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Introduction to the Concept of Exergy - for a Better Understanding of Low-Temperature-Heating and High-Temperature-Cooling Systems

Introduction to the Concept of Exergy – for a Better Understanding of Low-Temperature-Heating and High-Temperature-Cooling Systems

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Abstract

Chapter 1 describes the characteristics of a thermodynamic concept, exergy, in association with building heating and cooling systems. Exergy is the concept that explicitly indicates ‘what is consumed’. All systems, not only engineering systems but also biological systems including the human body, work feeding on exergy, consuming its portion and thereby generating the corresponding entropy and disposing of the generated entropy into their environment. The whole process is called ‘exergy-entropy process’. The features of ‘warm’ exergy and ‘cool’ exergy and also radiant exergy are outlined. General characteristics of exergy-entropy process of passive systems, which would be a prerequisite to realize low exergy systems, are discussed together with the exergy-entropy process of the global environmental system.

Chapter 2 introduces the various forms of exergy and the mathematical formulations used to evaluate them. The exergy balance on an open steady state system, which is much more relevant to thermodynamic analysis of energy systems, is also described, as well as the different exergetic efficiency factors introduced in the thermodynamic analysis of energy systems. Next, an exergy analysis example is outlined through an air-conditioning application. Air-conditioning applications are widely used in heating and cooling of buildings.

Chapter 3 introduces an example of exergy calculation for space heating systems. The issues to have a better understanding of low-exergy systems for heating and cooling are raised. It is suggested that a prerequisite for low exergy systems would be rational passive design of building envelope systems.

Preface

This report is part of the work done within the project realized for the International Energy Agency (IEA): Energy Conservation in Buildings and Community Systems Programme (ECBCS), "Annex 37: Low Exergy Systems for Heating and Cooling".

The general objective of the Annex 37 is to promote rational use of energy by means of facilitating and accelerating the use of low valued and environmentally sustainable energy sources for heating and cooling of buildings.

VTT Technical Research Centre of Finland is the coordinator of the Annex 37. Eleven countries participate: Canada, Denmark, Finland, France, Germany, Italy, Japan, The Netherlands, Norway, Poland and Sweden.

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Åsa Nystedt

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Appendix C: Forms of exergy

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List of symbols

C_0	:	Velocity relative to the earth surface (m/s)
c_p	:	Specific isobaric heat capacity (J/(kg·K))
E	:	Exergy (J)
e	:	Specific exergy (J/kg)
g	:	Acceleration due to gravity (m/s ²)
h	:	Specific enthalpy (J/kg)
I	:	Irreversibility (J)
M	:	Molecular weight (g/mole)
\dot{M}	:	Mass flow (kg)
P	:	Pressure (Pa)
Q	:	Heat (J)
R	:	Specific ideal gas constant (J/(kg·K))
S	:	Entropy (J/(kg·K))
s	:	Specific entropy (J/(kg·K))
T	:	Temperature (K)
v	:	Specific volume (m ³ /kg)
W	:	Work (J)
x	:	Mole fraction
Z_0	:	Altitude above sea level (m)

Greek Symbols

ϕ	:	Relative humidity (%)
η	:	Conventional exergetic efficiency (%)
η_u	:	Utilizable exergy coefficient (%)
τ	:	Non-dimensional exergetic temperature
ω	:	Specific humidity (%)
ψ	:	Rational exergetic efficiency (%)

Subscripts

0	:	Reference or ambient state
a	:	Dry air
ch	:	Chemical
elec	:	Electrical
f	:	Saturated liquid
g	:	Saturated vapor
gen	:	Generation
I	:	Input
In	:	incoming exergy flow
k	:	Kinetic
o	:	Output
out	:	Outgoing exergy flow
p	:	Potential
ph	:	physical or thermochemical
r	:	Heat reservoir
sh	:	Shaftwork
tr	:	Transiting
v	:	Water vapor
w	:	Work

Superscripts

Q	:	Heat
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1. Introduction to the concept

Masanori Shukuya

1.1 Introduction

Chapter 1 describes the general characteristics of a thermodynamic concept, exergy, which enables us to articulate what is consumed by all working systems, whether they are man-made systems such as thermo-chemical engines and electricity-driven heat pumps or biological systems such as microbes, plants, and animals including the human body. This article focuses especially on its application to describing building heating and cooling systems.

People often claim that energy is consumed; this is not only in everyday conversation but also even in scientific discussion associated with so-called energy and environmental issues. This claim, however, conflicts with the first law of thermodynamics stating that the total amount of energy is conserved even though forms of energy may change from one to another. All macroscopic natural phenomena happening around us involve the dispersion of energy and matter, which in due course change their forms from one to another, but the total amount of energy and matter involved is never consumed but necessarily conserved.

When we use such expressions as “energy consumption”, “energy saving”, and even “energy conservation”, we implicitly refer to “energy” as intense energy available from fossil fuels or from condensed uranium. But, it is confusing to use one of the most well-established scientific terms, energy, to mean “to be conserved” and “to be consumed” simultaneously. This is why we need to use the thermodynamic concept, exergy, to articulate what is consumed.

Over the last two decades various so-called “energy saving” measures have been conceived, developed, and implemented in building envelope systems and also their associated environmental control systems such as lighting, heating, and cooling systems. Those measures can be categorized into two groups: those for “passive” systems and those for “active” systems.

“Passive” systems are defined as building envelope systems to make use of various potentials to be found in the immediate environment such as the sun, wind, and others to illuminate, heat, ventilate, and cool the built environment. The history of passive systems is very long; we may say that it emerged with the evolution of human being.

The recent development of material science has brought about various building materials such as low-emissivity coated glass and others; this enables us to design advanced passive systems.

“Active” systems are the systems consisting of various mechanical and electric components such as fans, pumps, heat pumps, and others, all of which work by the use of fossil fuels. Most of the active systems available these days have been developed with an assumption of the abundant use of fossil fuels so that they do not necessarily work in harmony with passive systems.

Optimal thermal environmental design with thermally-well-insulated glazing materials with other thermally-well-insulated building-envelope materials having appropriate heat capacity enables us to realize “passive” solar heating systems. However, it does not mean that active heating systems are no longer required. We need new types of active systems that can work in harmony with advanced passive systems.

Low-temperature-heating systems are such kind of “active” heating systems which should fit the built environment to be conditioned primarily by “passive” heating systems. A good thermal-environmental condition within built spaces in the winter season can be provided basically with the installation of thermally-well-insulated building materials with appropriate heat capacity, which make it possible to utilize heat sources of lower temperature for heating.

In summer season, a moderate thermal-environmental condition within built spaces may be provided with a combination of nocturnal ventilation, the installation of appropriate shading devices for glass windows, and the reduction of internal heat gain in addition to the use of thermally-well insulating materials with appropriate heat capacity for building envelopes. This would allow the utilization of cold sources with higher temperature for cooling.

The use of the exergy concept in describing various heating and cooling systems, whether they are passive or active, would enable us to have a better picture of what low-temperature-heating and high-temperature-cooling systems are.

1.1.1 Description of a system as an exergy-entropy process

Let us assume a building environmental control system such as lighting, heating, or cooling systems. Energy and matter are supplied into the system so that it works. The

inputs are exactly the same as the outputs under steady-state conditions. This is due to the law of energy and mass conservation. If it is so, why do not we reuse the energy and matter as output directly? This very fundamental question in terms of life was once asked by Schrödinger some fifty years ago (Schrödinger, 1945). If we could have used the wasted energy and matter, most of the so-called energy and environmental problems would have been already solved.

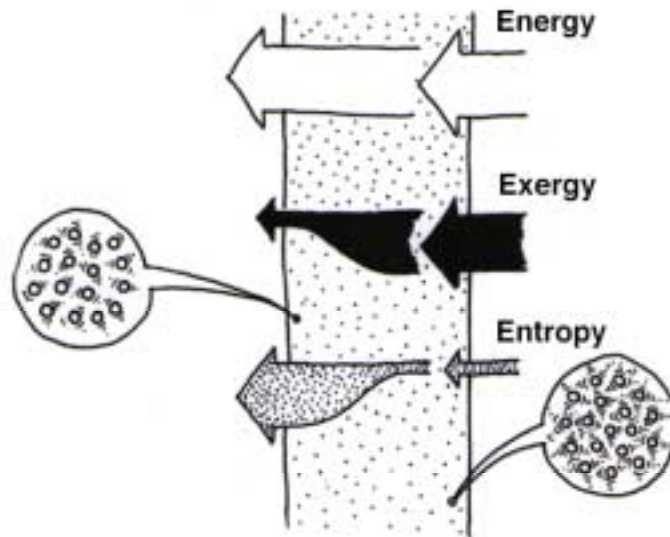


Fig. 1. Energy, exergy, and entropy flow in and out a building envelope system. The amounts of energy flowing in and out are the same under thermally steady-state condition according to the law of energy conservation; on the other hand, the amount of entropy flowing out is larger than flowing in according to the law of entropy increase. The amount of exergy flowing out is smaller than flowing in, since exergy is consumed within the system to produce entropy.

The most general answer to the above question would be that the energy and matter as input are different from those as output; or you may say that the energy and matter as output have something that a system in question must discard. To make the answer clearer, we use the concepts of exergy and entropy, which can express the difference in energy and matter between input and output explicitly. Exergy and entropy, both of which are thermodynamic concepts, can show us what is the resource and what is the waste; “exergy” is the concept to articulate “what is consumed” and “entropy” is “what is disposed of”. Stating in the other way, “exergy” is the concept, which quantifies the potential of energy and matter to disperse in the course of their diffusion into their environment and “entropy” is the concept which quantifies the state of dispersion, to what extent the energy and matter in question are dispersed.

Let us take a microscopic view in order to make the concepts easier to understand. Energy transfer like heat transfer is a transfer of the vibration of particles, which compose of, for example, a building envelope system as shown in Fig. 1. We assume a steady-state condition that the right-hand side of the system is warmer than the left-hand side. The particles in the warmer side of the building envelope vibrate rather strongly; that is, the energy flowing into the building envelope accompanies a certain amount of exergy. The vibration disperses in the course of energy transfer; that is, a part of the exergy is consumed as the exergy flows. As a result, the energy flowing out the building envelope is accompanied with a smaller amount of exergy.

As a result of the dispersion of vibration, the state of dispersion as a whole within the system increases. This is the generation of entropy, the law of entropy increase*, which is parallel to the law of energy and mass conservation. The amount of increased entropy is proportional to that of consumed exergy and the proportional constant is the ambient temperature in the Kelvin scale as described later.

Since the steady-state condition is being assumed, the distribution of the temperature inside the building envelope is unchanged. This implies that the amount of entropy contained by the whole of the building envelope system is constant. The entropy of a substance, which is a function of temperature and pressure, remains unchanged unless the temperature and the pressure of the substance increases or decreases. As described above, a certain amount of entropy is generated due to exergy consumption within the building envelope system. This generated entropy must be discarded into the surrounding, namely outdoors, from the building envelope system, otherwise it turns out to be contradictory with our assumption of the steady-state condition and the characteristics of the entropy as a function of temperature and pressure. It is important for us to recognize that the energy flowing out the building envelope is accompanied with not only a decreased amount of exergy but also an increased amount of entropy. Disposing of the generated entropy from the system makes room for feeding on exergy and consuming it again.

We call the process described above as exergy-entropy process (Shukuya and Komuro, 1996). Table 1 shows the four fundamental steps of exergy-entropy process. Any working systems perform these four steps in series and cyclically. Heating and cooling systems are no exception.

* There is a rather strong belief among scientists and engineers that entropy is one of the concepts which is most difficult to understand. I think that this is not necessarily true. Those who are interested more in the concept of entropy than described here in this article should consult, for example, a book written by Atkins [1984], which I think best describes the characteristics of entropy.

Table 1. Four Steps of Exergy-Entropy Process.

1.	Feed on exergy
2.	Consume Exergy
3.	Generate Entropy
4.	Dispose of Entropy
Disposing of the generated entropy from the system makes new room for feeding on exergy and consuming it again. Thus the process cycles.	

