energy produced can be regarded as the product of the fuel cell. The change in chemical exergy of fuel and oxidant is available as the exergy source. The thermo-mechanical exergy of the fuel and oxidant flow, however, also changes as a result of the generated heat in the cell. The thermo-mechanical exergy absorbed by these flows can be utilized elsewhere

in the system and may therefore be subtracted from the change in chemical exergy of the flows, i.e.:

$$Ex_{\text{source}} = (Ex_{\text{in}} - Ex_{\text{out}})_{\text{ch, fuel}} + (Ex_{\text{in}} - Ex_{\text{out}})_{\text{ch, oxidiser}} - (Ex_{\text{out}} - Ex_{\text{in}})_{\text{tm, fuel}} - (Ex_{\text{out}} - Ex_{\text{in}})_{\text{tm, oxidiser}} \quad (\text{A32})$$

This equation shows that the exergy source equals the change in total exergy of fuel and oxidiser flows. The functional efficiency then becomes:

$$\eta_{\text{ex, f(fuel cell)}} = \frac{P_{\text{electric, direct current}}}{(Ex_{\text{fuel, in}} - Ex_{\text{fuel, out}}) + (Ex_{\text{oxidiser, in}} - Ex_{\text{oxidiser, out}})} \quad (\text{A33})$$

Note In the fuel cell, the supplied chemical exergy is converted partly into electricity and partly into thermo-mechanical exergy (heat). In the Eq. (A33), the part converted into thermo-mechanical exergy is regarded as ballast flow. The larger the ballast flow, the smaller the quantity of exergy that can be converted into electricity. The objective of the fuel cell is to convert as much as possible of the chemical exergy into electricity. Equation (A33) does not directly indicate the extent to which the fuel cell has succeeded in this.

A.17 Scrubber, separator, saturator

It appears not to be possible to define a product when dealing with scrubbers, separators and saturators. Therefore, it is impossible to specify a useful functional efficiency.

A.18 Reactor

For reactors, a functional efficiency can be defined only after establishing the function of the reactor. For specific designs of reactors (for example, a combustion chamber and a reformer), very different specifications have been given. In these specifications differences occur depending on how heat is fed to the process and the way the exhaust flows can be used (for chemical or thermo-mechanical purposes). Therefore, it is not possible to define a useful general functional efficiency for reactors.

Note In cases in which a functional efficiency cannot be specified, it is always possible to indicate the quality of the conversion process by using the universal efficiency. In drawing conclusions, one must bear in mind that the universal efficiency always has a value larger than, or identical to, the functional efficiency.