6 Value diagrams and exergy efficiencies

Nico Woudstra

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 build plants. Therefore exergy efficiencies have appeared to be very useful. Exergy efficiencies however 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 provide briefly 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 figure 6.1a the temperature of both flows in a heat exchanger are given as a function of the heat transferred to the primary flow, the flow that is heated flow. In figure 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:

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

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 in principle 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 equation 6.1 from the inlet temperature $T_{s,in}$ to the outlet temperature $T_{s,out}$:

$$Ex_s = \int_{T_{s,in}}^{T_{s,uit}} (1-\frac{T_0}{T_s})\cdot dQ \tag{6.2}$$

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:

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

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:

$$Ex_p = \int_{T_{p,in}}^{T_{p,uit}} (1 - \frac{T_0}{T_p}) \cdot dQ \tag{6.4}$$

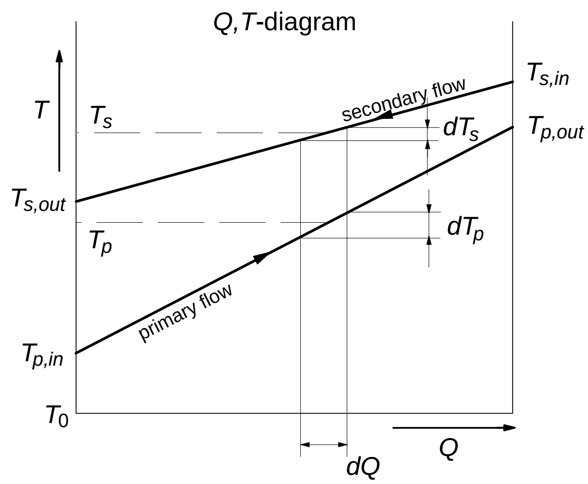


Figure 6.1a *Heat transfer process represented in a Q,T-diagram*

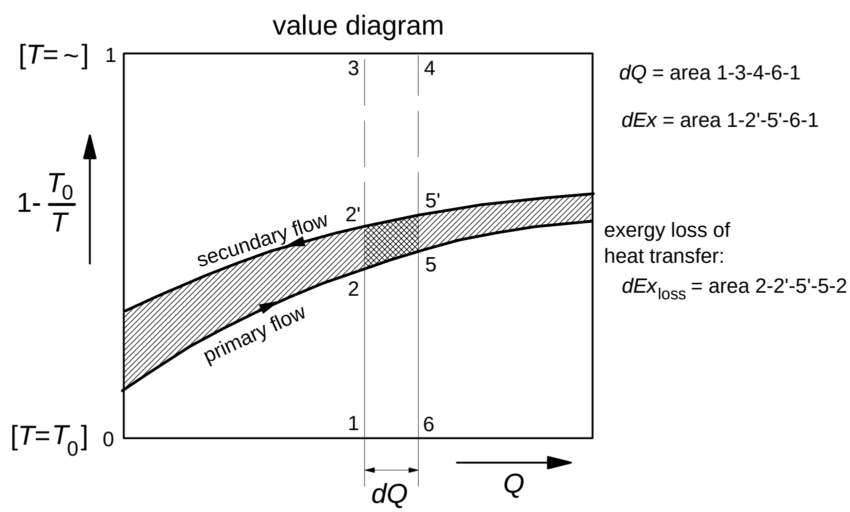


Figure 6.1b *Heat transfer process represented in a value diagram*

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.