1.2 Exergy balance equation

Let us introduce a general expression of exergy balance using the case of the above-mentioned simple building envelope system. The purpose here is to outline the structure of the exergy balance equation and we do not discuss the detailed mathematical expression. Those who are interested in the detailed mathematical expressions should refer to Bejan (1988), Shukuya (1994), and others, in addition to Chapter 2 of this report.

Energy is the concept to be conserved so that the energy flowing in must be equal to the sum of the energy stored within the system and the energy flowing out from the system. This energy balance can be expressed as follows.

$$(\text{Energy input}) = (\text{Energy stored}) + (\text{Energy output}) \quad (1.1)$$

Since the steady-state condition is being assumed here, there is no energy storage and hence the above equation turns out to be the following simpler form.

$$(\text{Energy input}) = (\text{Energy output}) \quad (1.2)$$

Secondly, let us set up the entropy equation consistent with the above two equations. Energy flowing into the system as heat is more or less dispersed energy. Heat is a energy transfer due to dispersion, thus entropy necessarily flows into the system as heat flows in and some amount of entropy is generated inevitably within the system in the course of heat transmission. The sum of the entropy input and the entropy generated must be in part stored or in part flows out of the system. Therefore the entropy balance equation can be expressed in the following form.

$$(\text{Entropy input}) + (\text{Entropy generated}) = (\text{Entropy stored}) + (\text{Entropy output}) \quad (1.3)$$

Since the steady-state condition is being assumed, there is no entropy storage as well as no energy storage. Therefore, the above entropy balance equation turns out to be

$$(\text{Entropy input}) + (\text{Entropy generated}) = (\text{Entropy output}) \quad (1.4)$$

The fact that the outgoing entropy from the system includes the entropy generated within the system suggests that the system disposes of the generated entropy with the entropy output.

Combining the energy and entropy balance equations brings about the exergy balance equation. Entropy (or entropy rate) has a dimension of J/K (or W/K) and energy (or energy rate) has a dimension of J (or W). Therefore we need a kind of trick to combine the two equations.

Generally speaking, energy contained by a body, which has an ability to disperse, is called an energy resource. Such an energy resource exists within the environmental space, which is filled with dispersed energy. The dispersed energy level of the resource surrounded by the environmental space can be expressed as the product of the entropy contained by the resource and its environmental temperature in the Kelvin scale. The same expression applies to the waste discarded by the system. Therefore the entropy balance equation can be rewritten as follows.

$$(\text{Entropy input}) \times T_e + (\text{Entropy generated}) \times T_e = (\text{Entropy output}) \times T_e \quad (1.5)$$

Where T_e is the environmental temperature. The product of entropy and environmental temperature is called “anergy”, which implies dispersed energy. Using the term “anergy”, the above equation can be expressed in the following form, anergy balance equation.

$$(\text{Anergy input}) + (\text{Anergy generated}) = (\text{Anergy output}) \quad (1.6)$$

Provided that “anergy” is a portion of energy that is already dispersed, then the other portion is not yet dispersed. Stating in another way, energy consists of two parts: the dispersed part and the part, which can disperse. The latter is “exergy”. Now let us take the difference of the two equations, energy balance equation (1.2) and anergy balance equation (1.6). This operation brings about

$$[(\text{Energy input}) - (\text{Energy input})] - (\text{Energy generated}) = [(\text{Energy output}) - (\text{Energy output})]. \quad (1.7)$$

“Energy generated” is such energy that originally had an ability to disperse and that has just dispersed. We can state this in the other way; that is, exergy is consumed. Energy generation is equivalent to exergy consumption. Using the term “exergy”, the above equation can be reduced to the following equation.

$$(\text{Exergy input}) - (\text{Exergy consumed}) = (\text{Exergy output}) \quad (1.8)$$

This is the exergy balance equation for a system under steady-state condition such as the building envelope system shown in Fig. 1. Exergy consumed, which is equivalent to energy generated, is the product of entropy generated and the environmental temperature.

$$(\text{Exergy consumed}) = (\text{Environmental temperature}) \times (\text{Entropy generated}) \quad (1.9)$$

Exergy consumed is exactly proportional to the entropy generated with the proportional constant of environmental temperature.

1.3 Warm exergy and cool exergy

The amount of exergy contained by a substance varies with its temperature and also with its environmental temperature. Fig. 2 shows an example of thermal exergy contained by 81 m³ (= 6m x 5m x 2.7m) of air as a function of its temperature in the case of an environmental temperature of 288 K (=15°C). It should be noted that air has a certain amount of exergy both when the air temperature is higher than the environment and when the air temperature is lower than the environment. Appendix A shows a mathematical formula used to draw Fig. 2.

The exergy contained by air at a temperature higher than its environment is an ability of thermal energy contained by the air to disperse into the environment. On the other hand, the exergy contained at a temperature lower than its environment is an ability of the air, in which there is a lack of thermal energy compared to the environment, to let the thermal energy in the environment flow into it. We call the former “warm” exergy and the latter “cool” exergy (Shukuya, 1996).

Either “warm” exergy or “cool” exergy described above is a quantity of state contained by a substance. We have room temperature higher than the outdoor environment when the space is heated. In such a case room air has “warm” exergy as a quantity of state. On the other hand, when we have a room temperature lower than the outdoor environment, room air has “cool” exergy as a quantity of state.

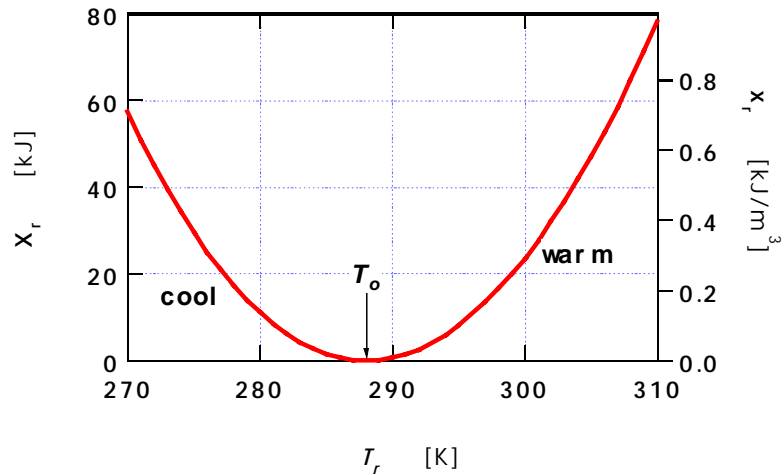


Fig. 2. Thermal exergy contained by air as a function of temperature, T_r . Air volume is assumed to be 81m^3 ($= 6\text{m} \times 5\text{m} \times 2.7\text{m}$). Environmental temperature, T_o , is 288K ($=15^\circ\text{C}$). Air at a temperature higher than the environmental temperature has “cool” exergy and the air at a temperature lower than the environmental temperature has “warm” exergy (See Appendix A, formula A.1).

Thermal exergy, whether it is warm exergy or cool exergy, flows through walls, by a combination of convection, conduction, and radiation. The case shown in Fig. 1 is when the environmental temperature, namely outdoor temperature, is lower than the indoor temperature. In this case, “warm” exergy flows in the internal surface and out the external surface of the building envelope system. If the environmental temperature is higher than the indoor temperature, namely the summer condition, the room air has “cool” exergy, which flows through the building envelope system.

The direction of energy flow changes depending on the temperature profile, whether the indoor temperature is higher or lower than the outdoor temperature, but the direction of exergy flow is always the same from the indoors to the outdoors, external environment. What changes is whether it is “warm” exergy or “cool” exergy depending on, whether indoor temperature is higher or lower than the outdoor temperature.

Space heating systems, whether they are low-exergy consuming or not, are the systems that supply and consume exergy for keeping “warm” exergy as a quantity of state

contained by room space in a certain desired range. Space cooling systems, on the other hand, whether they are low-exergy consuming or not, are the systems that supply and consume exergy for keeping “cool” exergy as a quantity of state contained by room space in a certain desired range. As described above, exergy consumption is always accompanied with entropy generation, thus the generated entropy must be discarded constantly from the room space to the outdoor environment to keep “warm” or “cool” exergy within a desired range.

1.4 Radiant exergy

Radiant exergy transfer plays more important role in low-temperature-heating or high-temperature-cooling systems than in conventional air heating or cooling systems, because they require heat sources with a rather large surface area whose temperature is only slightly higher than room air temperature.

For this reason, it would be very important to be able to evaluate radiant exergy. Fig. 3 shows an example of radiant exergy emitted by a black surface of 1 m^2 in the case of environmental temperature of 20°C ($=293 \text{ K}$) given by Takahashi et al. (2000). Appendix B shows a mathematical formula used to draw Fig. 3.

Supposing that there is a radiant panel of 2 m^2 with a surface temperature of 40°C , this panel emits 9 W of “warm” radiant exergy. If the surface temperature decreases from 40°C to 30°C , the “warm” radiant exergy drops dramatically from 9 W down to 2 W .

In the case of cold source of the surface temperature of 6°C , the panel emits 4 W of the “cool” radiant exergy. If the surface temperature increases from 6°C to 14°C , the “cool” exergy drops dramatically from 4 W down to 0.2 W .

This suggests that low exergy systems for heating and cooling of buildings are realized provided that heating and cooling exergy requirements for room space is decreased by the installation of rational building envelope systems, thus the heating and cooling is provided at a temperature close to room temperature.

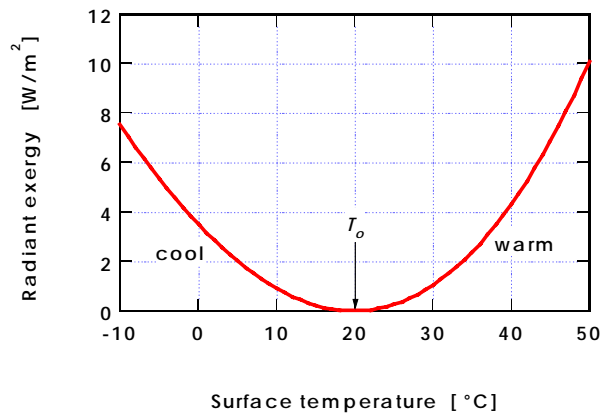


Fig. 3. An example of radiant exergy emitted by a black surface of 1 m^2 when the environmental temperature is assumed to be 293 K (20°C). A surface with a temperature lower than the environmental temperature emits “cool” radiant exergy and a surface with a temperature higher than the environmental temperature emits “warm” radiant exergy (See Appendix B, formula B.1).

1.5 Exergy-entropy process of passive systems

Here let us describe the general characteristics of six passive systems from the viewpoint of exergy-entropy process (see (Shukuya, 1998) and (Shukuya, 2000)). As suggested above, rational passive (bio-climatic) design would be prerequisite to realize low-exergy systems for heating and cooling.

Daylighting: this is to consume solar exergy for indoor illumination. Exergy consumption occurs as solar exergy is absorbed by the interior surfaces of building envelopes. “Warm” exergy is produced as a result of solar exergy consumption for lighting; this may be consumed for space heating (Asada and Shukuya, 1999). The entropy generated in the course of solar exergy consumption for lighting must be discarded into the atmosphere by ventilation cooling or mechanical cooling, hopefully by a low-exergy system for cooling.

Passive heating: this is to control the rate of solar exergy consumption during daytime and nighttime by forming the built-environmental space with the appropriate materials that have low thermal conductivity and high thermal-exergy storage capacity. It is also to consume, during nighttime, the thermal exergy produced during daytime. Most of the entropy generated is discarded spontaneously through the building envelopes into the atmosphere (Shukuya and Komuro, 1996).

