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Application of Exergy

Edited by Tolga Taner

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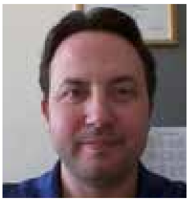
Meet the editor



Dr. Tolga Taner, PhD, is the head at the Department of Motor Vehicles and Transportation Technology of the Aksaray University since 2012. He received his BS degree in Mechanical Engineering in 1998 and MS degree in Mechanical Engineering in 2002 from the Pamukkale University, Denizli, Turkey, and PhD degree in Mechanical Engineering from the Gazi University of Engineering Faculty, Ankara, Turkey, in 2013. He was a part-time lecturer at the Department of Technical Programs, Middle East Technical, in 2003–2006. His current research interests include exergy, renewable energy, and PEM fuel cell. He has published over 20 scientific articles, 25 conference papers, and books. In addition, he has been a manager and researcher in many research projects.

Contents

Preface

This is a book on application of exergy with new technologies. In addition, the research of exergy application can overcome many critical problems that arise in an industrial energy. Furthermore, thermoeconomic analysis is also investigated for the cost of application and industry in the book.

This book will encourage both academic research and industrial application processes. From concept to publication, this book took approximately 1 year to complete. I would like to thank the publisher, IntechOpen, and in particular the Publishing Process Manager, Mr. Markus Mattila, for his support, help, and guidance. I would also like to thank all the chapter authors for their huge works. Finally, I would like to thank my family, my wife Öznur and my daughter Özüm, for their support.

I hope that this book will be helpful to people who read it. This book will make a scientific contribution to exergy workers, researchers, academics, PhD students, and other scientists in both the present and the future.

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Section 1

General Information About Exergy

Chapter 1

Introductory Chapter: Application of Exergy for All Energy Field and New Technologies

Tolga Taner

Additional information is available at the end of the chapter

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1. Introduction

The subject of exergy is one of the most important topics of thermodynamics. It also defines the efficiency of the second law of exergy thermodynamics. Exergy is an important concept in itself that explains the availability of energy. Although there are many studies related to exergy about more theoretical studies, there are not many studies in terms of application. This book is a study that will complement the application of exergy in terms of implementation and will also inspire future works. Application of exergy is available in many areas. Exergy applications need to be clarified in these research fields. This book contains exergy applications in these research fields.

This book poses application of exergy with current and new technologies. The main scope of this book is to emphasise exergy efficiency for all field of industry and other fields. In addition, the research of exergy application can overcome many critical problems for an industry and other field energies. Furthermore, thermoeconomic (exergoeconomic) analysis is also investigated for the cost of application and industry. This book impresses on the importance of an exergy with an overview of all of the energy systems. Energy and exergy efficiencies within thermodynamic laws are carried out for every technology. This book will both encourage the research of the academic community to be this direction and will lead to the more efficient use of application, industrial processes and new technologies.

Energy and exergy efficiency is very important as a result of thermodynamic calculations thanks to the first and second laws of thermodynamics. In many engineering and science areas, it is necessary to achieve the maximum energy saving and minimal loss of exergy in many areas. Exergy applications should be used to save energy in all industrial processes of engineering information. Because, the laws of thermodynamics and heat transfer principle are important for energy and exergy applications. Understanding and applying insufficient energy and exergy analyzes in many engineering applications leads to significant energy losses for factories. For the cause, loss of exergy is required to be minimised. This book aims to ensure a comprehensive overview of the application of exergy by finding solutions to these problems.

2. A brief of exergy and thermoeconomic analyses

Exergy is given according to the second law of thermodynamics, which defines exergy as the availability of energy [1]. The following explanations can be found in the literature regarding exergy: exergy is defined as maximum work and ensures that the system is balanced with the environment. In the exergy analysis, the final state is considered as the system's dead state. In this case, kinetic and potential energies are zero. Various enthalpy and other thermodynamic correlations can be taken from the steam thermodynamic diagrams [2].

Exergy destruction and loss of the system must be quantitatively calculated in the exergy analysis. Although this method is not a real improvement approach, it describes the efficiency of the system. Chemical energy defines the exergetic reaction related to the chemical degradation of the reaction. Thermal exergy is the maximum mechanical energy that can be measured, defined in the Carnot cycle. The maximum mechanical energy production is the heat that is caused by using the difference between the environment and the temperature. Energy conversion in control volume can be applied for energy, mass and types.

The irreversibility, which is the second law of thermodynamics, can also be understood as the amount of work accepted to bring the system to its original state. Entropy means the process of corruption. However, the amount of entropy formation in the system decreases with loss of exergy. Compound thermal and power plants are the best examples of the variation of exergy distribution. Heat and power can be produced from fuel.

However, the most exaggerated consumption of exergy is favourable for the combined heat production. When high-quality energy is used for energy production in turbines (exergy acquisition), combined heat production will have to use low-quality energy (low exergy) due to the relatively low temperature [3, 4]. The differences between energy and exergy are briefly summarised by Dincer and Cengel [5].

Exergy analysis is analysed in two parts as chemical and physical exergy [6]. Physical and chemical exergies are accounted for in exergy analysis calculations. For exergy calculations, various enthalpy and other thermodynamic correlations are taken from steam thermodynamic tables [4].

While thermodynamic laws are used for exergy analysis calculations, heat transfer equations are used for heat calculations. These calculations are used to analyse the applications of exergy. In exergy applications, cost analysis is performed by thermoeconomic analysis formulas taking these calculations into consideration.

3. Methods and equations for exergy analysis

Application of exergy calculations is based on energy and exergy efficiency. Prior to the exergy analysis, the input and output sections of the process flows are defined. Exergy calculations are based on input and output with mass flow rate, pressure, temperature, etc. parameters data in many applications by determining process flows. For the dead state, 25°C and 1 atm are assumed. If the mass input and output are equal in the control volume, the mass is preserved [3, 4]. It can be given, for instance, of control volume from thermodynamics equations. Exergy inputs and outputs are calculated. Exergy input and output are as follows [4]:

$$\sum \mathrm{Ex_i} = \mathrm{m_i}\left[(\mathrm{h_i} - \mathrm{h_\infty}) - \mathrm{T_\infty}(\mathrm{s_i} - \mathrm{s_\infty}) + \frac{\mathrm{V_i^2}}{2} + \mathrm{gz_i}\right] \tag{1}$$

$$\sum \mathrm{Ex_o} = \mathrm{m_o}\left[(\mathrm{h_o} - \mathrm{h_\infty}) - \mathrm{T_\infty}(\mathrm{s_o} - \mathrm{s_\infty}) + \frac{\mathrm{V_o^2}}{2} + \mathrm{gz_o}\right] \tag{2}$$

Exergy input and output differences show the exergy loss (irreversibility) in processes where the system is adiabatic and there is no work. Exergy loss can be defined as follows [4]:

$$\mathrm{Ex}_\ell = \mathrm{I} = \sum \mathrm{Ex}_\ell = \sum \mathrm{Ex_o} - \sum \mathrm{Ex_i} = \mathrm{T_\infty}\Delta S \tag{3}$$

Exergy equations are used in all application calculations of exergy. Besides, potential and kinetic energies can be neglected in practice because they are usually small. The specific flow exergy equations, which should be used for air and steam or other flows, can be given specific flow (steam or air) exergy as follows [1]:

$$\Psi = (\mathrm{h} - \mathrm{h_\infty}) - \mathrm{T_\infty}(\mathrm{s} - \mathrm{s_\infty}) \tag{4}$$

Specific enthalpy can be given as

$$\mathrm{h} = \mathrm{c}(\mathrm{T}–\mathrm{T_\infty}) \tag{5}$$

Specific entropy can be given as

$$\mathrm{s} = \mathrm{c}\ \ln(\mathrm{T}/\mathrm{T_\infty}) \tag{6}$$

Specific exergy can be given as

$$\psi = \mathrm{c}\ [\mathrm{T}–\mathrm{T_\infty}–\mathrm{T_\infty}\ \ln(\mathrm{T}/\mathrm{T_\infty})] \tag{7}$$

In the application of exergy, the efficiency of the second law of thermodynamics can be calculated as follows:

$$\eta_{\mathrm{ex}} = \frac{\Sigma Ex_{\mathrm{o}}(\text{Total exergy output})}{\Sigma Ex_{\mathrm{i}}(\text{Total exergy input})}\%100 \tag{8}$$

Exergy calculations of the application process can be performed by exergy of the auxiliary applications of exergy.

4. Application of thermoeconomic analysis

Thermoeconomic analysis is an analysis method that calculates the cost of exergy. In fact, it is a method of analysis that reveals the cost of energy availability. In other words, the thermoeconomic analysis is a detailed analysis of the cost of exergy. The cost of the exergy, which has many short definitions, is also called as the cost of the exergetic theory, exergy-economic cost or exergoeconomic. The most commonly used name is thermoeconomic. Thermoeconomic analysis approaches and methods are given by following steps.

The economic analysis of the exergy is excluded for exergy applications. Capital costs and operating costs are considered as investment costs. Capital investments are more strategic and have long-term effects. Since a capital investment often requires a large amount of money, the capacity of the project gains importance [7, 8].

The net present value method, using David Cantrell's approximate solution method [9, 10] to compare the economic cost effectiveness, is to calculate the interest rate (d) as follows:

$$d_r = \left[\left(1 + \left(\frac{P}{A}\right)\right)^{1/q} - 1\right]^q - 1 \tag{9}$$

where q = log[1 + (1/N)]/log 2, N is a period (=n × 12), P is a payment amount, A is an initial cost and N is a number of payments.

The CRF factor, which is the capital recovery factor calculated using a discount rate (d_r) and an amortisation (redemption) period (n) that determines a uniform annual cost to pay a debt or initial cost, is as follows [7–16]:

$$a^c = CRF = \frac{d_r(1 + d_r)^n}{(1 + d_r)^n - 1} \tag{10}$$

The cost of life cycle method can be calculated for technoeconomic analysis as follows [7, 8, 17]:

$$LCC_n = I_n + E_n + M_n + R_n - S_n \tag{11}$$

Simple payback period can indicate the number of years required to recover the initial investment with the project. It is considered that the project life is longer than the simple payback period [7, 8, 12]. The simple payback period can be shown as follows:

$$SPP = \frac{I_n}{\text{energy savings} - \text{maintenance cost}} \tag{12}$$

Thermoeconomic analysis is a method and technique analysis that can be used to analyse energy costs through a combination of the second law of thermodynamics and economic principles. Thermoeconomic analysis can be described as cost analysis and optimisation. This definition is the basis of thermoeconomic. The energy cost unit is expressed as [7, 18–21]

$$\dot{C}_{En}\ [\$/kW] = \frac{E_{cost}\ [\$]}{\dot{E}n_{net}\ [kW]} \tag{13}$$

Since this book contains exergy applications, the thermoeconomic analysis of the turbine power plant can be determined. If a turbine power plant is considered as an exergy application, the unit of exergy cost can be defined as follows [7, 8, 12, 18]:

$$\dot{C}_{Ex}\ [\$/kW] = \frac{E_{cost}\ [\$]}{Ex\ [kW]} \tag{14}$$

5. The contents of the book of exergy applications

Exergy threads used in different ways in different areas can be seen that when taken applications in many fields. There are many similar field exergy applications such as factory production processes, many technological product design, biotechnologies, renewable energy sources, power plants, military applications, cooling and air-conditioning.