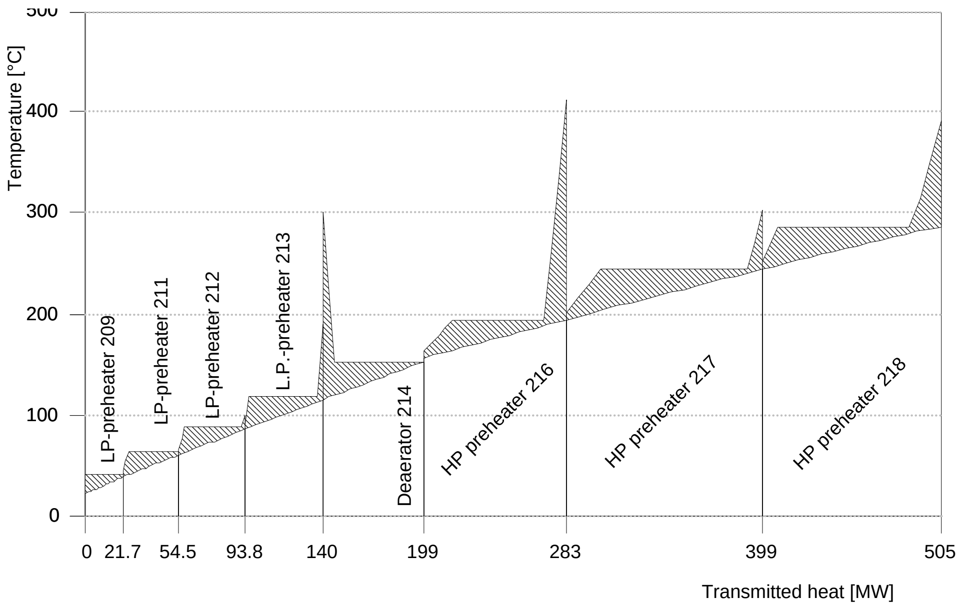


Figure 6.2 *Q,T-diagram for the heat transfer in a feedwater train*

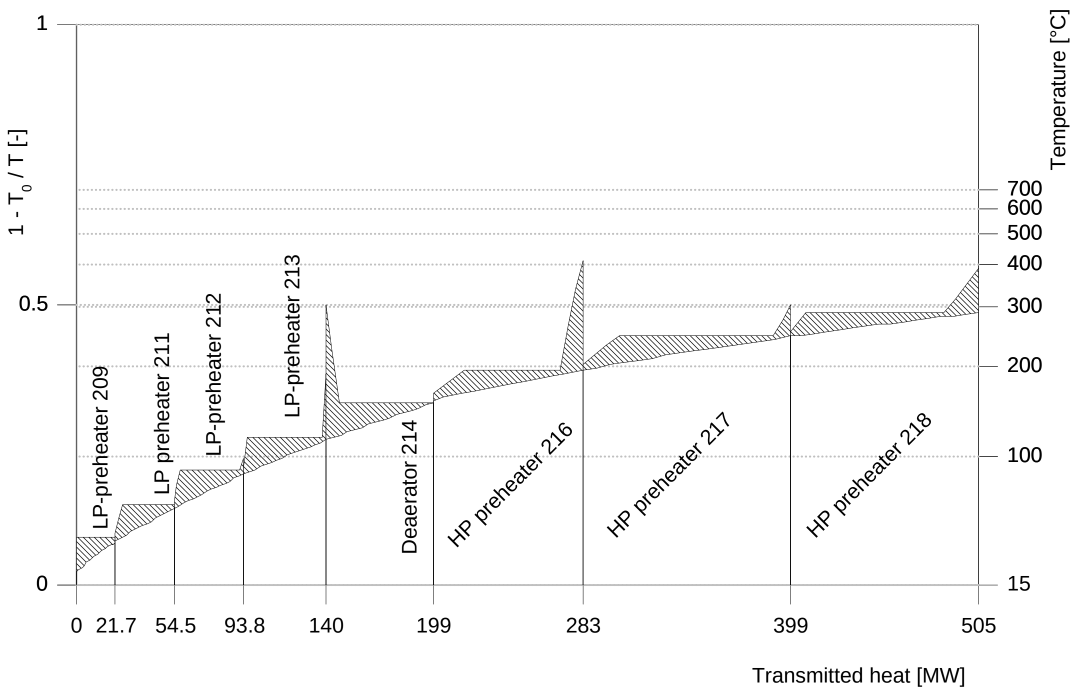


Figure 6.3 *Value diagram of the same feedwater train as in figure 5.2*

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.

The value diagram for this feedwater train is shown in figure 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.