- Shading: this is to let the excess solar exergy, namely the rest of exergy necessary for daylighting, be consumed before it enters the built environment. It is also to reduce the entropy generated within the built environment so that mechanical equipment for cooling is required to consume less exergy to remove the entropy generated within the built environment. Exterior shading devices are very much attractive in this regard, since the entropy generated at the devices is effectively discarded into the atmosphere by convection (Asada and Shukuya, 1999).
- Ventilation cooling: this is to consume kinetic exergy of atmospheric air, which is produced by the exergy-entropy process of the global environmental system described later (Shukuya and Komuro, 1996), for removing the entropy generated within the built environment, such as the entropy discarded from the body surface of the occupants and that from the lighting fixtures, electric appliances and others, into the near-ground atmosphere.
- (Free cooling)
- Water spraying: this is to consume the “wet” exergy contained by liquid water, which is very large compared to thermal exergy, namely “warm” or “cool” exergy, to decrease the “warm” exergy produced by solar exergy consumption and possibly to produce “cool” exergy (See (Nishikawa and Shukuya, 1999), and (Saito and Shukuya, 1998)). Roof spraying and *uchimizu*, which is to scatter rainwater on the road surface, are also due to this process. The consumption of “wet” exergy to produce “cool” exergy or to decrease “warm” exergy play a very important role in photosynthetic system of leaves (Saito and Shukuya, 1998) and the temperature-regulating system of human body (Saito and Shukuya, 2000).
- Composting: this is to let micro organisms consume actively a large amount of exergy contained by garbage and hence turn it into fertilizer. The “warm” exergy produced as a result of micro-organisms consuming chemical exergy can be rationally consumed for maintaining the temperature inside the container at a desired level. This is realized by making the walls of a container thermally well insulated (Takahashi and Shukuya, 1998). The entropy generated in the process of composting is discarded into the surrounding of the container and finally into the near-ground atmosphere.

With the view of passive (bio-climatic) design as exergy-entropy process, passive design is to design a route in which the exergy available from our immediate surroundings is rationally consumed and the generated entropy is rationally discarded into the atmosphere. Again, low-exergy systems for heating and cooling would be such systems consistent with passive design described above.

1.6 The global environmental system

Our near-ground atmosphere receives all the entropy that is generated and discarded by all systems involving lighting, heating, and cooling of the built environment. This also applies to any living systems such as bacteria, plants, and animals, since the involved biological phenomena can be reduced to the combination of chemical and physical phenomena, although such reduction alone cannot give us an answer to why the biological phenomena are so complex or how living systems evolve.

Since the entropy contained by a substance is, as described in the previous section, a function of temperature and pressure, the near-ground atmospheric temperature must rise if the near-ground atmosphere continues to receive the entropy discarded from various systems. But, what is actually occurring in the nature is different; the average atmospheric temperature is almost constant from year to year. This is due to the fact that the atmosphere has an exergy-entropy process that works feeding on and consuming solar exergy, thereby producing the entropy, and finally disposing of the produced entropy into the Universe. We call this the global environmental system.

Fig. 4 shows schematically and numerically the exergy-entropy process of the global environmental system (Shukuya and Komuro, 1996). The earth receives not only the solar exergy of 220.7 W/m^2 but also the “cool” radiant exergy of 102.2 W/m^2 from the Universe. These exergies are all consumed sooner or later within the upper or lower atmosphere. The figures in the squares, 176.6 W/m^2 and 146.3 W/m^2 , show the exergy consumption in the upper and lower atmospheres respectively. The convective air current near the ground surface, a part of which can be used for ventilation, has 0.73 W/m^2 of kinetic exergy. The rain drops inside the clouds before falling down towards the ground surface have 1.25 W/m^2 of potential exergy; a part of this exergy may be consumed to produce electric power. These kinetic and potential exergies are produced by the solar and the “cool” radiant exergy consumption. The resultant generated entropy due to the exergy consumption is delivered first into the upper atmosphere by convection, evaporation and long-wave radiation and then into the Universe by long-wave radiation. The total amount of the entropy generation is the difference between the input and the output entropy flows across the upper boundary surface of the upper

atmosphere. A portion of the 102.2 W/m^2 of “cool” radiant exergy coming from the Universe enables the global environmental system to have the outgoing entropy flow of $1.239 \text{ W/(m}^2\cdot\text{K)}$. It should be recognized that the “cool” radiant exergy of 102.2 W/m^2 is vitally important in addition to the solar exergy, because it is the exergy that finally sweeps away all the generated entropy within the upper and lower atmospheres, which includes the entropy generated by bio-climatically designed building envelope systems and also low-exergy systems for heating and cooling.

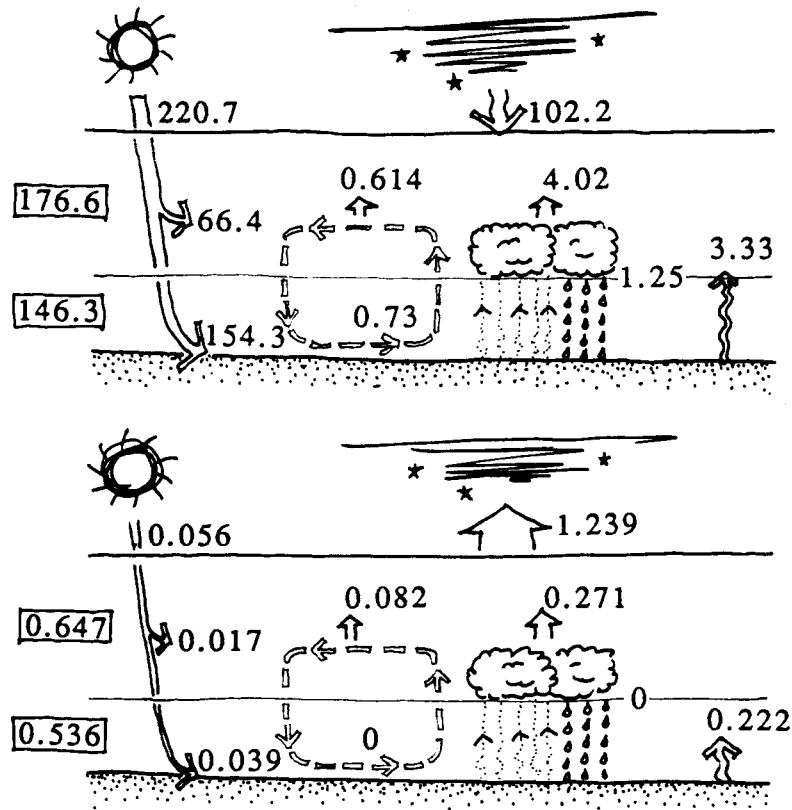


Fig. 4. Exergy-entropy process of the global environmental system. The drawing on the top is the exergy input, output, and consumption in W/m^2 . The other drawing on the bottom is the entropy input, output, and generation in $\text{W/(m}^2\cdot\text{K)}$. The amounts of the exergy consumption and the entropy generation are indicated by the figures in the squares.

1.7 Conclusion

A thermodynamic concept of exergy which explicitly indicates “what is consumed” was explained from the viewpoint of its application especially to describing building heating and cooling systems, together with an explanation of entropy, which explicitly indicates “what is disposed of”.

All working systems work as exergy-entropy process, in which exergy is supplied, a portion is consumed and thereby the resultant entropy is generated, and finally the generated entropy is discarded into the environment. The structure and function of the exergy balance equation were outlined, and the features of “warm” exergy and “cool” exergy were presented. The general characteristics of the exergy-entropy process of passive design were also described together with the global environmental system. What is suggested from the discussion here is that rational passive design is a prerequisite to realize low exergy systems for the heating and cooling of buildings.

2. Mathematical formulations

Abdelaziz Hammache

2.1 Introduction

Traditional methods of thermal system analysis are based on the first law of thermodynamics. These methods use an energy balance on the system to determine heat transfer between the system and its environment. The first law of thermodynamics introduces the concept of energy conservation, which states that energy entering a thermal system with fuel, electricity, flowing streams of matter, and so on is conserved and cannot be destroyed. In general, energy balances provide no information on the quality or grades of energy crossing the thermal system boundary and no information about internal losses. By contrast, the second law of thermodynamics introduces the useful concept of exergy in the analysis of thermal systems. Exergy is a measure of the quality or grade of energy and it can be destroyed in the thermal system. The second law states that part of the exergy entering a thermal system with fuel, electricity, flowing streams of matter, and so on is destroyed within the system due to irreversibilities. The second law of thermodynamics uses an exergy balance for the analysis and the design of thermal systems. The second part of this document describes the various forms of exergy and the mathematical formulations used to carry out the exergy balance. The different exergetic efficiency factors are also introduced and explained.

2.2 Exergy balance

One of the main uses of the exergy concept is in an exergy balance in the analysis of thermal systems.

The exergy balance is a statement of the law of degradation of energy. Degradation of energy is due to the irreversibilities of all real processes. Because open system analysis is much more relevant to the analysis of thermal plants or chemical systems than closed system analysis, the exergy balance of an open steady state system, as shown in Fig. 5, is presented. The exergy balance is stated around a control region delimited by specific boundaries.

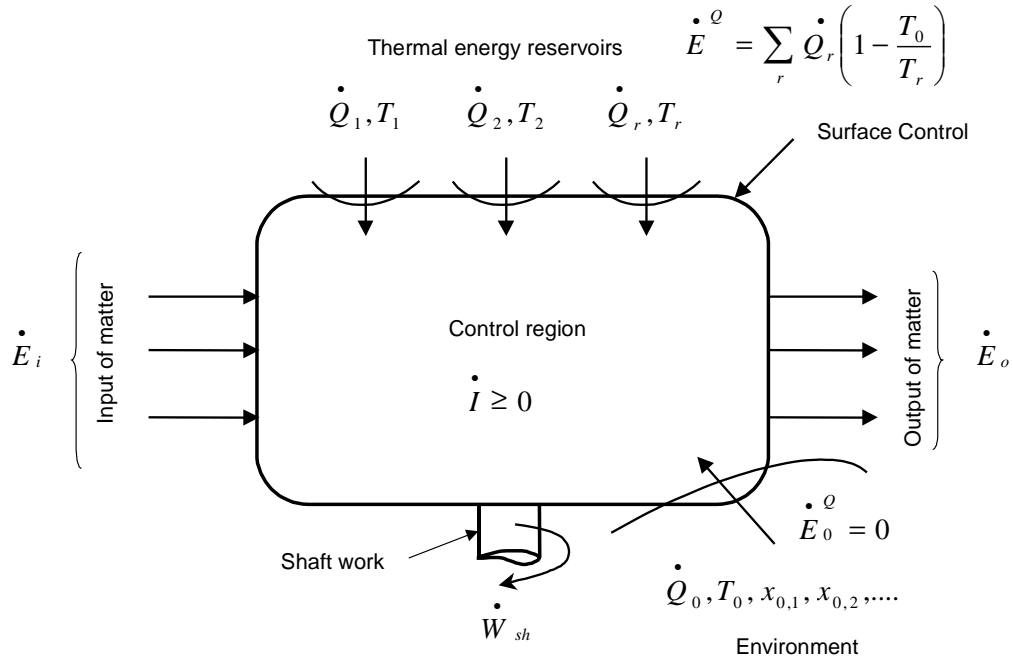


Fig. 5. Steady state process in an open control region.

$$\dot{E}_i + \dot{E}^Q = \dot{E}_o + \dot{W}_{sh} + \dot{I} \quad (2.1)$$

where:

$$\dot{E}_i = \sum_i \dot{m}_i e_i \quad (2.2)$$

$$\dot{E}_o = \sum_o \dot{m}_o e_o \quad (2.3)$$

$$\dot{E}^Q = \sum_r \dot{Q}_r \left(1 - \frac{T_0}{T_r} \right) \quad (2.4)$$

The expression for specific exergy is written as:

$$e = (h - h_0) - T_0 (s - s_0) + e_{ch} + \frac{C_0^2}{2} + gZ_0 \quad (2.5)$$

The exergy flow to the control region is always greater than that from the control region. The difference between the two, the rate of loss of exergy, is called the irreversibility

rate. The irreversibility rate is calculated from the Gouy-Stodola relation, which states that the irreversibility rate of a process is the product of the entropy generation rate for all systems participating in the process and the temperature of the environment.

$$\dot{I} = T_0 \dot{S}_{gen} = T_0 \left(\sum_o \dot{m}_o s_o - \sum_i \dot{m}_i s_i - \sum_r \frac{\dot{Q}_r}{T_r} \right) \quad (2.6)$$

The various exergy terms, which go into the exergy balance, are presented in Appendix C.

2.3 Definitions of exergetic efficiencies

Three definitions of exergetic efficiencies for steady state processes are found in the literature. These are the conventional or simple exergetic efficiency, the rational exergetic efficiency and the utilizable exergy coefficient.