In addition, application of exergy introduces many exergy subjects in the book's chapter. The chapters of the book cover the topics as follows: exergetic cost for thermal systems, exergy methods for commercial aircraft, thermodynamic performance of ice thermal energy storage systems, exergetic assessment in dairy industry, exergetic perspectives of various typical bioenergy systems, advanced exergy analysis of an integrated SOFC system, exergy analysis of wind turbine, applications of exergy analysis, application of exergy for vacuum cooling, exergy analysis in shell and helically coiled finned tube, human thermal behaviour and exergy consumption rate, low exergy solution to building heating and cooling, new exergetic methodology to promote improvements and performance of solar PV plant for exergy approach.

6. Conclusion

This book poses application of exergy and its technology in terms of development with new technologies. The main idea of this study is to research exergy application of all engineering and science field with current and new technologies. In addition, the research of the exergy application can indicate many critical issues for an engineering and science enhancement.

Moreover, compared to many types of exergy application, this book also researched many science areas. This book is both a source of inspiration for similar exergy applications and studies and can contribute to the resolution of energy problems.

This book emphasises that mass and energy balances will optimise exergy efficiency with the help of many design parameters. This study shows that energy problems, industrial and building energy savings, energy use in medical fields, cooling-heating-ventilation systems, power plants, many different energy systems, renewable energy systems and many other scientific problems can be solved by exergy optimisation.

This book will provide solutions to these types of energy problems through exergy applications that are aircraft, cooling, heating, fuel cell, wind energy, solar energy, motor vehicles, power plant and various different energy fields.

Nomenclature

Ex_i	Exergy input, kJ/kg
Ex_o	Exergy output, kJ/kg
h_∞	Dead state enthalpy, kJ/kg
h_i	Enthalpy input, kJ/kg
h_o	Enthalpy output, kJ/kg
s_∞	Dead state entropy, kJ/kg K
s_i	Entropy input, kJ/kg K
s_o	Entropy output, kJ/kg K
T_∞	Dead state temperature, K
z_i	Inlet height difference, m
z_o	Outlet height difference, m
η_{ex}	Exergy efficiency, %
c	Specific heat of the substance, kJ/kg K
g	Acceleration of gravity, m/s^2
h	Specific, air or steam enthalpy, kJ/kg
I	Irreversibility, kJ/kg
s	Specific, air or steam entropy, kJ/kg K
T	Sugar temperature, K
v_i	Fluid inlet velocity, m/s
v_o	Fluid outlet velocity, m/s
Ψ	Specific, air or steam specific flow exergy, kJ/kg
$\dot{C}_{En}$	The unit of energy consumption, \$/kW
CRF	Capital recovery factor, CRF = a^c

d_r	Discount rate (interest rate)
E_{cost}	Total energy cost, $
E_n	The present value of energy costs, $
Ex_i	Exergy input, kJ
Ex_o	Exergy output, kJ
I_n	Initial investment, $
i	Input (Inlet)
LCC	Life cycle cost, $
M_n	The present value of nonfuel operating and maintenance cost, $
$\dot{m}$	Mass flow rate, kg/s
$\dot{m}_i m_i$	Mass input, kg
$\dot{m}_o$	Mass output, kg
n	Amortisation period, year
o	Output (Outlet)
R_n	The present value of repairing and replacement costs, $
S_n	The present value of resale or salvage value, $
SPP	Simple payback period, year
∞	Dead state

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Section 2

Exergy Applications

Application of Exergy: A Low-Exergy Solution to Building Heating and Cooling

Lin-Shu Wang

Additional information is available at the end of the chapter

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Abstract

Considering heat a form of energy, the mechanical theory of heat (MTH), in making this historic advance of heat's ontological-category, made a relational-category error. The resulting energetic viewpoint of MTH sees all changes in nature as energy conversions and the problem of building heating and cooling as, quantitatively, energy-demand-and-supply problem. The IEA-ECBCS-programme introduced a correction to this energetic bias with the principle of "matching the quality levels between the energy supply and demand," which is known as LowEx approach. A recently formulated theory of heat, the predicative entropic theory of heat (PETH), is based on the cornerstone of correct categories of heat ontologically as well as predicatively (relationally). In the new theory, heat extraction plays a central role and, therefore, changes in nature are seen in terms of spontaneous entropy growth and heat extraction as powered by entropy growth potentials (EGPs). An alternative to the IEA-ECBCS's LowEx approach is suggested here based on heat extraction. Instead of matching of quality levels, LowEx can also be achieved by the management of natural EGPs: the *combined solar and heat pump systems* (S+HPs) can be transformed into LowEx S+HPs, a pure heat extraction system, by "eliminating" the energy conversion process of auxiliary heating.

Keywords: heat's categories, energy conversion, condensing boilers, IEA-ECBCS programme, low exergy approaches, reduction in exergy input to buildings, entropy growth potential, heat extraction, combined solar and heat pump systems (S+HPs), LowEx S+HPs

1. Introduction

The idea of *energy transformation* is central to the discussion of energy, energy efficiency, and the notion of sustainability. A suggestion is made that our misunderstanding of this critical

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concept hinders our ability to find the best solutions to building heating problem—and, in the long term, to have a clear vision in our pathway to sustainability.

The energy problem is really an exergy problem. One cannot address energy issues intelligently without the concept of exergy. In fact, the term energy is often used to mean high-exergy energy, not energy in the general sense as a term in physics. Recall energy is, standardly, defined as the capacity for doing work. This definition works only if it is applied to *high-exergy energy* or even *pure-exergy energy* (see Section 2), not energy in general. For instance, with a part of ocean as the heat reservoir, the vast energy of ocean's other parts, unless the temperature of these other parts is distinctively different from the one part, has no capacity for doing work.

This chapter considers the problem of building energy, especially buildings' energy need for their heating and cooling. That is, buildings' need of low-temperature heat during heating sessions and the removal of low-temperature heat during cooling sessions. The application of low-temperature heat differentiates discussion on exergy in this chapter from exergy considerations that deal with high-temperature heat applications. Ever since Kelvin, exergetic/entropic considerations for high-temperature heat applications are encapsulated, in large part, in terms of the Carnot-Kelvin formula. Kelvin himself never used the term of entropy in his writing; even so, entropic/exergetic consideration is partially captured by the formula without explicitly referring to entropy and/or exergy so long as one is dealing with high-temperature heat problems.

This is the reason why the Carnot-Kelvin formula (Carnot efficiency) occupies the central significance in thermodynamics treating high-temperature heat problems [1, 2]. For a heat source at a constant temperature, the formula is all one needs for defining the "perfection" of using the heat source. For other heat sources or other energy sources in which the temperatures do not remain constant, the formula is only an approximation. But, engineering practice guided by the formula is still approximately valid as a first step, and the exergy analysis yields the refinement over the first step. But, one would not suggest an unrefined engineering analysis without explicit exergetic content to be invalid.

Low-temperature heat problems for building heating are different matter. While the application of the corresponding formula for heat pump application to cooling is a part of the standard thermodynamics, no such formula is available in the usual treatment of building heating problems, which are usually handled in terms of the first law of thermodynamics alone. That better understanding of fuel's theoretical potential for building heating by explicitly making use of the formula has been suggested for the first time in [3].

In other words, whereas for high-temperature heat problems either the Carnot-Kelvin formula is the proxy for their entropic treatment or the full entropic/exergetic treatment is made available in the textbooks or the literature, the usual literature on building heating is deficient in being entirely based on the first law absent of the second law content. That deficiency has been pointed out by an International Energy Agency (IEA) group, energy conservation in buildings and community systems (ECBCS) programme. In an ECBCS programme research project, ECBCS Annex 49 [4], the case was made that exergetic analysis should be applied for high performance buildings and communities. This recommendation and conclusion deserve to be vigorously supported. It may be noted that this conclusion was consistent with what an earlier study by Ayres and Warr [5] suggested (see Section 4).

It is further noted that the theory of exergy is an integral part of the mechanical theory of heat (MTH) or classical thermodynamics, which is beset with inconsistencies (see [6]). In [6], Haddad wrote, "In fact, no other discipline in mathematical science is riddled with so many logical and mathematical inconsistencies, differences in definitions, and ill-defined notation as classical thermodynamics." The author shared with Haddad as well as many who commented on the matter before; it may even be suggested that MTH was a provisional theory, the logical conclusion of which is a new theory of heat, the predicative entropic theory of heat (PETH), [7] which was proposed in 2017 by the author (see Section 3).

The intent of this chapter is twofold: to support the LowEx approach initiated by IEA-ECBCS programme *and*, in this support, to formulate the LowEx approach with a different focus in terms of the PETH interpretation of the LowEx system approach—away from the IEA-ECBCS's focus of "matching the quality levels between the energy supply and demand" [4]. Our goal is the facilitation in the success of the LowEx approach with a specific proposed technology, LowEx combined solar and heat pump system (LowEx S+HP). Combined solar and heat pump (S+HP) systems was another IEA initiative (see below). LowEx S+HP represents a fundamental refinement of S+HP by cleansing MTH of its baggage of seeing all processes to be energy transformation processes: LowEx S+HP represents the idea of *electrification of heating with no energy transformation.*

2. Theory of exergy and universal energy transformation

As Ghoniem noted in the 2007 AIP-MIT Conference, "As we contemplate the impact of the inefficiencies associate with energy conversion…we realize that it is the quality of energy that matters and not the quantity, and that with each conversion step we lose, it is really Murphy's Law impersonated. In fact, some of us have come to conclude that we don't have an energy challenge, we have an entropy challenge" [8]. That is, the energy challenge cannot be dealt successfully with the first law reasoning alone. We must apply the second law as well.