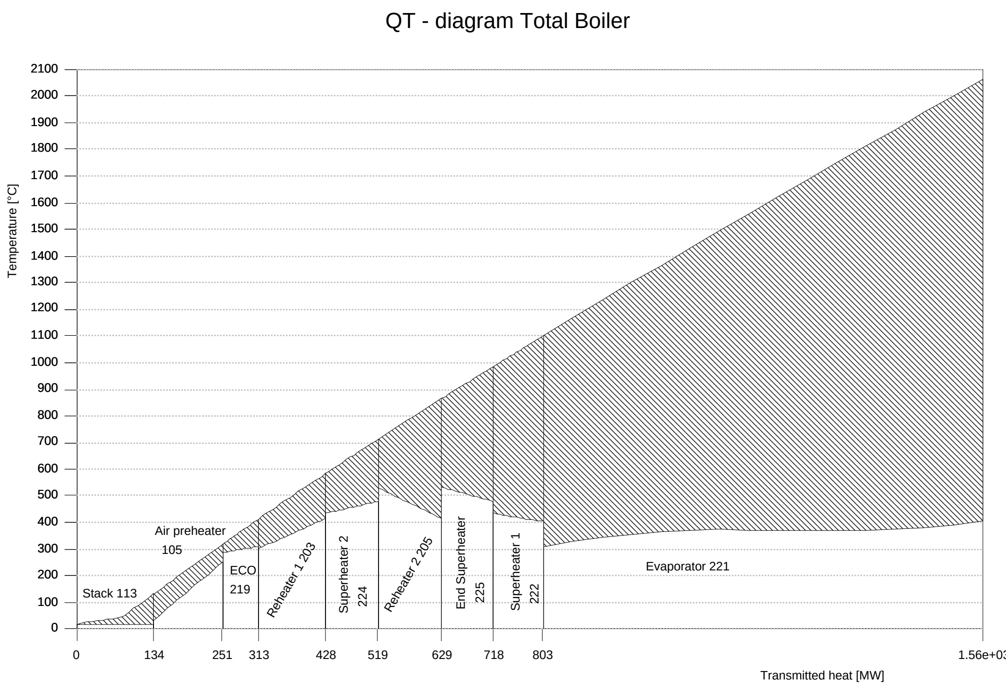


Figure 6.4 *Q,T-diagram of heat transfer in a large steam boiler*

The heat transfer in a large steam boiler is presented as another example. The *Q*,*T*-diagram is shown in figure 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 figure 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 economiser and air preheater. With exception of the economiser 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 figure 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.

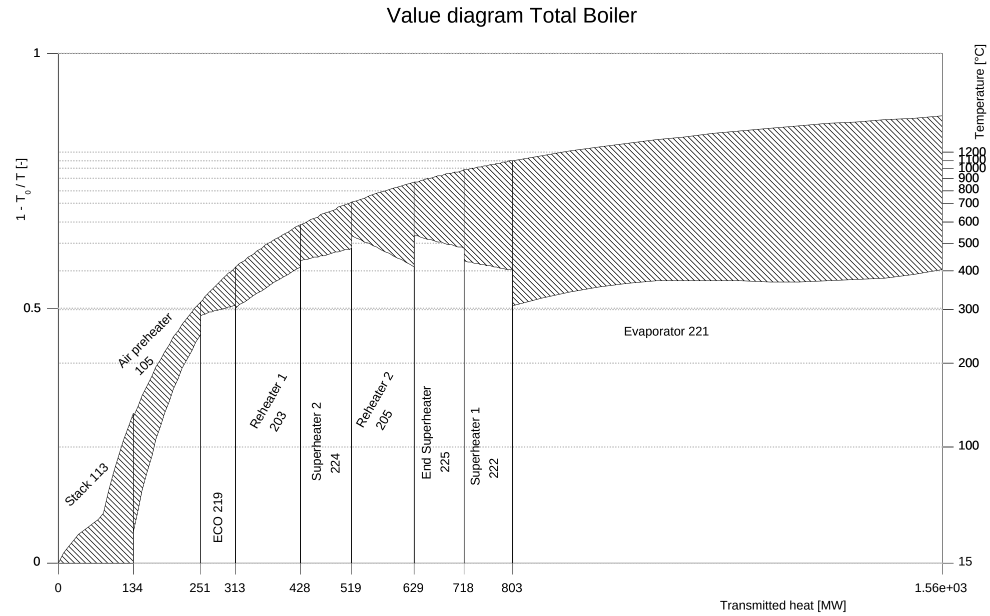


Figure 6.5 *Value diagram of heat transfer in a large steam boiler*

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:

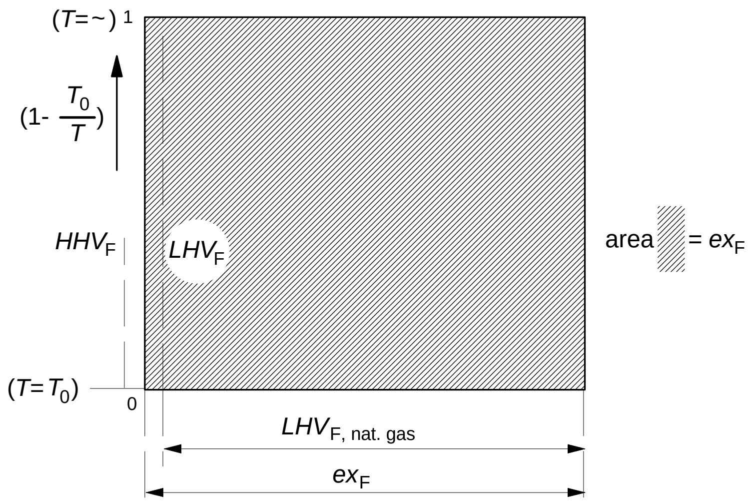
$$f_{ex,\mathrm{F}} = \frac{ex_\mathrm{F}}{LHV_\mathrm{F}} \qquad (6.5)$$

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 Slochteren quality natural gas.