2.3.1 Conventional exergetic efficiency

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

$$\dot{E}_{in} = \dot{E}_{out} + \dot{I} \quad (2.7)$$

Referring to Fig. 1:

$$\dot{E}_{in} = \dot{E}_i + \dot{E}^* \quad (2.8)$$

$$\dot{E}_{out} = \dot{E}_o + \dot{W}_{sh}^* \quad (2.9)$$

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

$$\eta = \frac{\dot{E}_{out}}{\dot{E}_{in}} \quad (2.10)$$

This is an unambiguous definition and can be used for all process plants and units. Unfortunately, it gives a good impression of the thermodynamic perfection of a system only when all the components of the incoming exergy flow are transformed to other components, e.g., in the case for power stations or for building heating and cooling systems. The traditional exergetic efficiency for power stations is expressed as:

$$\eta = \frac{\dot{W}_{elec}}{\dot{E}_{fuel}} \quad (2.11)$$

When all the components of the incoming exergy flows are not transformed to other component, the untransformed components give a false impression of the performance of the process plant or unit. For example, if we consider a chemical reactor with a zero reactive conversion factor, the input exergy rate will equal the output exergy rate and the traditional exergetic efficiency will equal 1. There are no irreversibilities in the reactor but it does not produce anything! In this case, the traditional exergetic efficiency gives a false impression of the thermodynamic performance of the reactor. To solve this problem other exergetic efficiencies have been proposed.

2.3.2 Rational exergetic efficiency

The rational exergetic efficiency is defined by Kotas (1985) as a ratio of the desired exergy output to the exergy used or consumed.

$$\psi = \frac{\dot{E}_{desired\ output}}{\dot{E}_{used}} = 1 - \frac{\dot{I}}{\dot{E}_{used}} \quad (2.12)$$

$\dot{E}_{desired\ output}$ is the sum of all exergy transfers from the system, which must be regarded as constituting the desired output, plus any by-product, which is produced by the system. The desired output is determined by examining the function of the system. \dot{E}_{used} is the required exergy consumed for the process to be performed.

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

As an example of formulation of rational efficiency consider the refrigeration plant evaporator shown in Fig. 6 (Kotas, 1985).

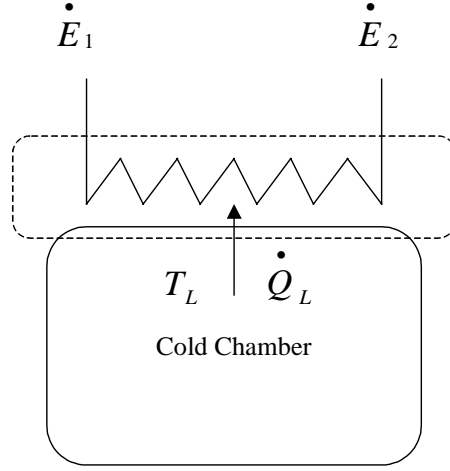


Fig. 6. Refrigeration plant evaporator.

The cold chamber may be considered as a thermal energy reservoir at a temperature $T_L < T_0$. Referring to Fig. 6, the exergy balance for the control region indicated by the dashed contour may be written:

$$\left(\dot{E}_1 - \dot{E}_2 \right) - \dot{I} = -\dot{Q}_L \left(1 - \frac{T_0}{T_L} \right) \quad (2.13)$$

The desired output is the increase in the exergy of the cold chamber, which since $T_L < T_0$, is associated with heat transfer from the cold chamber. Thus:

$$\dot{E}_{desired\ output} = -\dot{Q}_L \left(1 - \frac{T_0}{T_L} \right) \quad (2.14)$$

By incorporating Eqs. (13) and (14) into Eq. (12), the rational efficiency for this system is expressed as:

$$\psi = \frac{\dot{Q}_L \left(\frac{T_0}{T_L} - 1 \right)}{\dot{E}_1 - \dot{E}_2} = 1 - \frac{\dot{I}}{\dot{E}_1 - \dot{E}_2} \quad (2.15)$$

and the irreversibility rate is:

$$\dot{i} = T_0 \left[\left(\dot{S}_2 - \dot{S}_1 \right) - \frac{\dot{Q}_L}{T_L} \right] \quad (2.16)$$

2.3.3 Utilizable exergy coefficient

Brodyansky, Sorin and LeGoff (1994) introduced this form of exergetic efficiency, called utilizable exergy coefficient. This form of efficiency is an improvement on the traditional exergetic efficiency, because it subtracts the untransformed components from the incoming and outgoing streams. The following section introduces this concept.

To any material, heat and work stream can be associated an exergy content, which is completely defined by temperature, pressure and composition of the stream itself and of a reference state, which is normally the environment in which the system operates. It is, therefore, possible to compute the exergy content of all incoming and outgoing streams to and from a system and to establish an overall exergy balance over any system, as shown in Fig. 7. The total exergy input, E_{in} , of a real system is always higher than its exergy output, E'' , because a certain amount of exergy is irreversibly destroyed within the system. This exergy, generally referred to as the internal exergy losses or exergy destruction, I_{int} is directly linked to the thermodynamic irreversibilities in the system.

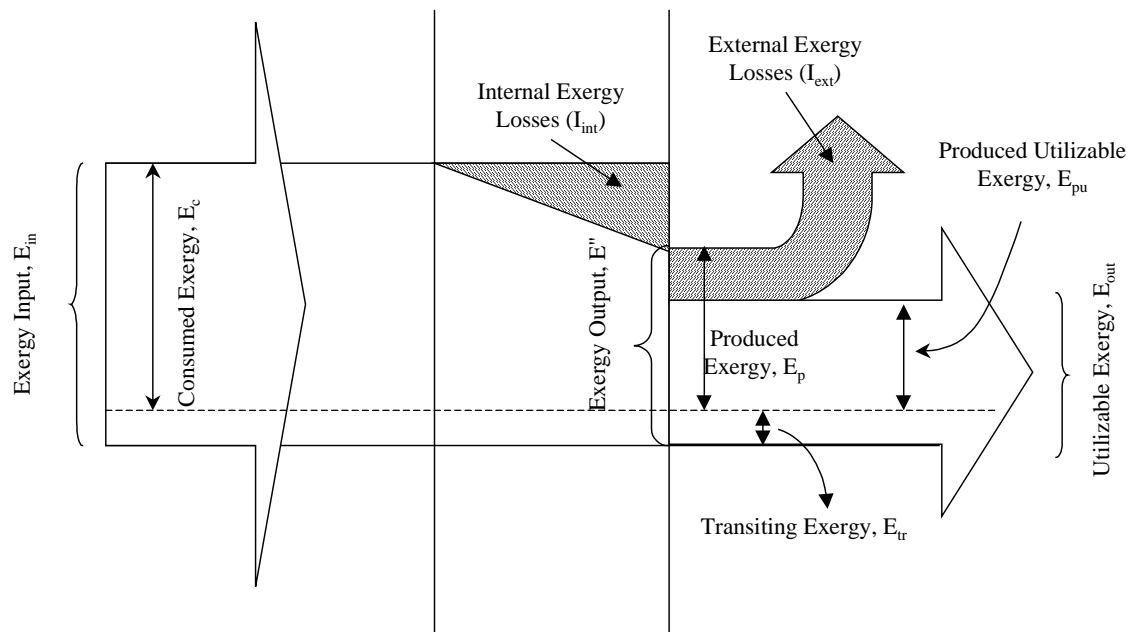


Fig. 7. Graphical presentation of overall exergy balance.

As illustrated in Fig. 7, part of the exergy output from the system may dissipate into the environment as heat losses, sewage waste or smokestack effluents, for example. This wasted exergy, no longer usable by subsequent processes, constitutes the external losses, I_{ext} . It is more appropriate, from the standpoint of downstream operations, to consider the exergy that remains utilizable, E_u , rather than the total output, E'' . Only part of the utilizable exergy is produced by the system through the physicochemical phenomena that take place within its boundaries. The rest of the exergy that leaves the system with the utilizable exergy stream is a part of the exergy input, which has simply gone through the system without undergoing any transformation. This fundamental fact was first recognized by Kostenko (1983), who gave the name transiting exergy, E_{tr} , to this fraction of the exergy supplied to a system. Typically in a chemical reactor, part (but not all, because of temperature and pressure changes) of the exergy associated with unreacted feed or inerts would constitute transiting exergy. Transiting exergy was further characterized and algorithms have been developed for computing it directly (Sorin and Brodyansky, 1985; Brodyansky et al. 1994). On the basis of these observations a new coefficient of thermodynamic efficiency, the utilizable exergy coefficient, η_u , has been defined (Sorin et al., 1998).

The exergetic efficiency with transiting exergy is defined as follows:

$$\eta_u = \frac{\dot{E}_{out} - \dot{E}_{tr}}{\dot{E}_{in} - \dot{E}_{tr}} = \frac{\dot{E}_{in} - \dot{I}_{int} - \dot{I}_{ext} - \dot{E}_{tr}}{\dot{E}_{in} - \dot{E}_{tr}} = \frac{\dot{E}_{pu}}{\dot{E}_c} \quad (2.17)$$

where \dot{E}_{tr} is the transiting exergy rate, \dot{E}_{pu} is the produced utilizable exergy rate and \dot{E}_c is the consumed exergy rate.

As has been demonstrated by Sorin and co-workers (1998), the decrease in the transiting exergy, \dot{E}_{tr} , improves the conversion performance of the system. The utilizable exergy coefficient decreases as \dot{I}_{int} , \dot{I}_{ext} and \dot{E}_{tr} decrease.

An example on the use of the various exergetic definitions is shown in Appendix D.

2.4 Air-conditioning applications

Air-conditioning applications are important and widely used in heating and cooling of buildings. This section presents the use of the concept of exergy in the assessment of air-conditioning applications. The concepts of physical exergy and chemical exergy

play an important role in assessing the true thermodynamic merit of air-conditioning applications.

The objective of most air-conditioning applications is to bring a humid air mixture to a state (temperature and composition) that differs from the conditions found in the atmospheric air.

The classical way of describing the thermodynamic properties of humid air is to view it as a perfect gas mixture of dry air (a) and water vapor (v) (Table 2). The ideal gas constants of these two components are the values corresponding to $T \cong 300 \text{ K}$ and the low-pressure limit (Bejan, 1988).

Table 2. Ideal gas constants of dry air (a) and water vapor (v).

Dry Air (a)	Water Vapor (v)
$R_a = 0.287 \text{ kJ}/(\text{kg}\cdot\text{K})$	$R_v = 0.4615 \text{ kJ}/(\text{kg}\cdot\text{K})$
$c_{p,a} = 1.003 \text{ kJ}/(\text{kg}\cdot\text{K})$	$c_{p,v} = 1.872 \text{ kJ}/(\text{kg}\cdot\text{K})$
$M_a = 28.97 \text{ kg}/\text{kmol}$	$M_v = 18.015 \text{ kg}/\text{kmol}$
$\bar{R} = 8.314 \text{ kJ}/(\text{kmol}\cdot\text{K})$	$\bar{R} = 8.314 \text{ kJ}/(\text{kmol}\cdot\text{K})$
$\bar{c}_{p,a} = 29.057 \text{ kJ}/(\text{kmol}\cdot\text{K})$	$\bar{c}_{p,v} = 33.724 \text{ kJ}/(\text{kmol}\cdot\text{K})$

The state of any humid air is specified by its temperature T , its pressure P , and one of the mole fraction x_a or x_v since $x_a + x_v = 1$.

The composition of humid air is described in the field of air conditioning by different ways:

The mass ratio called specific humidity or humidity ratio, ω , which represents the number of kilograms of water to 1 kg of dry air in the given mixture :

$$\omega = \frac{m_v}{m_a} \tag{2.18}$$

The mole fraction ratio, $\bar{\omega}$, which represents the number of moles of water corresponding to 1 mole of dry air in the given mixture :

$$\bar{\omega} = \frac{x_v}{x_a} \quad (2.19)$$

The relative humidity, ϕ , which represents the number moles of water in the actual mixture over the number of moles of water in the saturated mixture at temperature T :

$$\phi = \frac{x_v [T, P]}{x_{v, sat} [T, P_{sat}]} = \frac{P_v}{P_{sat}(T)} \quad (2.20)$$

The specific total flow exergy of humid air is deduced from the definition of the physical flow exergy applied to a mixture of ideal gases. It can be expressed differently depending on how the composition of humid air is described (Bejan, 1988):

The specific total flow exergy per mole of a humid air mixture is:

$$\bar{e}_t = \left(x_a \bar{c}_{p,a} + x_v \bar{c}_{p,v} \right) T_0 \left(\frac{T}{T_0} - 1 - \ln \frac{T}{T_0} \right) + \bar{R} T_0 \ln \frac{P}{P_0} + \bar{R} T_0 \left(x_a \ln \frac{x_a}{x_{0,a}} + x_v \ln \frac{x_v}{x_{0,v}} \right) \quad (2.21)$$

where, subscript 0 indicates ambient properties.