This entropy challenge is best handled with the introduction of exergy in the analysis of energy, that is, exergy analysis. In introducing exergy analysis, Bejan et al. [9] explained the necessity of doing so as follows:

> *Exergy analysis also provides insights that elude a purely first law approach. Thus, from an energy perspective, the expansion of a gas (or liquid) across a valve without heat transfer (throttling process) occurs without loss. That such an expansion is a site of thermodynamic inefficiency is well known, however, and this can be readily quantified by exergy analysis. From an energy perspective, energy transfers to the environment appear to be the only possible sources of power plant inefficiency. On the basis of first law reasoning alone, for example, the condenser of a power plant may be mistakenly identified as the component primarily responsible*

> *for the plant's seemingly low overall efficiency. An exergy analysis correctly reveals not only that the steam generator is the principal site of thermodynamic inefficiency owing to irreversibilities within it, but also the condenser [loss] is relatively unimportant [9].*

In the literature, the introduction of exergy concept was often attributed to Josiah Willard Gibbs in an 1873 publication. The concept was then, in the 1940s and 1950s, extended by Joseph Keenan for engineering applications, adapting it for the practical analysis of thermodynamic cycles. Gibbs' insight, however, is traced by Daub, as he noted in a historical study on Entropy and Dissipation, to Kelvin, "Although Gibbs never once mentioned Thomson in his work, he was indebted, I believe, to Thomson's concept of dissipation of energy via the good offices of Maxwell and his *Theory of Heat*" ([10], 351). Indeed, Maxwell wrote in a review of Tait's "*Thermodynamics,*" [11] "Thomson, the last but not the least of the three great founders [Clausius, Rankine, and Thomson], does not even consecrate a symbol to denote the entropy, but he was the first to clearly define the intrinsic energy of a body, and to him alone are due the ideas and definitions of the available energy and the dissipation of energy." This bit of history has an important consequence on our understanding of MTH as well as what the role the theory of exergy has occupied (see below) in MTH.

2.1. Exergy

First, the development of research on exergy analysis after Keenan led to the modern definition: *The exergy of a thermodynamic system S in a certain state S_A is the maximum theoretical useful work obtained if S is brought into thermodynamic equilibrium with the environment by means of ideal processes in which the system interacts only with this environment* [12].

Another influential definition was formulated by Rant [13] and Baehr [14]: *Exergy is the portion of energy that is entirely convertible into all other forms of energy; the remainder is anergy.* That is,

$$\text{energy} = \text{exergy} + \text{anergy} \tag{1}$$

We shall refer to them as the first exergy definition and the second exergy definition.

While one defines exergy in terms of "the portion of energy…," and considers the application of exergy analysis as well as speaks about exergy components, exergy balance, exergy transfer, etc. (see [9], chapter 3) as one does about energy analysis, energy components, energy balance and energy transfer, one fundamental difference of exergy from a property such as energy, which is defined for a system, is that exergy in general can only be defined for a system *and* the environment (heat reservoir) the system interacts with. This point is explicitly made in the first exergy definition. In the second exergy definition, the point is not explicit, but implicit because the determination of the portion of energy in a thermal energy system that is entirely convertible requires the specification of a heat reservoir (see below, but, of course, that determination for pure exergy energies does not have this requirement).

According to Bejan et al. [9] the total exergy of a system E can be divided into four components: physical exergy E^{PH}, kinetic exergy E^{KN}, potential exergy E^{PT}, and chemical exergy E^{CH},

$$E = E^{\mathrm{PH}} + E^{\mathrm{KN}} + E^{\mathrm{PT}} + E^{\mathrm{CH}} \tag{2}$$

Chemical exergy will not be discussed here for brevity. The kinetic and potential energies are in principle fully convertible to work as the system is brought to rest or to its reference level, respectively. Accordingly, for a system of mass *m*,

$$E^{KN} = KE = \frac{1}{2}mv^2 \tag{3}$$

$$E^{PT} = PE = mgz \tag{4}$$

where v and z denote velocity and elevation relative to the reference level. The physical exergy of a closed system at a specified state is given by the expression,

$$E^{PH} = (U - U_0) + p_0(V - V_0) - T_0(S - S_0) \tag{5}$$

where U, V, and S denote, respectively, the internal energy, volume, and entropy of the system at the specified state, and U_0, V_0, and S_0 are the values of the same properties when the system is at the *restricted dead state* (see [9] for detailed definition), that is, the state of thermodynamic equilibrium with the reference environment of the first exergy definition. For details of the derivation of Eq. (5), see reference [9].

Clearly, Eq. (5) is based on the first exergy definition and a system's physical exergy is defined in terms of the system *and* its environment the system interacts with. With the first exergy definition, the value of physical exergy is not subjected explicitly to the notion that it is a portion of energy as the second exergy definition declares. In the latter case, as Eq. (1) implies, *exergy*≤*energy* (in other words, all three quantities [*energy*, *exergy*, and *anergy*] are positive-definite and a negative *anergy* will be senseless). Yet, there are systems and their environments for which *anergy*s are found to be negative and, for these instances, the second exergy definition is problematic. However, the "problematic" second exergy definition is necessary for the concepts of kinetic exergy and potential energy as shown in Eqs. (3) and (4); it captures importantly that kinetic and potential exergies—as examples of *pure exergy energies*–can be completely converted into other forms of pure exergy energy such as electrical energy. Nonetheless, while both definitions are necessary, there is a conflict between the two definitions in that the first exergy definition allows the possibility of *exergy*≥*energy,* whereas the second exergy definition implies that *exergy*, as a portion of *energy*, is always smaller than *energy*.

Pure exergy energies and physical exergy are core parts of the theory of exergy, as treated in Bejan et al. [9]: The exergy change between two states, state 1 and state 2, of a closed system is determined as,

$$\mathrm{E_2 - E_1 = (U_2 - U_1) + p_0(V_2 - V_1) - T_0(S_2 - S_1) + (KE_2 - KE_1) + (PE_2 - PE_1)} \tag{6}$$

2.2. Closed system exergy balance

The exergy balance for a closed system is developed by combining the energy balance and entropy balance, which result in,

$$(U_2 - U_1) + (KE_2 - KE_1) + (PE_2 - PE_1) = \int_1^2 \delta Q - W + T_0 \left[(S_2 - S_1) - \int_1^2 \left(\frac{\delta Q}{T} \right)_b - S_{gen} \right] \quad (7)$$

The terms inside the [] in the above expression represent the balance of entropic entities—entropic change, entropic transfer, and entropic generation—according to the second law. Moving $T_0(S_2 - S_1)$ to the left side of the equation and introducing Eq. (6) after adding $p_0(V_2 - V_1)$ to the left side as well as the right side, then collecting terms involving δQ on the right side, this equation can be rewritten as,

$$E_2 - E_1 = \left\{ \int_1^2 \left(1 - \frac{T_0}{T_b} \right) \delta Q - \left[W - p_0(V_2 - V_1) \right] \right\} - T_0 S_{gen} \quad (8)$$

The term on the left side of Eq. (8) is the exergy change of the closed system. The terms on the right side depend on processes: The first group represents two kinds of exergy transfers. The exergy transfer associated with the transfer of heat, E_q, is shown as

$$E_q = \int_1^2 \left(1 - \frac{T_0}{T_b} \right) \delta Q \quad (9)$$

The exergy transfer associated with the transfer of work, E_w, as

$$E_w = W - p_0(V_2 - V_1) \quad (10)$$

The last term on the right-side accounts for the destruction of exergy due to irreversibilities within the system as related to the entropy generation or entropy growth,

$$E_D = T_0 S_{gen} \quad (11)$$

In the literature, the expression of E_D is also known as the Gouy-Stodola theorem.

As long as both pure exergy energies and physical exergy are core parts of the theory of exergy, the theory is rest on both the first definition and the second definition—with the aforementioned contradiction between them. The only escape out of the dilemma is for the first definition to accept the restriction implied in the second definition, *energy = exergy + anergy*. That is, the energetic interpretation. This interpretation-restriction is in fact the widely held understanding of thermodynamics since Kelvin. In this interpretation, energy is then divided into four kinds, inclusively:

- Pure exergy energy: electrical energy, kinetic energy, and potential energy
- High exergy energy: high temperature heat
- Low exergy energy: low temperature heat
- Zero exergy energy: heat of a body that is in thermodynamic equilibrium with its environment.

The common definition of energy, *energy is the capacity for doing work*, is clearly applicable only to the pure exergy energies and inapplicable to low exergy energy and zero exergy energy. A better definition will be: *exergetic content of an energy system is the capacity for doing work*, which is applicable to all four cases.

2.3. Kelvin, origin of the exergy concept, and the notion of universal energy conversion

As Maxwell stated, "Thomson…does not even consecrate a symbol to denote the entropy, but he was the first to clearly define the intrinsic energy of a body, and to him alone are due the ideas and definitions of the available energy and the dissipation of energy," two important points were made in this sentence: Thomson (Later, Lord Kelvin) was the originator of the idea of available energy or exergy; secondly, the two papers, one paper on the universal dissipation of mechanical energy [15] and another paper on dynamical theory of heat [16], which originated the idea of available energy, also represented Thomson's formulation of the second law. Both points are not widely known, but deserve to be better disseminated for better appreciating the real meaning of the theory of exergy.

While the universal dissipation paper, though a short one, *is* widely disseminated, Thomson's idea on available energy was not explicitly shown in the 1851 paper, though the paper was Thomson's most significant publication. Rather, the idea was recorded in a passage of the draft for the paper:

> *The difficulty which weighed principally with me in not accepting the theory so ably supported by Mr Joule was that the mechanical effect stated in Carnot's Theory to be* absolutely lost *by conduction, is not accounted for in the dynamical theory otherwise than by asserting that* it is not lost *[i.e., the assertion of energy conservation]; and it is not known that it is available to mankind. The fact is, it may I believe be demonstrated that the work is* lost to man *irrecoverably; but [even though energy is] not lost in the material world. Although no destruction of energy can take place in the material world without an act of power possessed only by the supreme ruler, yet transformations take place which removes irrecoverably from the control of man sources of power which, if the opportunity of turning them to his own account had been made use of, might have been rendered available [17, 18].*

This passage was identified by Kelvin's biographers [18] to have a critical role in the evolution of Kelvin's scientific thought. Taking these two papers together, they represent Thomson's contribution to the second law by adding to the idea of the conservation of energy the idea of the availability of energy: the first law is in terms of the conservation of energy, while the second law is, in Thomson's formulation, in terms of the availability of energy. This (second part of) understanding has been called the *energy principle* [19], noting that the energy principle is different from the principle of the conservation of energy (the first part).