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 figure 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.



Figuur 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 oxidiser 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 figure 6.7 the temperature curve of the flue gas during cooling down is represented, assuming that the exergy factor $f_{ex,\mathrm{F}}$ is higher than 1. At the right border of the diagram the temperature of the gas is the adiabatic combustion temperature.

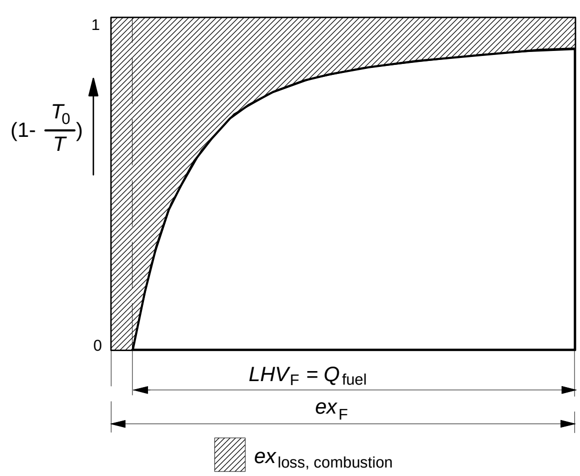


Figure 6.7 *Value diagram with the temperature curve of flue gas during cooling from adiabatic combustion till ambient temperature*

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.

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 figure 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 figure 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.

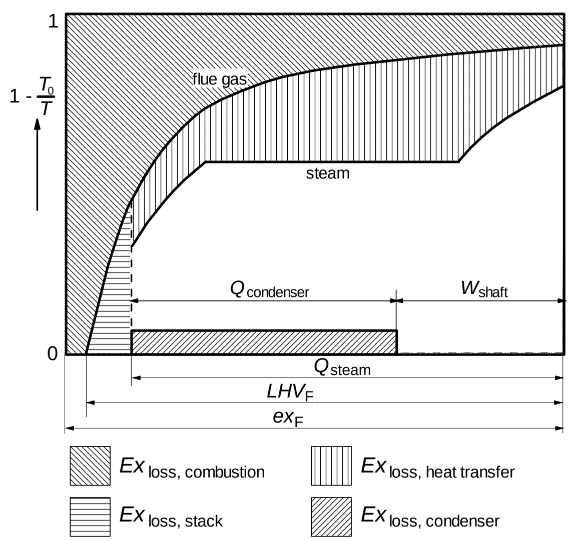


Figure 6.8 *Value diagram of a conventional steam cycle power station*

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 are 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 figure 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 figure 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.

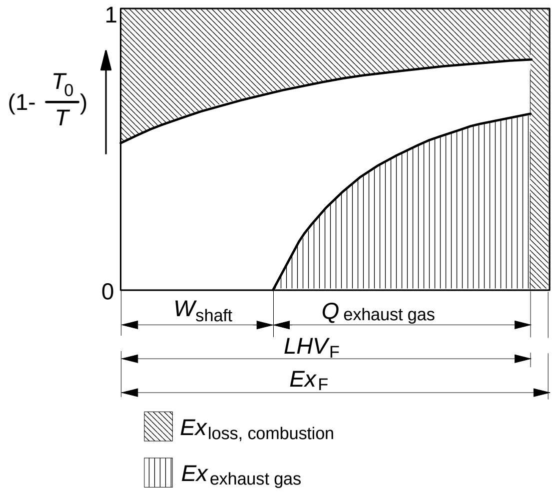


Figure 6.9 *Value diagram of a gas turbine cycle*

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 figure 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 figure 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.

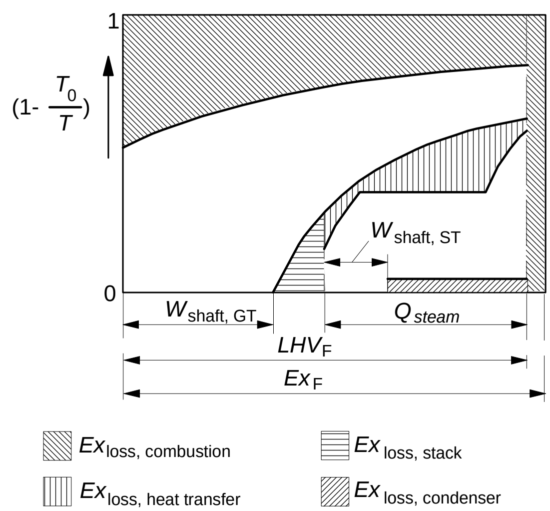


Figure 6.10 *Value diagram of combined cycle plant*

After expansion in the turbine the temperature of the flue gasses is rather high (around 500 ˚C). The lower curve in figure 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.