Two alternative versions of this equation are used for engineering calculations. The first alternative uses the mole ratios $\bar{\omega}$ and $\bar{\omega}_0$ to describe the composition of the actual and the ambient air mixtures:

$$\bar{e}_t = \frac{\bar{c}_{p,a} + \bar{\omega} \bar{c}_{p,v}}{1 + \bar{\omega}} T_0 \left(\frac{T}{T_0} - 1 - \ln \frac{T}{T_0} \right) + \bar{R} T_0 \ln \frac{P}{P_0} + \bar{R} T_0 \left((1 + \bar{\omega}) \ln \frac{1 + \bar{\omega}_0}{1 + \bar{\omega}} + \frac{\bar{\omega}}{1 + \bar{\omega}} \ln \frac{\bar{\omega}}{\bar{\omega}_0} \right) \quad (2.22)$$

The second alternative reports the specific total flow exergy per kilogram of dry air:

$$\bar{e}_t = \frac{\bar{c}_{p,a} + \bar{\omega} \bar{c}_{p,v}}{1 + \bar{\omega}} T_0 \left(\frac{T}{T_0} - 1 - \ln \frac{T}{T_0} \right) + \bar{R} T_0 \ln \frac{P}{P_0} + \bar{R} T_0 \left((1 + \bar{\omega}) \ln \frac{1 + \bar{\omega}_0}{1 + \bar{\omega}} + \frac{\bar{\omega}}{1 + \bar{\omega}} \ln \frac{\bar{\omega}}{\bar{\omega}_0} \right) \quad (2.23)$$

(2.24)

$$e_t = (c_{p,a} + \omega c_{p,v})T_0 \left(\frac{T}{T_0} - 1 - \ln \frac{T}{T_0} \right) + (1 + \bar{\omega})R_a T_0 \ln \frac{P}{P_0} + R_a T_0 \left((1 + \bar{\omega}) \ln \frac{1 + \bar{\omega}_0}{1 + \bar{\omega}} + \bar{\omega} \ln \frac{\bar{\omega}}{\bar{\omega}_0} \right)$$

The specific total flow exergy of dry air is deduced by setting ω and $\bar{\omega}$ to zero.

$$e_{t,a} = c_{p,a} T_0 \left(\frac{T}{T_0} - 1 - \ln \frac{T}{T_0} \right) + R_a T_0 \ln \frac{P}{P_0} + R_a T_0 \ln(1 + \bar{\omega}_0) \quad (2.25)$$

The specific total flow exergy of liquid water is also required for the case of exergy analysis of air-conditioning applications. The specific total flow exergy per kilogram of liquid water, w , yields:

$$e_{t,w} = h_w(T, P) - h_0(T_0, P_{0,w}) - T_0 (s_w(T, P) - s_0(T_0, P_{0,w})) \quad (2.26)$$

where the partial pressure of water vapor in atmospheric air is given by:

$$P_{0,w} = x_{0,v} P_0 \quad (2.27)$$

The total flow exergy of liquid water can be approximated by using the properties of respective neighboring states on the two-phase dome of the Mollier chart (Bejan, 1988):

$$e_{t,w} \cong h_f(T) - h_g(T_0) - T_0 (s_f(T) - s_g(T_0)) + [P - P_{sat}(T)] v_f(T) - R_v T_0 \ln \phi_0 \quad (2.28)$$

An Evaporative Cooling Example (Bejan, 1988).

The evaporative cooling system is shown in Fig. 8.

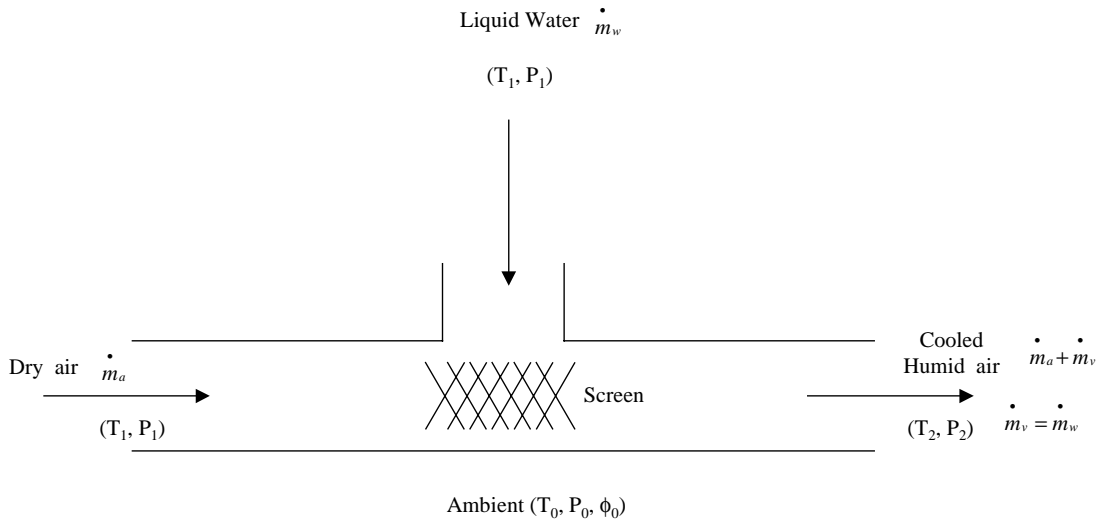


Fig. 8. Adiabatic evaporative cooling process.

Question: How much water is needed in order to lower the temperature of the outgoing mixture to a prescribed level T_2 ?

First law for the control volume:

$$\dot{m}_a h_a(T_1) + \dot{m}_w h_w(T_1, P_1) = \dot{m}_a h_a(T_2) + \dot{m}_v h_v(T_2, P_{v2}) \quad (2.29)$$

$$h_a(T_1) + \frac{\dot{m}_w}{\dot{m}_a} h_w(T_1, P_1) = h_a(T_2) + \frac{\dot{m}_v}{\dot{m}_a} h_v(T_2, P_{v2}) \quad (2.30)$$

$$h_a(T_1) + \omega h_w(T_1, P_1) = h_a(T_2) + \omega h_v(T_2, P_{v2}) \quad (2.31)$$

By using approximate relations, we obtain an explicit relationship for calculating the needed humidity ratio when the exit temperature is specified.

$$h_a(T_1) + \omega h_f(T_1) \cong h_a(T_2) + \omega h_g(T_2) \quad (2.32)$$

$$\omega \cong \frac{c_{p,a}(T_1 - T_2)}{h_g(T_2) - h_f(T_1)} \quad (2.33)$$

Question: How much exergy is being destroyed during the evaporative cooling process?

The exergy balance for the control volume is:

$$\dot{m}_a e_{t,a} + \dot{m}_w e_{t,w} = \dot{m}_a e_t + T_0 \dot{S}_{gen} \quad (2.34)$$

where e_t is the specific total flow exergy of the humid air expressed per kg of dry air.

The rate of exergy destruction per kg of dry air is, therefore,

$$\frac{T_0 \dot{S}_{gen}}{\dot{m}_a} = e_{t,a} + \omega e_{t,w} - e_t \quad (2.35)$$

The exergy efficiency of the evaporative cooler can be defined as the ratio:

$$\varepsilon = \frac{e_t}{e_{t,a} + \omega e_{t,w}} \quad (2.36)$$

Numerical Application:

$$T_0 = 25^\circ\text{C} = 298.15 \text{ K}$$

$$P_0 = 1 \text{ atm}$$

$$\phi_0 = 0.6$$

$$P_{\text{sat}}(T_0) = 0.03166 \text{ bar} = 0.0313 \text{ atm}$$

$$\omega_0 = \frac{0.622}{\frac{P_0}{\phi_0 P_{\text{sat}}(T_0)} - 1} = \frac{0.622}{\frac{1}{0.6 \times 0.0313} - 1} = 0.0119 \quad (2.37)$$

$$\bar{\omega}_0 = \frac{M_a}{M_v} \omega_0 = \frac{28.97}{18.015} (0.0119) = 0.0191 \quad (2.38)$$

Assume that $P_1 = P_2 = P_0$ and $T_1 = T_0$

Assume that the function of the evaporative cooling process is to lower the temperature to $T_2 = 15^\circ\text{C} = 288.15\text{ K}$

From steam tables:

$$h_g(288.15\text{ K}) = 2529.1\text{ kJ/kg}$$

$$h_f(298.15\text{ K}) = 104.77\text{ kJ/kg}$$

$$\omega \cong \frac{1.003(298.15 - 288.15)}{2529.15 - 104.77} = 0.00414$$

$$\bar{\omega} \cong \frac{28.97}{18.015}(0.00414) = 0.00665$$

$$e_{t,a} = c_{p,a}T_0 \left(\frac{T_1}{T_0} - 1 - \ln \frac{T_1}{T_0} \right) + R_a T_0 \ln \frac{P_1}{P_0} + R_a T_0 \ln(1 + \bar{\omega}_0) \quad (2.39)$$

Since $P_1 = P_0$ and $T_1 = T_0$

$$e_{t,a} = R_a T_0 \ln(1 + \bar{\omega}_0) = 0.287 \times 298.15 \ln(1 + 0.0191) = 1.620\text{ kJ/kg} \quad (2.40)$$

$$e_{t,w} \cong h_f(T_1) - h_g(T_0) - T_0(s_f(T_1) - s_g(T_0)) + [P_1 - P_{sat}(T_1)]v_f(T_1) - R_v T_0 \ln \phi_0 \quad (2.41)$$

Since $P_1 = P_0$ and $T_1 = T_0$

$$h_f(T_1) - h_g(T_0) - T_0(s_f(T_1) - s_g(T_0)) = h_{fg}(T_0) - T_0 s_{fg}(T_0) = 0 \quad (2.42)$$

$$[P_1 - P_{sat}(T_1)]v_f(T_1) \cong 0 \quad (2.43)$$

then

$$e_{t,w} \cong -R_v T_0 \ln \phi_0 = -0.461 \times 298.15 \times \ln(0.6) = 70.21\text{ kJ/kg} \quad (2.44)$$

$$\omega e_{t,w} = 0.00414 \times 70.21 = 0.290 \text{ kJ/kg} \quad (2.45)$$

$$e_t = (c_{p,a} + \omega c_{p,v}) T_0 \left(\frac{T_2}{T_0} - 1 - \ln \frac{T_2}{T_0} \right) + \quad (2.46)$$

$$(1 + \bar{\omega}) R_a T_0 \ln \frac{P_2}{P_0} + R_a T_0 \left((1 + \bar{\omega}) \ln \frac{1 + \bar{\omega}_0}{1 + \bar{\omega}} + \bar{\omega} \ln \frac{\bar{\omega}}{\bar{\omega}_0} \right)$$

Since $P_2 = P_0$

$$e_t = (c_{p,a} + \omega c_{p,v}) T_0 \left(\frac{T_2}{T_0} - 1 - \ln \frac{T_2}{T_0} \right) + R_a T_0 \left((1 + \bar{\omega}) \ln \frac{1 + \bar{\omega}_0}{1 + \bar{\omega}} + \bar{\omega} \ln \frac{\bar{\omega}}{\bar{\omega}_0} \right) \quad (2.47)$$

$$e_t = (1.003 + 0.00414 \times 1.872) \times 298.15 \left(\frac{288.15}{298.15} - 1 - \ln \frac{288.15}{298.15} \right) + \quad (2.48)$$

$$0.287 \times 298.15 \left((1 + 0.00665) \ln \frac{1 + 0.0191}{1 + 0.00665} + 0.00666 \ln \frac{0.00665}{0.0191} \right) = 0.632 \text{ kJ/kg}$$

Exergy efficiency:

$$\varepsilon = \frac{e_t}{e_{t,a} + \omega e_{t,w}} = \frac{0.632}{1.620 + 0.290} = 0.33 \quad (2.49)$$

Which means that the evaporating cooling process due to thermodynamic irreversibility destroys two thirds of the exergy brought to the control volume.