In a series of papers on the Clausius inequalities which cumulated in the 1864 entropy paper, Clausius formulated the second law in terms of entropy and universal growth of entropy—which has been universally accepted to be *the* second law of thermodynamics, especially since Boltzmann and Planck. The intriguing question is whether the energy principle is synonymous with the entropy principle. Many students of thermodynamics view them to be synonymous.

This would be a mistake [7, 19]. The correct view is that the energy principle is subsumed under the entropy principle [7, 19, 20]. Nonetheless, the energy principle is very important because the entropy principle, though universally true, has been viewed by physicists and chemists to have a negative degradation meaning only, as Prigogine recounted,

> *Among all those perspectives opened by thermodynamics, the one which was to keep my interest was the study of irreversible phenomena, which made so manifest the "arrow of time." From the very start, I always attributed to these processes a constructive role, in opposition to the standard approach, which only saw in these phenomena degradation and loss of useful work…The fact is that it appeared to me that living things provided us with striking examples of systems which were highly organized and where irreversible phenomena played an essential role…Those problems had confronted us for more than 20 years, between 1947 and 1967, until we finally reached the notion of "dissipative structure" [21].*

What Prigogine described was the view of physicists and chemists he had been striving to overcome to achieve his *dissipative structure* breakthrough. For engineers, however, the energy principle interpretation of Kelvin has long been developed by Gibbs and others into the theory of exergy, which views pure exergy energy and high exergy energy to be the driving constructive force of making things happen.

That understanding is the understanding of universal energy conversion or transformation. In this understanding, the Carnot heat engine is interpreted as "a theoretical power cycle of maximum efficiency for converting thermal into mechanical energy" [22]. In fact, every change in nature and in man-made world can be viewed in terms of energy transformation. This understanding is the great contribution of Joule, Thomson, and the theory of exergy. The modern world would not have existed without this understanding. Except, it comes to a point today that this understanding also hinders our going forward from this point.

3. The predicative entropic theory of heat

Joule, Thomson, and Clausius formulated MTH by correcting, famously, the categorical error in the then prevalent theory of heat, the caloric theory of heat, which considered heat to be an invisible and weightless substance. Instead, MTH considered heat and its nature, namely its category, as: "heat is not a substance, but a dynamical form of mechanical effect" [16] as noted by Thomson.

But, this correction also created its own problem in that MTH committed the classical mistake of conflating correlation, or connection, with causality. It interpreted the connection between heat and work according to Joule's principle of *mechanical equivalent of heat* (MEH) to be the existence of causality between heat and work: not only work (mechanical energy) causes heat but also heat causes work. That is, MEH has been universally interpreted to be the principle of heat and work interconvertibility. Correspondingly, thermodynamics is the subject that mainly concerned with the transformations of heat into mechanical work and the opposite transformations of mechanical work into heat. In this understanding, we find heat's apparent utility.

3.1. Energy and its exergetic content

As it was pointed out above, Kelvin also formulated the second law in terms of what is called the *energy principle*—the concept of availability of energy and the idea that, whereas energy never disappear, its availability dissipates. For instance, mechanical energy and high-grade energy dissipate universally. In the paper [15], in which he declared the universal dissipation of mechanical energy, he did not so much prove the assertion as simply declared it to be a self-evident proposition. Remarkably, those who followed Kelvin accepted it so as well and Von Baeyer was typical with this assessment, "Inasmuch as the second law is one of the pillars of physics, this was Thomson's most significant contribution to the science of thermodynamics, and overshadowed his invention of the absolute scale of temperature, his early recognition of the importance of James Joule's work…" [23].

But, in fact, the energy principle is not synonymous with the entropy principle. The energy principle can be shown to be subsumed under the entropy principle [19] rather than being a universal principle. Nonetheless, the energy principle, though defective as a universal principle, remains to have an important role in MTH. Without it, we only understand "heat's apparent utility" in terms of the Carnot-Kelvin formula. The role of any energy system in the production of mechanical work would have to go through the heat release phase, that is, only indirect energy conversion would be physically possible.

This is clearly not true as evinced by the existence of direct energy conversion, for example, direct energy conversions corresponding to Gibbs free energy or Helmholtz free energy, as well as photovoltaic effect and fuel-cell processes [22]. These concepts have been generalized into the concept of exergy.

The theory of exergy affirms the importance of the energy principle by going beyond the idea of heat's apparent utility as the driver of nature to the more general idea of energy, more precisely the *exergetic content in energy,* to be the driver of nature. Furthermore, by incorporating the entropy principle into the meaning of exergy, the theory of exergy discloses the constructive meaning in the entropy principle in addition to the principle's usual destructive meaning. This last point is no small matter—which is pretty much how MTH or thermodynamics should be understood as encapsulated in the notion that all processes in nature are energy conversion or transformation processes.

With this advance, we took the step from "thermodynamics is mainly concerned with the transformations of heat into mechanical work and the opposite transformations of mechanical work into heat" to the notion that thermodynamics deals with universal energy transformation.

Note that energy would be, without the concept of exergy, a lame concept; it is the exergetic content in energy that gives energy its capacity or usefulness.

3.2. Entropy growth potential

This is how heat and energy enter every student of thermodynamics' mind concerning their importance. However, this understanding is deeply misleading: both the idea of the *consumption* of heat as the cause of work production, and the notion that exergy is a *part* or *portion* of energy, are wrong.

In the latter case, as it was already pointed out that the notion was self-imposed one, one that is contradicted by known facts, for the sole purpose of avoiding the contradiction between the first exergy definition and the second exergy definition. Exergies considered in the theory of exergy are incomplete, and there are driving forces outside the set of pure exergy energy, high exergy energy, and low exergy energy; the theory of exergy and, correspondingly, MTH are contingent or provisional.

For the former case, the idea of consumption appears when during high-temperature heat being transferred to a low-temperature sink, a part of this high-temperature heat is converted into work with the balance of heat going to the sink. In association with this picture, the accepted way of defining heat emphasizes the fact that heat is defined only as a process associated with its transition not as an entity: *heat is energy in transit.* There is an awkwardness in handling the notion of heat in the literature [19, 24, 25].

The awkwardness in explaining heat in the absence of heat in transition can be avoided: it was the mistaken result of giving heat the role that it cannot fulfill to begin with. The problem in both cases has to do with the question, what is the causation driving all phenomena in nature? The difficulty of considering energy (and heat) as *the* driver is removed once it is shown that the real driver is *entropy growth potential* (EGP), a concept formulated in Refs. [7] and [19]. Energy is only the proxy of the real driver because of its EGP or exergy.

The way to understand **heat's apparent utility** is that there exists EGP in association with heat transfer process, and this EGP drives the heat-to-work conversion by extracting heat from the heat reservoir (while it was considered to be a heat sink to the transfer process) converting extracted heat into work. The correct way to understand MEH is that equivalence exists between *extracted* heat and work, not *consumed* heat and work.

The way to understand **energy with its exergetic content** is that, again, there exists EGP in association with energy conversion process and it is this EGP which drives the conversion process involving the "energy system." If EGP in this case is greater than energy in the "energy system," this would explain that this is a case of negative anergy. Other than these special cases, for the cases in which anergy is positive the concept of energy with its exergetic content remains serviceable: the language of exergy is useful in its proper context; it is just not a universal language.

We see the evolution of the ideas from heat's apparent utility to energy with its exergetic content to entropy growth potential. MTH succeeded in placing heat in its ontological category; PETH succeeds in placing heat in its predicative category. MTH is obviously an important phase in this evolutionary arc: there will be no PETH without going through MTH. But, MTH, which is beset with inconsistencies because of predicative-category error, is a provisional theory of heat. The logical conclusion of the evolutionary arc is PETH.

3.3. PETH and heat extraction: energy thinking vs. entropy thinking

MTH is encapsulated in the notion that all processes in nature are energy conversion or transformation processes. The new theory of heat, PETH, points out that Kelvin's energy principle is not a universal principle: mechanical energy dissipates spontaneously not universally, and there are EGPs that do not involve degradation of energy, [19] that is, conversion of energy. That is, not all processes in nature are energy conversion processes and that there are processes of "purely spontaneous" kind [19] involving no energy conversion. In PETH, the expanded set of "exergies" is made of the following:

- #1_Purely spontaneous systems: system EGP does not involve change in system energy; this characteristic applies to all isolated systems with spontaneous tendency
- #2_Negative-anergy energy: energy systems whose "exergy" are greater than change in system energy
- #3_Pure exergy energy: electrical energy, kinetic energy, and potential energy
- #4_High exergy energy: high-temperature heat
- #5_Low exergy energy: low-temperature heat
- #6_Zero exergy energy: heat of a body that is in thermodynamic equilibrium with its environment.

Furthermore, EGPs can be divided into *stock* EGPs and *ongoing* or *natural* EGPs. Examples of the latter are solar phenomena and wind phenomena for which the entropy growth is ongoing. Unlike systems of stock EGPs (i.e., stock energies) for which accelerating entropy growth happens only when the systems are brought into use, natural EGPs are phenomena that entropy growth is ongoing [19]. Therefore, unlike for systems of stock EGPs, their use always leads to increase in entropy growth in accordance with the second law, the management of natural EGPs does not intrinsically lead to faster entropy growth [19]. That is, the second law asserts only that entropy growth cannot be negative, not that the rate of ongoing entropy growth cannot be slowed. Therefore, a further insight on the kind of "exergetic" phenomena is the addition to the above list of the phenomena:

- #7_Natural entropy growth potentials (EGPs) phenomena

In this list of seven, #1, #2, and #7 are new conceptual advance PETH brings forth to the theory of heat.

An important inference of PETH [7, 19] is that all reversible processes and reversible-like processes are heat extraction processes. Heat pump is one example of heat extraction process. Notably, heat engine and Carnot heat engine are also examples of heat extraction processes: in so far as a Carnot heat engine is the perfect inverse of a Carnot heat pump, the interpretation of the Carnot heat engine as a device of perfect heat extraction driven by EGP is obviously a more satisfactory way of seeing it. Consequently, the essence of the new theory of heat, PETH, is that the pathway to reach high efficiency is through thinking in terms of heat extraction, entropy, and natural EGPs, rather than the old fashion way of thinking in terms of energy transformation, energy balance, and stock energy alone (see **Table 1**).