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. 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:

$$\eta_{Ex,\mathrm{u}} = \frac{\sum Ex_{\mathrm{out}}}{\sum Ex_{\mathrm{in}}} \tag{6.6}$$

In which:

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

$\sum Ex_{\mathrm{in}}$ 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.

$$\sum Ex_{\mathrm{in}} = \sum Ex_{\mathrm{out}} + \sum Ex_{\mathrm{loss}} \tag{6.7}$$

For (6.6), we can also write:

$$\eta_{ex,\mathrm{u}} = \frac{\sum Ex_{\mathrm{in}} - \sum Ex_{\mathrm{loss}}}{\sum Ex_{\mathrm{in}}} \tag{6.8}$$

[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.

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.

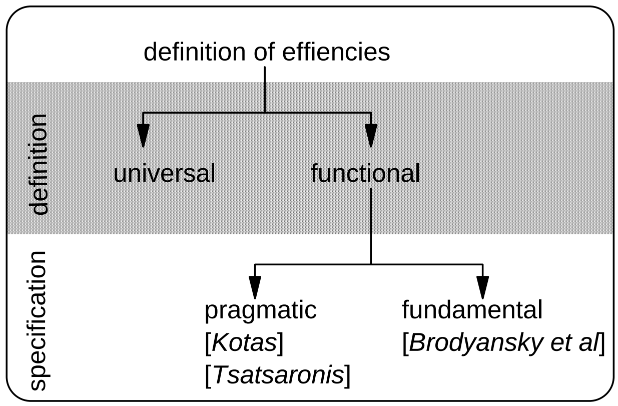


Figure 6.11 *Possible division of exergy efficiencies*

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:

$$\eta_{ex,\mathrm{f}} = \frac{\sum Ex_{\mathrm{product}}}{\sum Ex_{\mathrm{source}}} \tag{6.9}$$

In which:

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

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

$$\sum Ex_{\mathrm{source}} = \sum Ex_{\mathrm{in}} - \sum Ex_{\mathrm{ballast}} \tag{6.10}$$

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

$$\sum Ex_{\mathrm{product}} = \sum Ex_{\mathrm{out}} - \sum Ex_{\mathrm{ballast}} \tag{6.11}$$

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.:

$$\sum Ex_{\mathrm{source}} = \sum Ex_{\mathrm{product}} + \sum Ex_{\mathrm{loss}} \tag{6.12}$$

The equations (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 $Ex_{\mathrm{ballast.}}$ In addition to the definition given, a more detailed elaboration/specification (see also Figure 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.

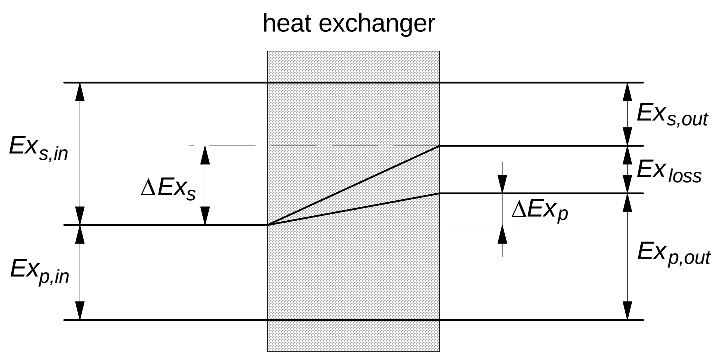


Figure 6.12 *Change in exergy quantities during heat transfer*

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

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

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 equation (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:

$$\eta_{Ex,\mathrm{f(heat\ exchanger)}} = \frac{\Delta Ex_\mathrm{p}}{\Delta Ex_\mathrm{s}} = \frac{Ex_\mathrm{p,out} - Ex_\mathrm{p,in}}{Ex_\mathrm{s,in} - Ex_\mathrm{s,out}} \tag{6.14}$$

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 ($Ex_\mathrm{s,in} + Ex_\mathrm{p,in}$), while in equation (6.14) the exergy loss is related to the exergy change in the secondary flow (ΔEx_s). Since ΔEx_s is always smaller than ($Ex_\mathrm{s,in} + Ex_\mathrm{p,in}$), the functional efficiency (6.14) is more sensitive to changes in exergy loss than the universal efficiency (6.13).

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