One important observation that must be dealt with is that the choice of the ambient conditions affects quite strongly the numerical results of the exergy analysis. The choice of $P_0 = 1 \text{ atm}$ is quite obvious, however there is a lack of a convention for the selection of T_0 and ϕ_0 . Fig. 9 shows how the numerical results of the exergy efficiency are affected by the selection of T_0 and ϕ_0 .

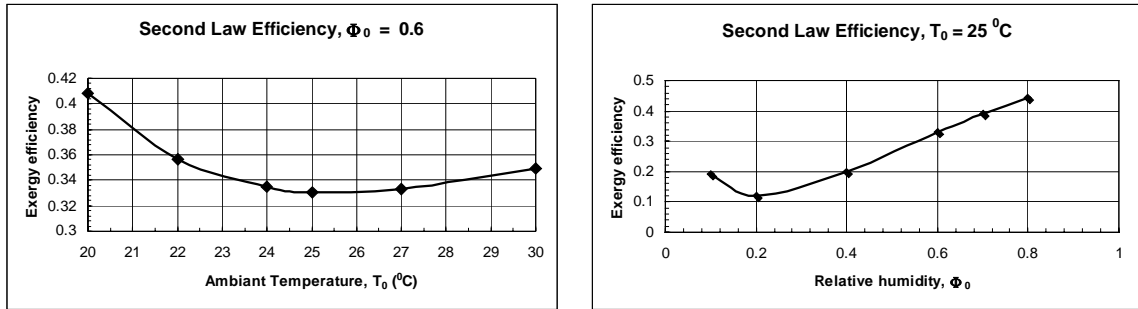


Fig. 9. Exergy efficiency of the adiabatic evaporative cooling process as a function of ϕ_0 and T_0 .

2.5 Conclusion

The exergy analysis of a thermal energy system has been presented through the exergy balance of an open system. The mathematical formulations of the various forms of exergy and the exergy efficiency factors were presented. Few examples were also presented to illustrate the use of the mathematical formulation. In the context of space heating and cooling in buildings, it is obvious that the use of either a conventional or a rational exergetic efficiency is sufficient to compare between different heating and cooling systems since no chemical reactions are involved in heating and cooling applications.

3. Space heating example

Masanori Shukuya and Abdelaziz Hammache

3.1 An example of heating exergy calculation

Let us compare three numerical examples of exergy consumption during the whole process of space heating from the power plant, through the boiler to the building envelope in the steady state as shown in Fig. 10 (Shukuya, 1994). Case 1 assumes that the thermal insulation of the building envelope system is poor; that is, single window glazing and an exterior wall with only a thin insulation board, and a boiler with a moderate thermal efficiency. Case 2 meanwhile assumes that the thermal insulation of the building envelope is improved by a combination of double window glazing and an exterior wall with improved insulation, while the boiler efficiency remains unchanged. Case 3 assumes in addition that the boiler efficiency is improved to near its limit. Table 3 summarizes the assumptions for calculation in three Cases.

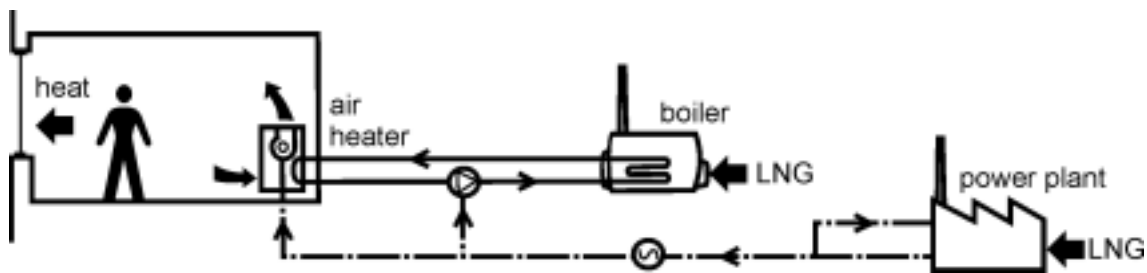


Fig. 10. A space heating system assumed for example calculation of exergy consumption.

Fig. 11 shows respective three series of exergy input, exergy consumption, and exergy output from the boiler, to the water-to-air heat exchanger, to the room air, and finally to the building envelope in three Cases.

Exergy consumption within the boiler system is the largest among the sub-systems. Consuming a lot of exergy is unavoidable when extracting thermal exergy by a combustion process from the chemical exergy contained in LNG. Because of this, one may consider that the improvement of boiler efficiency is essential. The dashed line indicated below Case 1 shows the result of the improvement of boiler efficiency from 0.8 to 0.95 in Case 1. The decrease of exergy consumption is marginal. One may, then, consider that increasing the outlet water temperature of the boiler makes exergy output

from the boiler larger and hence the boiler more efficient. This, however, results in the consumption of more exergy within the water-to-air heat exchanger and also within the room air, in which the required temperature is 293 K (20°C). These facts imply that an extremely high boiler efficiency alone cannot necessarily make a significant contribution to reducing exergy consumption in a whole process of space heating.

Table 3. Assumptions for example calculation of exergy consumption.

Case	Heat loss coefficient of building envelope		Thermal efficiency of boiler
1	108.7 W/K	(3.0 W/m ² K)	80 %
2	57.1	(1.59)	80
3	57.1	(1.59)	95

Heat-loss-coefficient values in the brackets are those per unit floor area. 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 W/m²K for Case 1; 3.6 and 1.14 for Cases 2 and 3. The number of air changes due to infiltration is 0.8 h⁻¹ for Case 1; and 0.4 h⁻¹ for Cases 2 and 3. The room air temperature is ideally controlled and 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. The rates of electric power supplied to a fan and a pump are 30 W and 23 W in Case 1; 16 W and 12 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.

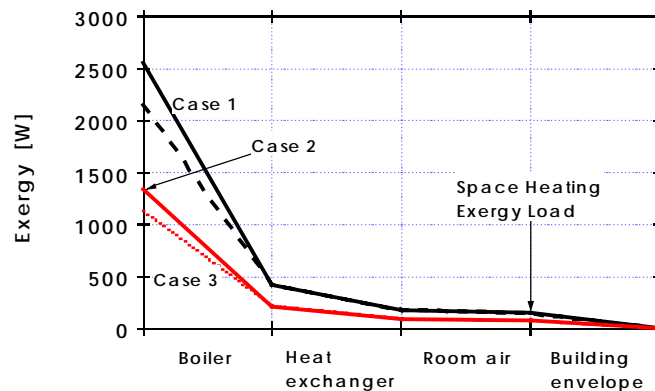


Fig. 11. A comparison of exergy consumption for four stages of the space heating systems. Exergy consumption is the difference in exergy between input and output; for example, in Case 1, 2554 W of exergy is supplied to the boiler and 420 W of “warm” exergy is produced and delivered to the heat exchanger by hot water circulation so that their difference, namely 2134 W (=2554-420), is consumed inside the boiler.

The heating exergy load, which is the exergy output from the room air and the exergy input to the building envelope is 148 W in Case 1 and 78 W in Case 2 and 3. It is only 6 to 7 % of the chemical exergy input to the boiler so that one may regard a measure reducing the heating exergy load as marginal. But, as can be seen from the difference in the whole exergy consumption profile between Case 1 and Case 2, it is more beneficial to reduce the heating exergy load by installing thermally well-insulated glazing and exterior walls than to develop a boiler with an extremely-high thermal efficiency, in order to decrease the rate of total exergy consumption. The reduction in exergy consumption of the boiler sub-system indicated by the difference between Case 2 and Case 3 due to the improvement in boiler efficiency turns essentially meaningful together with the improvement of building-envelope thermal insulation.

Those interested in numerical calculation of the example explained above are encouraged to consult Appendix E, which describes the detailed calculation procedure to obtain Fig. 2.

3.2 Conclusion

A few examples of exergy calculations were presented. What is suggested from the discussion here is that rational passive design is a prerequisite to realize low exergy systems for the heating and cooling of buildings.

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Appendix A: Thermal exergy contained by a volume of room air

$$X_{ra} = c_{air} m_{ra} \left\{ (T_r - T_o) - T_o \ln \frac{T_r}{T_o} \right\} \quad (\text{A.1})$$

where X_{ra} is thermal exergy contained by a volume of room air [kJ], c_{air} is specific heat of air [kJ/(kg·K)], m_{ra} is mass of room air [kg], T_r is room air temperature in the Kelvin scale, and T_o is outdoor air temperature in the Kelvin scale, which is the environmental temperature for room space. The value of c_{air} is assumed to be 1.005 kJ/(kg·K); m_{ra} to be 97.2 kg (= 1.2 kg/m³ x 6m x 5m x 2.7m); and T_o to be 288 K (= 15°C).

Appendix B: Radiant thermal exergy emitted by a wall surface

$$x_{rad} = \varepsilon \sigma \left\{ (T_s^4 - T_o^4) - \frac{4}{3} T_o (T_s^3 - T_o^3) \right\} \quad (\text{B.1})$$

where x_{rad} is radiant thermal exergy emitted by a wall surface of unit area [W/m^2], ε is emittance of the surface, σ is Stephan-Boltzmann constant [$\text{W}/(\text{m}^2 \cdot \text{K})$] (5.67×10^{-8}), T_s is surface temperature in the Kelvin scale, and T_o is outdoor air temperature in the Kelvin scale, which is the environmental temperature for room space. The value of T_o is assumed to be 293 K (20°C) and ε to be unity, namely black surface.

Appendix C: Forms of exergy

Exergy related to work transfer

Because exergy is defined as the maximum work potential, the work transfer rate, \dot{W} , is equivalent to the exergy transfer rate, \dot{E}_w , in every respect.

$$\dot{E}_w = \dot{W} \quad (\text{C.1})$$

Exergy related to heat transfer

Assuming a uniform temperature distribution in a thermal energy reservoir, the exergy transfer rate, \dot{E} , connected with the heat transfer rate, \dot{Q} , can be calculated by the following formula:

$$\dot{E} = \dot{Q} \times \tau \quad (\text{C.2})$$

where $\tau = 1 - \frac{T_0}{T}$ is a non-dimensional exergetic temperature, T_0 is the ambient temperature, which is set to 298 K (Szargut et al, 1988) and T is the heat source temperature.

Exergy related to a stream flow in steady state

Exergy transfer rate associated with material streams can be calculated with the following formula:

$$\dot{E} = \dot{E}_k + \dot{E}_p + \dot{E}_{ph} + \dot{E}_{ch} \quad (\text{C.3})$$

where:

$\dot{E}_k = m \left(\frac{C_0^2}{2} \right)$ represents the kinetic exergy rate, where C_0 is the speed of the stream flow relative to the earth surface.

$\dot{E}_p = \dot{m} g Z_0$ represents the potential exergy rate, where g is the earth gravity and Z_0 the stream altitude above the sea level.

\dot{E}_{ph} represents the thermomechanical exergy based on the temperature and the pressure of the stream.

\dot{E}_{ch} represents the chemical exergy based on the chemical potentials of the components in the stream.

The specific exergy is written as:

$$e = e_k + e_p + e_{ph} + e_{ch} \quad (\text{C.4})$$

where $e = \frac{\dot{E}}{\dot{m}}$ and \dot{m} is the mass flowrate of the stream.

Physical exergy

Physical exergy, known also as thermomechanical exergy, is the work obtainable by taking the substance through reversible process from its initial state (T, P) to the state of the environment (T₀, P₀). The specific physical exergy is written as:

$$e_{ph} = (h - h_0) - T_0(s - s_0) \quad (\text{C.5})$$

For a perfect gas with a constant c_p :

$$e_{ph} = c_p(T - T_0) - T_0 \left(c_p \ln \frac{T}{T_0} - R \ln \frac{P}{P_0} \right) \quad (\text{C.6})$$

For solids and liquids when assuming a constant specific heat c :

$$e_{ph} = c \left[(T - T_0) - T_0 \ln \left(\frac{T}{T_0} \right) \right] - v_m (P - P_0) \quad (\text{C.7})$$

Where v_m is the specific volume, determined at temperature T₀.