It is useful, therefore, to go beyond energy thinking to apply entropy thinking, more specifically heat extraction thinking, to understand nature as well as contrive engineering devices and systems. In the following, we shall make the case that by going beyond the conventional

Energy problems are in terms of…in MTH	Energy efficiency are achieved in terms of…in PETH
Energy transformation efficiency	Heat extraction
Energy balance	Entropic management minimizing entropy production
Stock energy	Natural EGPs

Table 1. Different approaches to energy efficiency in MTH and in PETH.

framework that all devices are energy transformation devices, for short energy transformers, a new way of viewing what a heat pump is will introduce a better way of using the heat pump technology for space heating.

4. The LowEx system approach: reduction in exergy input to buildings

Energy requirement for building heating is a form of low-temperature heat. An evaluation of how we used low-temperature heat in the twentieth century was reported in a study by Ayres and Warr [5], a tabulated summary of the study is reproduced as **Table 2**. The numbers in **Table 2** are the exergy efficiencies (i.e., the second law efficiency) of five categories of energy uses and progresses made in their practices in the twentieth century; what the numbers show is that efficiency in *low temperature space heat* category was singularly poor by one order-of-magnitude in comparison with the other four categories. Intuitively, this gap suggests, as a matter of physics, that there is a large room for improvement. It is this situation giving rise to the idea that we should be able to make very significant strides in the twenty-first century toward high performance buildings (easy goals as low hanging fruits).

Year	Electric power	Transportation	High temperature industrial heat	Medium temperature industrial heat	Low temperature space heat
1900	3.8%	3	7	5	0.25
1910	5.7	4.4			
1920	9.2	7			
1930	17.3	8			
1940	20.8	9			
1950	24.3	9			
1960	31.3	9			
1970	32.5	8	20	14	2
1980	32.9	10.5			
1990	33.3	13.9	25	20	3%

Table 2. The exergy efficiencies of five categories of energy applications in the twentieth century.

On the other hand, what the gap suggests is not that improvement can be easily made but, instead, our poor comprehension of the science of heat leading to persistency of the gap, which, unless a new course set on a better understanding of heat is formulated, will persist.

Such a correct course was pointed out by IEA-ECBCS in ECBCS Annex 49 [4]: it is necessary to treat the problem of energy in terms of its exergy content and to recognize that some of what buildings need is not high exergy energy, but low exergy energy. The reason that the twentieth century record of handling low temperature heat in buildings was so appalling and the twenty-first century result is not much better is because we satiated and continue to satiate the building low exergy energy need with high exergy energy.

On our understanding of energy, IEA_ECBCS Annex 49 has this to say:

> *The quantity of energy is given by the first law of thermodynamics, and is calculated from energy balances for a system. Current energy systems in buildings are designed and improved based on this law. This means that of course the quantity of energy supplied is matched with the quantity of energy required. Highly efficient condensing boilers, with efficiency of up to 98% are a straightforward result of such an analysis framework…In the case of the highly efficient boilers mentioned above when used to supply low temperature heat, the potential to produce work (exergy) of the fuels fed into the boiler is almost completely lost in the combustion process. Due to this loss of energy potential, a large consumption of exergy occurs. Exergy efficiencies for such building systems are lower than 10% ([4], 7).*

Such conclusion is consistent with Ayres and Warr's analysis as shown in **Table 2**, as well as with our findings [3]. The bold step taken by EU's Ecodesign Standards [26] banning the sale of inefficient boilers of non-condensing kinds as of 26 September, 2015, in view of this analysis, was not bold enough: the step from inefficient boilers to highly "efficient" condensing boilers was only a timid step. As a practical matter (of short-term energy savings), though, it was a courageous move by European Commission and the move was made possible only "after years of grueling negotiation between the Commission and industry representatives" [26]. From energy efficiency point of view, the move was as significant as the phasing out of incandescent lightings.

For long-term energy saving, IEA_ECBCS argued for the application of exergy method for building operation: "*The core and first principle of the exergy method applied to the design of energy systems is to match the quality levels of the energy supplied and the energy demanded*" ([4], page 33). This first principle of the exergy method is reiterated in its conclusions:

> *The energy approach, both on a building and community levels, intends to reduce energy demands in buildings by increasing insulation levels or increasing the air tightness of the building envelope, that is, optimizing the building shell. The exergy approach at both levels focuses on matching the quality levels between the energy supply and demand. Therefore, it requires the use of low quality sources for low quality demands like space heating. Demands requiring higher quality levels, such as lighting, electrical appliances or mobility, would in turn need the use of high quality sources" ([4], 69).*

That is, the exergy approach intends to reduce exergy input to buildings by inputting energy of the exergy level closer to the actual exergy level demanded for heating buildings.

IEA_ECBCS argued against using combustion processes for providing the low-temperature heat demands in buildings, but did not oppose the use of combustion processes generally: district heating and CHP systems are good examples of "matching the quality levels between the energy supply and demand" by generating power for meeting demands of high-quality level and heat for demands of low-quality level.

In this chapter, the author suggests an alternative way of achieving the goal of LowEx, reduction in exergy input to buildings. It is a case for eliminating combustion processes generally for *electrification of heating*.

5. Clean electricity and heat pumps for space heating

Heating has been achieved since the dawn of civilization with the discovery of fire. In contrast, there had been no reliable way of space cooling before the nineteenth century, that is, even though there had had assorted ways of cooling devised by mankind in the past, but none of them was reliable enough to become dominant practice at any given time. That of course changed in the later part of the nineteenth century with the invention of heat pump, or air-conditioning equipment, which has become the universally adopted practice of space cooling. Again, this reliable cooling came into existence only about a little over one century, a tiny fraction of the long duration that man has mastered fire for heating.

It should be noted that the concept of heat pump was introduced by Kelvin [27] for application for both cooling and heating. Furthermore, the conceptualization and invention of heat pump was made by Carnot and Kelvin at the same time with the conceptualization and invention of heat engine. Carnot famously introduced the concept of reversible machine, which can serve as a heat engine, as well as a heat pump when the heat engine machine reverses its operation. As the invention of heat engines amounted to the invention of the *second use of fire* by mankind (see Ref. [19]), the invention of heat pump can be viewed to be an integral part of that giant step taken by mankind. The intrigue point is that why this progress in cooling has not been accompanied with similar technological-progress in heat pump based heating, instead of continuingly relying on the same old combustion process, that is, condensing boilers are the ultimate perfection of that ancient practice.

One reason is that while heat pump is the only reliable means for cooling, the use of heat pump for heating must compete with the long-established practice of fire for heating. Absence of competition, heat pump for cooling achieved instant success. With the low-cost combustion process, heat pump must compete in terms of cost and reliability.

Performance of heat pumps is measured in terms of coefficient of performance (*COP*), which is the ratio of heat supplied by the HP unit to the electric energy required by the HP compressor. However, the real measure of a heat pump system is the system's overall COP_{System}, the ratio of heat supplied by the HP system to the sum of all electric energy required by the whole system, including compressor requirement as well as all pumps requirement and, especially, the energy requirement of the auxiliary heating.

Since the thermal efficiency of a conventional power plant is approximately 33%, it takes 3 unit of primary energy to produce 1 unit of electricity. Consider a perfect condensing boiler of 100% boiler efficiency. To have a HP system to compete with a condensing boiler, the HP system's COP_{System} must be higher than 3.

The kinds of HP systems include air source heat pump (ASHP), ground source heat pump (GSHP), and combined solar and heat pump (S+HP, see [28]), which is also known as solar-assisted heat pumps (SAHPs, see [29]). Freeman et al. [30, 31] investigated the energy used for heating the 120 m^2 floor area house, located in Madison, Wisconsin, with stand-alone ASHP, conventional active solar, series SAHP, and parallel SAHP with a 3-ton heat pump unit and solar collectors of collector area of **30 m^2**. Their result, which did not include GSHP, is reproduced in **Figure 1**.

There are important conclusions to be drawn from these findings on the breakdown of heating contributions: What matters are the purchased heats (which is purchased electrical energy), which include the auxiliary electrical resistive heat and the portion of heat delivered by heat pump associated with compressor energy input. The total heat delivered by the heat pump unit is the sum of that portion *and* the Q_{Air} portion in the case of parallel SAHP or the Q_{Solar} portion in the case of series SAHP. *COPs* shown in the captions are the COP of the heat pump

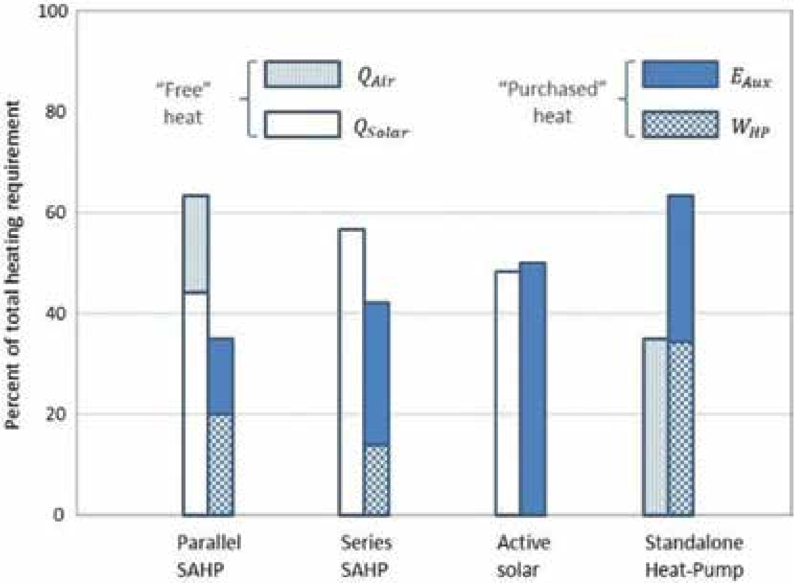

Figure 1. Heating contributions from all sources for standalone heat pump system, standalone active solar system, series SAHP system, and parallel SAHP system. Collector areas for the three cases using collectors are 30 m^2. The *COPs* of standalone heat pump, series SAHP, and parallel SAHP are 2.07, 2.84, and 2.00, respectively.

units, while the system COP_{System}, as read from the figure (taking the right column value in units X, $COP_{System} = 100/X$), are as follows:

Standalone heat pump: COP_{System} is smaller than 2

Active solar: COP_{System} is about 2

Series SAHP: COP_{System} is above 2 but smaller than 3

Parallel SAHP: COP_{System} is a shade short of 3

The results suggest that both series and parallel SAHPs, two examples of S+HPs, show much improvement over the baselines of standalone heat pump and standalone solar, but, so far as this study for heat pump use at Madison, Wisconsin is concerned, none of the four heat pump systems can overtake the performance of condensing boiler. The key reason for that is that, for all cases, auxiliary heating is required (and allowed).