Chemical exergy

Chemical exergy is equal to the maximum amount of work obtainable when the substance under consideration is brought from the environmental state (T_0, P_0) to the dead state (T_0, P_0, μ_{0i}) by processes involving heat transfer and exchange of substances only with the environment. The specific chemical exergy e_{ch} at P_0 can be calculated by bringing the pure component in chemical equilibrium with the environment.

For pure reference components, which also exist in the environment, the chemical exergy consist of the exergy, which can be obtained by diffusing the components to their reference concentration P_{00} . The specific molar chemical exergy of a reference component i present in the environment at partial pressure $P_{00,i}$ is:

$$e_{0i} = RT_0 \ln \frac{P_0}{P_{00,i}} \quad (\text{C.8})$$

When a substance does not exist in the reference environment, it must first react to reference substances in order to get in equilibrium with the environment. The reaction exergy at reference conditions equals the standard Gibbs energy change. So the overall specific chemical exergy term becomes:

$$e_{ch,i} = \sum_j \nu_j e_{0j} - \Delta_r g_i \quad (\text{C.9})$$

The chemical exergy of a gaseous mixture or a mixture of ideal liquids is given by:

$$e_{ch} = \sum_i x_i [e_{0i} + RT_0 \ln(x_i)] \quad (\text{C.10})$$

x_i represents the molar fraction of component i and e_{0i} its standard chemical exergy.

The chemical exergy of real solutions can be computed from:

$$e_{ch} = \sum_i x_i [e_{0i} + RT_0 \ln(\gamma_i x_i)] \quad (\text{C.11})$$

where γ_i if the activity coefficient of the component i .

The chemical exergies of gaseous fuels are computed from the stoichiometric combustion chemical reactions. The standard chemical exergies of various fuels are published in the literature (e.g. Kotas (1985), Bejan (1988)).

For many fuels, the chemical exergy can be estimated on the basis of the Net Combustion Value (NCV). The relation between the NCV and the chemical exergy is:

$$e_{ch} = \phi \bullet NCV \quad (C.12)$$

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

Appendix D: Example of exergetic efficiencies

Air separation example

The application of the three different forms of efficiencies will be shown on an air distillation column (Cornelissen, 1997), which is shown in Fig. D.1.

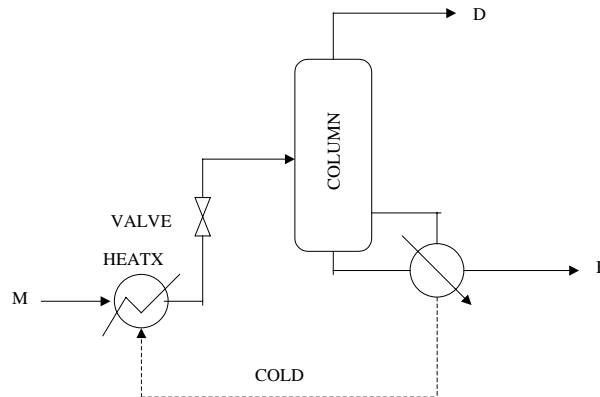


Fig. D.1. Air distillation column.

The incoming mixture M is first cooled by the reboiler and cooled further by the throttling process. The incoming and outgoing flows are vapors and assumed to be perfect gases. The properties and exergy of the streams are shown in Table D.1.

Table D.1. Table of properties and exergy of the streams.

Stream	T (K)	P (MPa)	Flow rate (kmol/h)	Composition (mole %)		Chemical exergy (kW)	Physical exergy (kW)	Total exergy (kW)
				O ₂	N ₂			
M	139	4	7	21	79	0	23.83	23.83
B	90	0.1	1	99	1	1.03	1.23	2.26
D	79	0.1	6	10	90	0.18	8.6	8.78

The column is assumed to operate adiabatically. The composition of air in the environment has been assumed to be 0.21 O₂ and 0.79 N₂ fractions by mole and the activity coefficients are taken all to be 1. The environmental temperature T₀ and pressure P₀ are assumed to be 298 K and 0.10 MPa, respectively.

Conventional exergetic efficiency

$$\eta = \frac{\dot{E}_{out}}{\dot{E}_{in}} = \frac{\dot{E}_B + \dot{E}_D}{\dot{E}_M} = \frac{2.26 + 8.78}{23.83} = 0.463 \quad (\text{D.1})$$

Rational exergetic efficiency

Depending on the desired output two cases have been discussed.

In case (i), the desired output is considered to be the increase in chemical exergy. Thus:

$$\psi = \frac{\dot{E}^{desired-output}}{\dot{E}_{used}} = \frac{\dot{E}_B^{ch} + \dot{E}_D^{ch} - \dot{E}_M^{ch}}{\dot{E}_M^{ph} - \left[\dot{E}_B^{ph} + \dot{E}_D^{ph} \right]} = \frac{1.03 + 0.18}{23.83 - (1.23 + 8.60)} = 0.0864 \quad (\text{D.2})$$

In case (ii), the desired output is considered to be the increase in both the chemical exergy and the thermal component of exergy.

$$\psi = \frac{\dot{E}^{desired-output}}{\dot{E}_{used}} = \frac{\left(\dot{E}_B^{ch} + \dot{E}_D^{ch} - \dot{E}_M^{ch} \right) + \left(\dot{E}_B^{\Delta T} + \dot{E}_D^{\Delta T} - \dot{E}_M^{\Delta T} \right)}{\dot{E}_M^{\Delta P} - \left[\dot{E}_B^{\Delta P} + \dot{E}_D^{\Delta P} \right]} = 0.271 \quad (\text{D.3})$$

The thermal and mechanical exergy components of streams M, B and D can be computed using the relations for perfect gases using constant c_p values.

Utilizable exergy coefficient

To compute the utilizable exergy coefficient, the transiting exergy in the different components must be calculated using the algorithms found in Sorin and Brodyansky, 1985 or Brodyansky et al. 1994.

The transiting chemical exergy in each part of the stream is:

$$\dot{E}_{tr(M-B)}^{ch} = \dot{n}_B (\min[e_M^{O_2}; e_B^{O_2}] + \min[e_M^{N_2}; e_B^{N_2}]) = \dot{n}_B (e_M^{O_2} + e_B^{N_2}) \quad (D.4)$$

$$\dot{E}_{tr(M-D)}^{ch} = \dot{n}_D (\min[e_M^{O_2}; e_D^{O_2}] + \min[e_M^{N_2}; e_D^{N_2}]) = \dot{n}_D (e_D^{O_2} + e_M^{N_2}) \quad (D.5)$$

The transiting physical components of exergy in each part of the stream is:

$$\dot{E}_{tr(M-B)}^{ph} = \dot{n}_B \min[e_M(p_B, T_M); e_B(p_B, T_M)] = \dot{n}_B e_B(p_B, T_M) \quad (D.6)$$

$$\dot{E}_{tr(M-D)}^{ph} = \dot{n}_D \min[e_M(p_D, T_M); e_D(p_D, T_M)] = \dot{n}_D e_D(p_D, T_M) \quad (D.7)$$

The chemical exergy change for the output streams B and D is given by:

$$\Delta \dot{E}_{out}^{ch} = \dot{n}_B (e_B^{O_2} + e_B^{N_2}) + \dot{n}_D (e_D^{O_2} + e_D^{N_2}) - \dot{n}_B (e_M^{O_2} + e_B^{N_2}) - \dot{n}_D (e_M^{N_2} + e_D^{O_2}) = \dot{n}_B e_B^{O_2} + \dot{n}_D e_D^{N_2} \quad (D.8)$$

$$\text{since : } e_M^{O_2} = e_M^{N_2} = 0$$

The physical exergy change for the output streams B and D is given by:

$$\Delta \dot{E}_{out}^{ph} = \dot{n}_B (e_B(p_B, T_B) - e_B(p_B, T_M)) + \dot{n}_D (e_D(p_D, T_D) - e_D(p_D, T_M)) \quad (D.9)$$

which is the increase in the thermal component of exergy of flow B at pressure p_B and in the thermal component of exergy of flow D at pressure p_D .

The chemical exergy change for the input stream M is given by:

$$\Delta \dot{E}_{in}^{ch} = \dot{n}_M (e_M^{O_2} + e_M^{N_2}) - \dot{n}_B (e_M^{O_2} + e_B^{N_2}) - \dot{n}_D (e_D^{O_2} + e_M^{N_2}) = -\dot{n}_B e_B^{N_2} - \dot{n}_D e_D^{O_2} \quad (D.10)$$

$$\text{since : } e_M^{O_2} = e_M^{N_2} = 0$$

The physical exergy change for the input stream M is given by:

$$\Delta \dot{E}_{in}^{ph} = \dot{n}_B (e_M(p_M, T_M) - e_B(p_B, T_M)) + \dot{n}_D (e_M(p_M, T_M) - e_D(p_D, T_M)) \quad (D.11)$$

which is the reduction of exergy flows in the system caused by the pressure drop at temperature T_M .

Using data from Table D1, the utilizable exergy coefficient, η_e , for the air separation example can now be calculated:

$$\eta_u = \frac{\dot{\Delta E}_{out}^{ch} + \dot{\Delta E}_{out}^{ph}}{\dot{\Delta E}_{in}^{ch} + \dot{\Delta E}_{in}^{ph}} = 0.369 \quad (D.12)$$

If we compare, the three exergetic efficiencies, It is seen that the traditional exergetic efficiency gives the highest efficiency compared to the other efficiencies. However, if the chemical exergy of the mixture was much higher as in crude oil separation, the conventional exergetic efficiency would be close to unity. Therefore, the conventional exergetic efficiency is of little use and may lead to false conclusions. The rational efficiency is easy to apply, however, to define the desired output exergy and the used exergy for a unit it is necessary to be clear about the function of the unit in the context of the plant. The utilizable exergy coefficient, on the other hand, is a straightforward method for an appropriate definition of efficiency, which gives an unambiguous answer.

Appendix E: Exergy calculation for space heating

First, we calculate the space heating energy demand using the window and wall areas, their corresponding U values, the number of air change, and the exterior-interior temperature difference shown in Table 3 in text p. 38. For Case 1, the space heating demand, q , turns out to be as follows.

$$q = \{(1.5 \times 6) \times 6.2 + (1.5 \times 6) \times 2.67 \times (20 - 0)\} + 1005 \times 1.2 \times 0.8 / 3600 \times (6 \times 6 \times 3) \times (20 - 0) \quad (\text{E.1})$$

$$= 2175 \text{ W}$$

In the above equation, 1005 is the specific heat of air in the unit of J/(kg · K), 1.2 is the density of air in kg/m³, and 0.8/3600 is the number of air change in s⁻¹. For Case 2, $q = 1143 \text{ W}$.

Space heating exergy load for the room space in question is given by

$$x_{rm} = \left(1 - \frac{T_o}{T_r}\right) q \quad (\text{E.2})$$

Since T_o is assumed to be 273 K (0°C) and T_r to be 293 K (20°C), x_{rm} turns out to be 148 W for Case 1 and 78 W for Case 2. In Fig. 11 in text p. 38, these two values are indicated as the input to “Building envelope”, which is namely the output from the “Room air”.

Next, we determine the mass flow rate of air transmitting through the water-to-air heat exchanger from the following equation.

$$m_a = \frac{q}{c_{air} (T_h - T_r)} \quad (\text{E.3})$$

where m_a is mass flow rate of air [kg/s], c_{air} is specific heat of air [J/(kg · K)] (=1005), T_h is outlet air temperature of the heat exchanger and T_r is room air temperature. Assuming that T_h is 303 K (30°C), the amount of m_a turns out to be 0.216 kg/s for Case 1 and 0.114 kg/s for Case 2.

The following step is to calculate the net “warm” exergy delivered by the air circulated through the water-to-air heat exchanger, which is the difference between the “warm” exergies calculated from eq. (A.1) substituting m_a for m_{ra} , T_h for T_r in the case of outlet air and T_r for T_r in the case of inlet air. The net “warm” exergy delivered are 182 W for Case 1 and 93 W for Case 2. These two values are the input thermal exergy to “Room air”.

The fan equipped with the water-to-air heat exchanger delivers these thermal exergies. The exergy requirement of the fan is assumed to be 30 W for Case 1 and 16 W for Case 2; this corresponds to a condition that fan efficiency is 0.6 and the pressure decrease between the air inlet and outlet of the heat exchanger is about 100 Pa. Then, the chemical exergy to produce the electricity for the fan becomes 81 W ($= 30 \times 0.94 / 0.35$) for Case 1 and 43 W ($= 16 \times 0.94 / 0.35$) for Case 2. Here, the number, 0.94, is the ratio of chemical exergy to the higher heating value of liquefied natural gas (LNG); and the number, 0.35, is thermal efficiency of the LNG-fired power plant. The chemical exergy of 81 W or 43 W of chemical exergy required by the fan are comparable to the amounts of space heating exergy load, 148 W or 78 W. This suggests that it is also very important to design a system having as short duct length and low mass flow rate of air as possible.

The next step is to do exergy calculation in terms of water circulation. Here, we assumed the inlet and outlet water temperatures of the water-to-air heat exchanger to be 343 K (70°C) and 333 K (60°C), respectively. The mass flow rate of water is determined as was done for the air flow rate using eq. (E.3) substituting the specific heat of water, 4186 J/(kg · K), for c_{air} and the inlet water temperature, 343 K (70°C) for T_h and the outlet water temperature, 333 K (60°C) for T_r . The values of mass flow rate obtained are 0.0519 kg/s for Case 1 and 0.0273 kg/s for Case 2. Then the warm exergy contained by water can be calculated from eq. (A.1) substituting 4.186 kJ/(kg · K) for c_{air} , the mass flow rate of water for m_{ra} , and the inlet water temperature of 343 K for T_r in case of inlet water and the outlet water temperature of 333 K for T_r in case of outlet water. The net “warm” exergy delivered by water circulation is the difference in thermal exergy contained by water between the inlet and the outlet; it becomes 419 W for Case 1 and 219 W for Case 2. In Fig. 11, these are indicated as the input exergy to “Heat exchanger”.

A pump circulates the water as a working fluid to deliver thermal exergy from the boiler to the water-to-air heat exchanger. The exergy requirement for the pump is assumed to be 23 W for Case 1 and 12 W for Case 2; this corresponds to a condition that pump efficiency is 0.7 and the pressure decrease within the pipe is about 300 kPa. Then, the chemical exergy to produce the electricity for the pump becomes 62 W ($= 23 \times 0.94 / 0.35$) for Case 1 and 32 W ($= 12 \times 0.94 / 0.35$) for Case 2. The chemical exergy for the pump is smaller than that for the fan, but it is still comparative to the space heating

exergy load especially of Case 2. It is also important to design a system having as short pipe length and low mass flow rate of water as possible.

The chemical exergy supplied to the boiler is determined by the following equation.

$$x_{LNG} = \frac{\alpha q}{\eta_b} \quad (\text{E.4})$$

where α is the ratio of chemical exergy to the higher heating value of LNG (= 0.94), and η_b is the thermal efficiency of the boiler (= 0.8). The value of x_{LNG} turns out to be 2554 W for Case 1 and 1342 W for Case 2. In Fig. 11, these are indicated as the input exergy to “Boiler”, respectively. If the boiler efficiency is improved from 0.8 to 0.95, x_{LNG} turns out to be 2151 W for Case 1; this is indicated, in Fig. 11, as the starting point of the dashed line below the solid line of Case 1. The value of x_{LNG} with the boiler efficiency of 0.95 together with the improved thermal insulation of building envelope becomes 1130 W; this is indicated as the exergy input to the boiler for Case 3.

The detailed excel files for the different cases are shown in Figures E1–E4 with the input data and the results obtained to generate Fig. 11.

Room	Length	Width	Height	Volume
	m	m	m	m ³
	6.00	6.00	3.00	108.00

Window	Length	Height	Area
	m	m	m ²
	6.00	1.50	9.00

Exterior Wall	Length	Height	Area
	m	m	m ²
	6.00	1.50	9.00

Air Conditions			
Room temperature	°C		20.00
Outside air temperature	°C		
Design ΔT	°C		20.00
Q-Factor			0.0682
Specific heat of air	J/(kg K)		1,005.00
Density of air	kg/m ³		1.20

Miscellaneous			
Power plant	Conventional Plant	η_{plant}	0.35
Energy source	Liquified Natural Gas	τ_{NLG}	0.94
Specific heat of liquid water	J/(kg K)		4,186.00

Case Assessed	Case 1		
U-Window		W/(m ² K)	6.20
U-Wall		W/(m ² K)	2.67
Air infiltration		h ⁻¹	0.80
Boiler efficiency			0.80
Pump power		W	23.00
Fan power		W	30.00
Space heating demand		W	2,175.48
Space heating exergy		W	148.42

Emission System	Heat Exchanger		
Water supply temperature T^{supply}		°C	70.00
Water return temperature T^{return}		°C	60.00
Output Air Transmitting Temperature $T_{\text{air}}^{\text{output}}$		°C	30.00
Mass flowrate of air through emission system		kg/s	0.22
Net exergy delivered to room		W	182.23
Mass flowrate of water through emission system		kg/s	0.0520
Net exergy delivered by water		W	418.05

Chemical exergy to produce fan power	W	80.57
Chemical exergy to produce pump power	W	61.77
Chemical exergy supplied to boiler	W	2,556.19

Fig. E1. Exergy calculations for case 1.

Room	Length	Width	Height	Volume
	m	m	m	m ³
	6.00	6.00	3.00	108.00

Window	Length	Height	Area
	m	m	m ²
	6.00	1.50	9.00

Exterior Wall	Length	Height	Area
	m	m	m ²
	6.00	1.50	9.00

Air Conditions			
Room temperature	°C		20.00
Outside air temperature	°C		
Design ΔT	°C		20.00
Q-Factor			0.0682
Specific heat of air	J/(kg K)		1,005.00
Density of air	kg/m ³		1.20

Miscellaneous			
Power plant	Conventional Plant	η_{plant}	0.35
Energy source	Liquified Natural Gas	τ_{NLG}	0.94
Specific heat of liquid water	J/(kg K)		4,186.00

Case Assessed	Case 1-b		
U-Window	W/(m ² K)		6.20
U-Wall	W/(m ² K)		2.67
Air infiltration	h ⁻¹		0.80
Boiler efficiency			0.95
Pump power	W		23.00
Fan power	W		30.00
Space heating demand	W		2,175.48
Space heating exergy	W		148.42

Emission System	Heat Exchanger		
Water supply temperature T^{supply}	°C		70.00
Water return temperature T^{return}	°C		60.00
Output Air Transmitting Temperature $T_{\text{air}}^{\text{output}}$	°C		30.00
Mass flowrate of air through emission system	kg/s		0.22
Net exergy delivered to room	W		182.23
Mass flowrate of water through emission system	kg/s		0.0520
Net exergy delivered by water	W		418.05

Chemical exergy to produce fan power	W	80.57
Chemical exergy to produce pump power	W	61.77
Chemical exergy supplied to boiler	W	2,152.58

Fig. E2. Exergy calculations for case 1b.

Room	Length	Width	Height	Volume
	m	m	m	m ³
	6.00	6.00	3.00	108.00

Window	Length	Height	Area
	m	m	m ²
	6.00	1.50	9.00

Exterior Wall	Length	Height	Area
	m	m	m ²
	6.00	1.50	9.00

Air Conditions			
Room temperature	°C		20.00
Outside air temperature	°C		
Design ΔT	°C		20.00
Q-Factor			0.0682
Specific heat of air	J/(kg K)		1,005.00
Density of air	kg/m ³		1.20

Miscellaneous			
Power plant	Conventional Plant	η_{plant}	0.35
Energy source	Liquified Natural Gas	τ_{NLG}	0.94
Specific heat of liquid water	J/(kg K)		4,186.00

Case Assessed	Case 2		
U-Window	W/(m ² K)		3.60
U-Wall	W/(m ² K)		1.14
Air infiltration	h ⁻¹		0.40
Boiler efficiency			0.80
Pump power	W		17.00
Fan power	W		16.00
Space heating demand	W		1,142.64
Space heating exergy	W		77.96

Emission System	Heat Exchanger		
Water supply temperature T^{supply}	°C		70.00
Water return temperature T^{return}	°C		60.00
Output Air Transmitting Temperature $T_{\text{air}}^{\text{output}}$	°C		30.00
Mass flowrate of air through emission system	kg/s		0.11
Net exergy delivered to room	W		95.71
Mass flowrate of water through emission system	kg/s		0.0273
Net exergy delivered by water	W		219.57

Chemical exergy to produce fan power	W	42.97
Chemical exergy to produce pump power	W	45.66
Chemical exergy supplied to boiler	W	1,342.60

Fig. E3. Exergy calculations for case 2.

Room	Length	Width	Height	Volume
	m	m	m	m ³
	6.00	6.00	3.00	108.00

Window	Length	Height	Area
	m	m	m ²
	6.00	1.50	9.00

Exterior Wall	Length	Height	Area
	m	m	m ²
	6.00	1.50	9.00

Air Conditions			
Room temperature	°C		20.00
Outside air temperature	°C		
Design ΔT	°C		20.00
Q-Factor			0.0682
Specific heat of air	J/(kg K)		1,005.00
Density of air	kg/m ³		1.20

Miscellaneous			
Power plant	Conventional Plant	η_{plant}	0.35
Energy source	Liquified Natural Gas	τ_{NLG}	0.94
Specific heat of liquid water	J/(kg K)		4,186.00

Case Assessed	Case 3		
U-Window	W/(m ² K)		3.60
U-Wall	W/(m ² K)		1.14
Air infiltration	h ⁻¹		0.40
Boiler efficiency			0.95
Pump power	W		17.00
Fan power	W		16.00
Space heating demand	W		1,142.64
Space heating exergy	W		77.96

Emission System	Heat Exchanger		
Water supply temperature T^{supply}	°C		70.00
Water return temperature T^{return}	°C		60.00
Output Air Transmitting Temperature $T_{\text{air}}^{\text{output}}$	°C		30.00
Mass flowrate of air through emission system	kg/s		0.11
Net exergy delivered to room	W		95.71
Mass flowrate of water through emission system	kg/s		0.0273
Net exergy delivered by water	W		219.57

Chemical exergy to produce fan power	W	42.97
Chemical exergy to produce pump power	W	45.66
Chemical exergy supplied to boiler	W	1,130.61

Fig. E4. Exergy calculations for case 3.

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Author(s) Shukuya, Masanori & Hammache, Abdelaziz			
Title Introduction to the Concept of Exergy – for a Better Understanding of Low-Temperature-Heating and High-Temperature-Cooling Systems			
Abstract <p>Chapter I describes the characteristics of a thermodynamic concept, exergy, in association with building heating and cooling systems. Exergy is the concept that explicitly indicates ‘what is consumed’. All systems, not only engineering systems but also biological systems including the human body, work feeding on exergy, consuming its portion and thereby generating the corresponding entropy and disposing of the generated entropy into their environment. The whole process is called ‘exergy-entropy process’. The features of ‘warm’ exergy and ‘cool’ exergy and also radiant exergy are outlined. General characteristics of exergy-entropy process of passive systems, which would be a prerequisite to realize low exergy systems, are discussed together with the exergy-entropy process of the global environmental system.</p> <p>Chapter II introduces the various forms of exergy and the mathematical formulations used to evaluate them. The exergy balance on an open steady state system, which is much more relevant to thermodynamic analysis of energy systems, is also described, as well as the different exergetic efficiency factors introduced in the thermodynamic analysis of energy systems. Next, an exergy analysis example is outlined through an air-conditioning application. Air-conditioning applications are widely used in heating and cooling of buildings.</p> <p>Chapter III introduces an example of exergy calculation for space heating systems. The issues to have a better understanding of low-exergy systems for heating and cooling are raised. It is suggested that a prerequisite for low exergy systems would be rational passive design of building envelope systems.</p>			
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