But heat pump systems do have an important advantage if the electric grids become less dependent on fossil-fired power plants and more dependent on renewable energies. Such renewables powered grid is called *clean electricity.* Energy Transitions Commission, a leading industries, investors and climate advocates group, in its 25 April, 2017 press-release made the case, "Falling costs of renewables and batteries make cost-effective, clean electricity unstoppable and essential to the transition to a low-carbon, energy abundant world." It is *clean electricity* that makes electrification of transport unstoppable. Clean electricity and electrification of buildings is another potentially unstoppable combination if heat pump system for heating can achieve a clear-cut advantage in COP_{System} of multiples of 3. The following outlines a pathway of reaching the goal.

6. The PETH interpretation of LowEx approach: LowEx S+HP systems

In MTH, the Carnot heat engine is interpreted as "a theoretical power cycle of maximum efficiency for converting thermal into mechanical energy" [22]. In fact, every change in nature and in man-made world can be viewed in terms of energy transformation. PETH, however, suggests that there are processes that cannot be captured in terms of energy conversions in the traditional understanding of MTH and these processes are represented by the three kinds of processes: #1_Purely spontaneous systems: system EGP does not involve change in system energy; #2_Negative-anergy energy; #7_Natural entropy growth potentials (EGPs) phenomena (see also **Table 1**). While the first two kinds are of theoretical interest, the last one, natural EGPs, is of special significance in formulating a way for a sustainable future.

The concept of heat extraction is the new central concept in PETH for harnessing natural EGPs. The concept is derived from the proposition that equivalence exists not between consumed heat and produced work but between extracted heat and produced work. Not only a Carnot heat pump should be approached as a heat extraction machine, but also a Carnot heat engine is better comprehended as a heat extraction machine that is powered by EGP associated with the heat transfer tendency of temperature difference. In the strict sense heat is never consumed thus converted, but extracted in a process driven by EGP.

Unfortunately, in the MTH dominated physics students of thermodynamics are trained to think in terms of energy transformation instead of heat extraction. Emden in the 1938 Nature article, *Why do we have Winter Heating?* offered these responses:

> *The layman will answer: "To make the room warmer." The student of thermodynamics will perhaps so express it: "To import the lacking (inner, thermal) energy." If so, then the layman's answer is right, the scientist's wrong.*

The scientist thinks in terms of energy transformation and, by importing the energy which is guaranteed by physics to be transformed into heat in accordance with the energy conservation principle, heat is delivered to meet the demand.

LowEx approach offers a better response by matching the quality levels between the energy supply and demand. That is, to import the lacking energy with energy of a closely matched quality level. This is a response that is half way between the first law that thinks in terms of demand and supply and the second law in terms of quality of energies.

The advent of clean electricity offers a third response—instead of a matter of demand and supply—"To make the room warmer" suggests that the solution can be a matter of reversible-like operation driven by clean electricity. Clean electricity, a form of pure exergy energy, is the perfect driver of heat extraction operation, a reversible-like operation, for harnessing natural EGPs such as solar → infrared radiation phenomenon.

Evidence shown in **Figure 1** shows that S+HP systems enjoy significant improvement over standalone heat pumps. What is also obvious is that in the world of seeing every process in terms of energy transformation the use of auxiliary heating is allowed without hesitation. With such compromise, S+HP systems are no longer straightforward heat extraction devices, but are partial energy transformation devices, or for short, energy transformers.

6.1. Parallel S+HP vs. series S+HP, searching for LowEx S+HP

We propose a different definition of LowEx approach. The term LowEx is hereby used as the general goal of reduction in exergy input to the building (see definition in [4], Fig. 4.2, page 33). This goal can be met by matching of the exergy level of the input energy with the low exergy level of demanded energy, or by the following approach.

LowEx approach is defined not by excluding the use of high exergy energy, but by limiting high exergy energy or pure exergy energy use for maximizing the application of natural EGPs. A LowEx S+HP is, therefore, a *pure* heat extraction S+HP system of adequate solar collector area and sufficient TES capacity for eliminating auxiliary heating under normal operation.

Note that normal operation is defined to be the time during which 99% of a building heating need is called for. That is, for a "pure" heat extraction system equipped building its auxiliary heat is no more than 1% of the annual heating need of the building.

The following gives an outline of determining the requirement of collector area and TES tank capacity of a 200.1 m^2 floor area building in Stony Brook, NY, location, for meeting the LowEx goal. Simulation is used for this determination. The assumptions of simulation are as follows.

ACTIVE SOLAR

	5 STPs					
	200 gal TES tank		700 gal TES tank		1500 gal TES tank	
	Electric→heat	System total	Electric→heat	System total	Electric→heat	System total
Heat delivered (kW-hr)	5510	8019	4800	7712	4100	7435
Electric energy input(kW-hr)	5510	6006	4800	5330	4100	4654

PARALLEL S+HP

	5 STPs					
	200 gal TES tank		700 gal TES tank		1500 gal TES tank	
	Electric→heat	System total	Electric→heat	System total	Electric→heat	System total
Heat delivered	?	11,750	?	12,010	?	12,270
Electric energy input	?	3083.2	?	2943.2	?	2932.7
	10 STPs					
	200 gal TES tank		700 gal TES tank		1500 gal TES tank	
	Electric→heat	System total	Electric→heat	System total	Electric→heat	System total
Heat delivered	?	12,368			?	13,509
Electric energy input	?	2840.4			?	2639.2
	15 STPs					
	200 gal TES tank		700 gal TES tank		1500 gal TES tank	
	Electric→heat	System total	Electric→heat	System total	Electric→heat	System total
Heat delivered					?	14,696
Electric energy input					?	2439

SERIES S+HP

	5 STPs					
	200 gal TES tank		700 gal TES tank		1500 gal TES tank	
	Electric→heat	System total	Electric→heat	System total	Electric→heat	System total
Heat delivered	585	?	247	12,183	212	11,710
Electric energy input	585	3729	247	2012	212	1898
	10 STPs					
	200 gal TES tank		700 gal TES tank		1500 gal TES tank	
	Electric→heat	System total	Electric→heat	System total	Electric→heat	System total
Heat delivered	149.5	11162.5	121	11,189	107	11,015
Electric energy input	149.5	1581.2	121	1401	107	1296
	15 STPs					

	200 gal TES tank		700 gal TES tank		1500 gal TES tank	
	Electric→heat	System total	Electric→heat	System total	Electric→heat	System total
Heat delivered					51.8	10309.8
Electric energy input					51.8	898.4

Table 3. Three heating systems' simulation results for determining their ability in meeing LowEx crit.

- Building envelope: ASHRAE 90.1-2008 zoon 4A minimum requirement
- Climate: TYM3 data
- Equipment:
    - ASHP-WH-MDC12C9E8
    - WSHP-DAIKIN WRA 036
    - STP-SunMaxx ThermoPower-VDF30 (Aperture area of each panel = 2.67 m^2)

The simulation results are shown in **Table 3**.

As shown by Freeman et al. [30] the electric energy consumption of active solar systems is high unless very large solar collector area is used. In that case, however, the cost will be prohibitive. Combined solar and heat pump system (S+HP), therefore, "has the potential of alleviating the limitations each system [i.e., standalone solar system or standalone heat pump system] experiences individually in cold weather" [29], and "has great potential for improving the energy efficiency of house and hot water heating systems" [32]. By the comparative study in **Table 3** of parallel S+HP and series S+HP, this potential in the feasibility of combined systems for meeting LowEx goal can be assessed.

6.2. Discussion

In the literature as cited by Andrews et al. [33] series S+HP was judged inferior to parallel S+HP. Same conclusion has been made by IEA-Task 44. [32]. The principal reason was pointed out in [33] that it is due to, for series system with inadequate solar collector area, heat pump starvation, the phenomenon that when water temperature becomes too low the water source heat pump is starved from source thermal energy—necessitating the input of electric→heat auxiliary heating. Our result agrees with this assessment in the case of 5 STPs and 200 gal tank, which shows electric energy input (i.e., pure exergy energy input) of 3729 and 3083 kW-hr for series and parallel, respectively.

On the other hand, series combined system is the pure expression of heat extraction configuration, the configuration that can benefit from "alleviating the limitations each system experiences individually in cold weather," while the parallel combined system does not enjoy the same level of synergy. For instance, the solar collector of a series S+HP operates at lower temperature, thus, has higher performance and can be made of lower-cost construction. It is, therefore, interesting to examine the impact of TES tank thermal storage and the impact of larger solar collector area on mitigating heat pump starvation.

It is significant that for the case of small solar collector area, the impact of thermal storage on energy input for parallel combined system is 3083 → 2943 → 2933, while for series combined system 3729 → 2012 → 1898, a drastically greater reduction in pure exergy energy input as a result of thermal storage application.

The impact of solar collector area on energy input can be seen for the case of 200 gal thermal storage for parallel combined system 3083 → 2840 vs. for series combined system 3729 → 1581. Also, for the case of 1500 gal thermal storage for parallel combined system 2933 → 2639 → 2439 vs. for series combined system 1898 → 1296 → 898. In fact, theoretical consideration would conclude that as solar collector area approach infinitely large, the parallel combined system would forfeit any synergetic advantage and have the same performance as standalone active solar, which is of course a poor performer.

6.3. Pure heat extraction process: LowEx S+HP

Even though **Table 3** shows only incomplete simulation results of our study, which is ongoing, the above discussion offers sufficient evidence to support our theoretical argument on the superiority of pure heat extraction process as manifested in the form of series combined solar and heat pump system. With adequate solar collector area and adequate thermal storage, the *electric → heat* becomes less than 1% of *System total heat delivered* (e.g., 107, which is 0.97% of 11,015), such series S+HP meets the LowEx criterion and becomes LowEx series S+HP.

7. Conclusion: beyond energy transformation

This chapter presents an alternative approach of achieving the goal of LowEx, reduction in exergy input to buildings.

In describing the original approach to the LowEx goal, IEA-ECBCS argued, "combustion processes should not be used for the production of low temperature heat" ([4], 33). This is a partial step against the practice of combustion energy transformation since it did not argue against combustion processes generally. Undoubtedly, IEA-ECBCS made a good case against boilers and condensing boilers, a bad example of using combustion, in favor of district heating with combined heat and power (CHP), example of better use of combustion.

The author makes here the general case of beyond energy transformation, that is, against combustion processes generally in favor of clean electricity of solar and wind farms powered grids *and* the corresponding electrification of transport and electrification of buildings. In making the specific case of electrification of building heating, the author has introduced two innovations: The first is the recently formulated new theory of heat, PETH, which represents the logical conclusion of MTH by establishing the theory of heat on the solid cornerstone of correct categories of heat ontologically as well as predicatively. The case is an application of the first innovation as shown in **Table 1**. The second innovation is in the details, as suggested in **Table 3**, of how the application should be carried out: improvement of the series combined solar and heat pump systems (series S+HPs) by using thermal-storage/STPs for "eliminating" the energy conversion process of auxiliary heating so that the improved systems, LowEx S+HPs, can achieve clear-cut efficiency superiority over condensing boilers.

Both combined systems, combined heat and power and combined solar and heat pump systems, can supplant condensing boilers with superior efficiency. The new heat pump-based heating and cooling systems have the additional advantage of facilitating the goal toward clean electricity ecosystem.

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Chapter 3

Exergetic Assessment in Dairy Industry

Seda Genc

Additional information is available at the end of the chapter

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Abstract

Food industry is one of the most energy-consuming industries. Exergy analysis for several food production lines had been the subject of some studies in the past. However, dairy industry has a significant importance among food industry from energetic point of view since it covers many heating and cooling processes. Energy and exergy analyses are commonly used techniques for performance assessment of thermal system where exergy analysis is known as the powerful tool. In this context, exergy calculation methodology and a review of exergetic assessment in dairy industry are the subject of this study for the first time to the best of author's knowledge. In this chapter, first of all, exergy analysis methodology is presented, and next, assessment of exergy analysis of some dairy processes including milk pasteurization, milk powder process, flavored yoghurt production, yoghurt production and yoghurt drink processes are reviewed, comprehensively. Application of the exergy analysis in dairy industry supplies possibility to perform exergetic design, simulation, analysis and performance assessment.

Keywords: exergy, food, dairy products, milk processing, liquid milk, milk powder, yoghurt, yoghurt drink

1. Introduction

Milk is one of the most consumed foodstuff with its perfect nutritional qualities. It is a good source of not only carbohydrate, protein and fat but also vitamins (A, B2, B12, D) and minerals (calcium, potassium, phosphorous, iodine) [1]. The constituents of milk provide to have stronger bones, support immunity and give energy to the body. However, after milking, milk is processed immediately into several dairy products namely yoghurt, cheese, butter, and so on because of its easily spoiled characteristics [2].

Total milk production is 704 M tones/year, globally and the top three producer countries are the USA, India and China [3]. The World dairy industry market value is USD 336 billion in 2014; however, due to its very dynamic pattern, its market value displays continually growing trends which are forecasted to increase 32% generating USD 442 billion value in 2019. According to FAO, consumption of dairy products is consequently expected to increase by 20% or more before 2021. In this context, dairy production and dairy processing clearly appear as industries of significant importance for next decades [4, 5].

Dairy plants with different capacities and products are located all over the world. In general, it can be classified into different subsectors based on dairy products such as liquid milk, butter, yoghurt, cheese, milk powder, and so on. All the production lines at dairy industry include a number of unit operations namely cooling, thermisation, separation, homogenization, heat treatment, evaporation, drying, and so on. The major dairy products and their processing lines are schematized in **Figure 1**. After milking, physical, chemical, biochemical and microbial changes occur in the milk. The first process applied to the milk is cooling to prevent most of changes mentioned above. In some cases, milk is immediately cooled at the farm and transported to the factory with cold chain and continue to be stored in cooled tank in the factory. For some dairy plants with high capacity, the pasteurization or processing following raw milk reception is not possible. Then, milk is stored in cooling tanks to prevent deterioration, but if storage time will be long, pre-heat treatment known as thermisation is applied to the milk. The aim of thermisation is to inhibit microbial growth before pasteurization process and is applied at 63–65°C for 15 s operating conditions which is below pasteurization temperature [1].

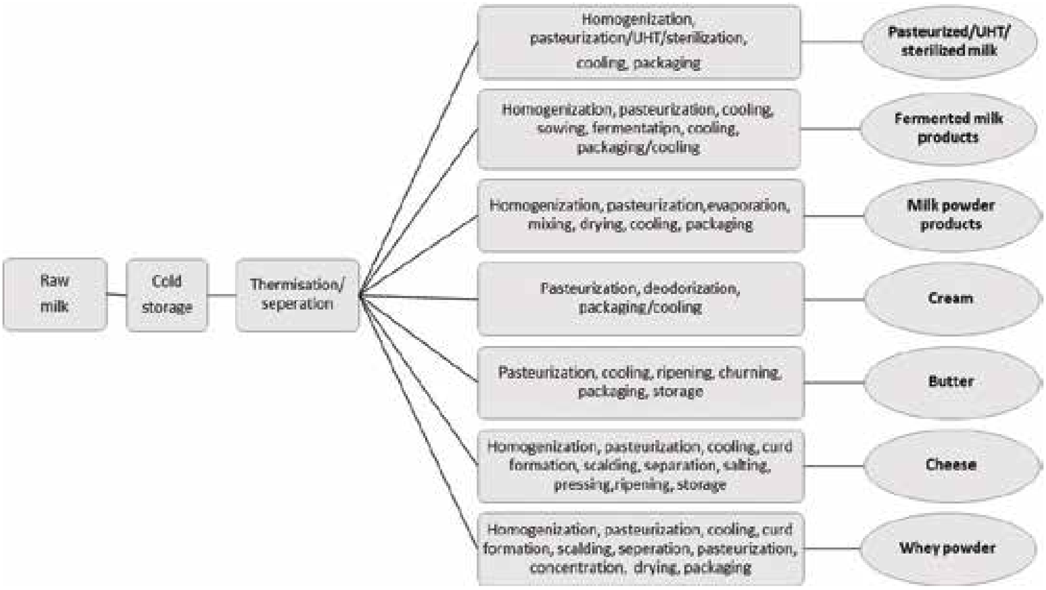

Figure 1. The major dairy products and unit operations take place their processing line [3].

In many dairy processes, it is necessary to separate cream from milk to be able to standardize the milk. In this context, milk is fed to the centrifuge to obtain cream and skim milk, individually. Then, cream is processed into cream and butter production and skim milk goes through other production lines. Depending on product type, fat content of milk is standardized by mixing skim milk and fat for other production processes. Homogenization is commonly used operation in dairy industry. It is carried out to disrupt fat globules of milk into smaller ones. Homogenizers are a kind of high pressure pump which forces milk through a close-fitting hole. This process is mostly used for long-life liquid milk production, fermented milk products, milk powder and condensed milk production.

Heat treatment (pasteurization and sterilization) is the most important and used process in dairy industry to kill unwanted microorganisms causing spoilage. During heat treatment, several changes occur depending on length and temperature of heating. For example, pasteurization can be applied low or high temperatures in which the former inhibits most of the microorganisms and some enzymes' activity with a condition of almost 74°C, 15 s. On the other hand, the latter is generally carried out at higher temperatures (e.g., 90°C, 15 s) by killing all vegetative microorganisms and inactivating most enzymes. The other heat treatment, sterilization (e.g., 20 min at 118°C), kills all microorganisms including spores leading to some undesirable changes in the milk. To be able to minimize these changes, ultrahigh temperature heating (UHT) was developed to sterilize milk at higher temperature and lower time compared to sterilization (e.g., at 145°C for a few seconds). Evaporation is another unit operation applied to remove water from milk to obtain more concentrated product. The other process called as drying is used to manufacture dairy product in powder form [1, 2].

Nowadays, energy shortage and carbon emission increase in the atmosphere are two major problems facing human kind. For that reason, the need to control and improve energy-consuming process and decrease emissions is paramount. As explained above, the production of dairy products needs significant amount of energy since most of the production lines have heating and cooling applications beside electricity. Recently, there is an increase attention to utilize energy in more efficient way and decrease energy consumption because energy costs are increasing depending on fossil fuel shortage and rise in energy prices affects mainly energy-consuming sectors including dairy industry. Despite the fact that energy sector is in the search of renewable energy sources, dependence on fossil fuels is of still concern.

A thermodynamical analysis is performed according to the first and second law of thermodynamics. The first law deals with conservation of energy which means that the input of energy equals the output of energy at steady-state condition, and energy can be converted from one form to another or transferred due to mass, heat and work transport [6, 7]. It should be noted that different types of energies (e.g., heat, work) have different work capacity or quality. For example, if the quality of energy is high, its work capacity would be high. However, the first law does not take into account quality of energy, and it says nothing about how to decrease energy consumption in the system of interest. In other respects, the second law of thermodynamics concerns about quality of energy and exergy or availability term appears which can simply be defined as available energy.

In reality, all processes are irreversible and result in entropy production due to friction, diffusion, etc. and the second law of thermodynamics states whenever energy is transferred due to heat, internal irreversibilities occur. Since irreversibilities lead to exergy destruction or exergy loss in the system, it is not possible to have 100% efficiency from a system [8]. The exergy analysis or second law analysis is performed to indicate irreversibilities in the system where the improvements can be made to decrease exergy losses [9–12]. It should be noted that availability or maximum useful work terms represent uppermost level of the work that can be obtained from a system without violating the laws of thermodynamics.

The value of actual work and maximum useful work are not equal to each other except ideal cases. Then, all efforts are to improve the system conditions and catch the minimum difference between availability and actual work value. It is important to realize that the availability of a system in a definite state is strictly attached to the reference conditions (surrounding) as well as properties of the system. In the analysis of availability, firstly initial state is determined and then the work obtained from the system has to be maximized as much as possible in a given context. The entire system has to be in the reference environment at the end of the process. When a system is in equilibrium with its surrounding, it has zero useful work (availability). The thermodynamic equilibrium is classified into three groups: thermal, chemical and mechanical equilibrium [7, 8]. Therefore, availability is an extensive property depending on the system and environment conditions.

In the literature, exergy analysis was conducted on different systems from machine system to fuel cells and biological systems by various authors. For example, Esen et al. [13] investigated energy and exergy analysis of a ground-coupled heat pump system. They showed that increase in heat source (ground) temperature leads to increase in energetic and exergetic efficiencies of the system. In another study, they studied performance assessment of a ground heat pump system beside technoeconomic analysis [14]. Taner and Sivrioglu [15] performed exery and thermoeconomic analysis of sugar factory having power turbine plant to improve energy economy of the production line, and their results showed that the effect of mass and energy on economic cost was found significant, and it is possible to increase total energy yield depending on design parameters and scenarios. They also studied a model sugar factory to obtain the best energy and exergy efficiency of the system [16]. Taner [17] assessed energetic and exergetic performance of PEM fuel cell to improve efficiency. Taner [18] also optimized drying plant energy and exergy efficiencies by changing mass and energy balance. On the other hand, exergy analysis was conducted on some biological systems on the cellular level [19, 20] to determine exergetic efficiency of metabolic pathway. In case of food industry, Genc and Hepbasli studied exergy analysis of potato crisp processes, and they found that the increase in mass flow rate of potato resulted in a rise in the fryer's exergy destruction rate [21]. Genc studied industrial grape molasses production to determine energy need to produce 1 kg of grape molasses [22]. Genc et al. [23] applied exergy analysis to red wine processing line to assess system's performance. For dairy industry, Yildirim and Genc [24] performed thermodynamic analysis of a milk pasteurization process assisted by geothermal energy and they conducted on energy and exergy analysis of the milk powder production line [25]. Furthermore, they optimized operating conditions of the system in order to increase exergy efficiency [26]. Jokandan et al. [27] applied comprehensive exergy analysis of an industrial-scale yoghurt production plant. Erbay et al. [28] studied to optimize operating condition of white cheese powder production. Munir et al. [29] evaluated a milk powder plant as a case study located in New Zealand by applying energy and exergy analyses where the drier has the lowest value and the milk silo has the highest value. Sorgüven and Ozilgen [30] computed energy loss, carbon dioxide emission and exergy loss during flavored yoghurt production. Soufiyan et al. [31] performed exergy analysis of long-life milk production process based on the real factory data and yoghurt drink production line was assessed exergetically by Soufiyan and Aghbashlo [32].

Exergy is one of the keystones of performance assessment. As seen from abovementioned studies, it can be applied to all processes varying from industrial systems to living organisms. Exergy analysis provides to quantify exergy losses by indicating irreversibilities throughout the processes and systems. Nevertheless, it is necessary to express some basic concepts of thermodynamics before introducing the methodology of exergy analysis on food processes. To the best of author's knowledge, a detailed survey of energy and exergy analysis of dairy processes performed up to date have never been studied before in the open literature. Therefore, the aim of this study is to explain the basic of energy and exergy analysis methodology in food processes and to review some important studies on exergy analysis of dairy industry including milk pasteurization, milk powder process, flavored yoghurt production, yoghurt production and yoghurt drink process, comprehensively for the first time.

2. Methodology

2.1. Fundamentals of exergy

Exergy is known as the work potential or as a measure of available energy. Szargut et al. [8] explained exergy as follows: *Exergy is the amount of work obtainable when some matter brought to a state of thermodynamic equilibrium with the common components of the natural surroundings by means of reversible processes.* In contrast to energy, exergy is destroyed in all real world processes as entropy is produced. Exergy is not destroyed only in reversible processes. However, in reality, all the processes are irreversible, and exergy is lost during the process.

2.2. Reference environment (dead state)

The ability of work for a system is assessed according to a base state. This base state is called as reference, environment or dead state [8]. For example, if the system and the reference environment (reference state or dead state) are in equilibrium, no spontaneous change can occur in the system, and as a result, work cannot be produced. The characterization of reference environment depends on the equilibrium type. If the reference environment and the system are in thermomechanical equilibrium, reference state is characterized by temperature, pressure, height and velocity. If the system under consideration is in chemical equilibrium with reference environment, the reference environment is defined by temperature, pressure and chemical composition. Szargut et al. [8] offered reference environment with three types which are gaseous and solid reference found in the atmosphere and reference species dissolved in seawater. The exergy values of these reference substances are assumed as zero. The standard chemical exergies of pure reference species are listed in Szargut et al. [8].

2.3. Calculation of exergy of food

Food commodities are mainly composed of water, protein, fat, carbohydrate, fiber and ash, and their thermal properties are crucial to apply thermodynamic analysis of unit operations such as cooling, heating, freezing, drying, and so on. In this part, calculation of exergy of food produces will be explained step by step.

2.3.1. Specific heat of foods

Specific heat is a function of temperature and in the study of Choi and Okos [12], the model for specific heat determination of food components was presented as shown in **Table 1**.

The specific heat of food shows a large difference above and below freezing point. The specific heat is relatively constant with respect to temperature above its freezing point. On the other hand, it has a large decline below the freezing temperature [33].

Generally, the specific heat of a food above the freezing point is the sum of mass average of the specific heat capacities of the food components.

$$c_{p,food} = \sum_{i=1}^{n} c_{p,i} w_i \tag{1}$$

where $c_{p,i}$ and w_i are the specific heat capacity and mass fraction of the individual food components, respectively. If the food component has a complex structure and all of its ingredients are not known, the formula developed by Chen [33] is used to compute the specific heat of unfrozen food produce

Component	**Specific heat equation (c_p, J/(kg.K), T °C)**
Protein	$c_{p,protein} = 2.0082 \times 10^3 + 1.2089T - 1.3129 \times 10^{-3}T^2$
Fat	$c_{p,fat} = 1.9842 \times 10^3 + 1.4733T - 4.8008 \times 10^{-3}T^2$
Carbohydrate	$c_{p,carbohydrate} = 1.5488 \times 10^3 + 1.9625T - 5.9399 \times 10^{-3}T^2$
Fiber	$c_{p,fiber} = 1.8459 \times 10^3 + 1.8306T - 4.6509 \times 10^{-3}T^2$
Ash	$c_{p,ash} = 1.0926 \times 10^3 + 1.8896T - 3.6817 \times 10^{-3}T^2$
Water (−40°C ≤ T ≤ 0°C)	$c_{p,water} = 4.0817 \times 10^3 - 5.3062T + 9.9516 \times 10^{-1}T^2$
Water (0°C ≤ T ≤ 150°C)	$c_{p,water} = 4.1762 \times 10^3 - 9.0864 \times 10^{-2}T + 5.4731 \times 10^{-3}T^2$
Ice	$c_{p,ice} = 2.0623 \times 10^3 + 6.0769T$

Table 1. Specific heat of different food components as a function of temperature (−40°C ≤ T ≤ 150°C) [12].

$$c_{p,food} = 4190 - 2300w_s - 628w_s^3 \tag{2}$$

where w_s refers to the mass fraction of the solid components found in the food produce.

If food is frozen, Eq. (3) is used to calculate apparent specific heat [33]

$$c_{p,a} = 1550 + 1260w_s + \frac{w_s R T_0^2}{M_s T^2} \tag{3}$$

where R is ideal gas constant, T_o is freezing point of water (K), T is temperature of food (°C) and M_s is effective molar mass of food solids.

When effective molar mass (M_s) of the soluble solid is not known, Eq. (4) is used to estimate apparent specific heat capacity

$$c_{p,a} = 1550 + 1260w_s - \frac{(w_{w0} - w_b)L_0 T_f}{T^2} \tag{4}$$

In this equation, w_s, w_{w0} and w_b are mass fraction of solid, ice and free water and bound water, respectively. L_0 is the latent heat of fusion of water and T is temperature of food in °C.

2.3.2. Enthalpy of foods

Specific enthalpy consists of sensible energy, and it is derived by integrating expressions of specific heat capacity with respect to temperature above the freezing point [34].

$$dh = c_p dT \tag{5}$$

The enthalpy of food produce is the sum of mass average of the enthalpy of the food components as shown in Eq. (6).

$$h_{food} = \sum_{i=1}^{n} h_i w_i = \sum \int c_i w_i dT \tag{6}$$

Below the freezing temperature, the equation becomes [33]

$$h = h_f + \left(T - T_f\right)\left(4190 - 2300w_s - 628w_s^3\right) \tag{7}$$

where h_f is the specific enthalpy at the initial freezing point and T and T_f are temperature and initial freezing temperature of food, respectively.

2.3.3. Entropy of foods

Specific entropy is derived by integrating expressions of specific heat capacity with respect to temperature as shown below

$$ds = c_p \frac{dT}{T} \tag{8}$$

The enthalpy of food produce is the sum of mass average of the entropy of the food components

$$s_{food} = \sum_{i=1}^{n} s_i w_i \tag{9}$$

where s is the entropy of food and w_i is the mass fraction of the components [8].

2.3.4. Calculations

Exergy of a product is composed of its physical, chemical, potential and kinetic exergy [8].

$$Ex_{total} = Ex_{ph} + Ex_{ch} + Ex_{pot} + Ex_{kin} \tag{10}$$

However, thermal exergy which is composed of physical and chemical exergy is important from engineering thermodynamics point of view and also food engineering applications.

2.3.4.1. Physical exergy

The physical exergy is the maximum useful work obtained from a system by passing the unit of mass of a component of its state to the environmental state (T_0, P_0). If potential and kinetic exergy is ignored, the physical exergy value is calculated by using enthalpy and entropy value of the component as follows:

$$ex_{ph} = [(h(T,P) - h_0(T_0,P_0)] - T_0[(s(T,P) - s_0(T_0,P_0)] \tag{11}$$

where the subscript zero refers to reference environment at P_0 and T_0 (in K).

If the enthalpy and entropy values of the food components are unknown, the enthalpy and entropy differences can be calculated and placed to the equation above (Eq. (11)) [8].

In the food engineering operations, water and air are mostly used components for drying, cooling, washing, and so on. For that reason, it is necessary to mention about their exergy calculations besides food commodities. In the case of enthalpy calculation of pure or homogeneous mixture (e.g., water), enthalpy equation is as follows:

$$dh = c_p dT + v dP \tag{12}$$

and the entropy term is shown in Eq. (13)

$$ds = c_p \frac{dT}{T} \tag{13}$$

After necessary derivation, the physical exergy of substances can be computed rough as follows:

$$ex_{ph} = c_p(T - T_0) - T_0 c_p ln\frac{T}{T_0} + v(P - P_0) \tag{14}$$

This equation might be applied for incompressible condensed substance for small temperature and pressure differences.

In the case of ideal gas substance (pure or constant composition, e.g., air), the temperature-dependent enthalpy difference is calculated by the formula

$$h - h_0 = \int_{T_0}^{T} c_p dT \tag{15}$$

For the entropy, the difference is explained as below

$$s - s_0 = \int_{T_0}^{T} c_p \frac{dT}{T} - R\int_{P_0}^{P} \frac{dP}{P} = \int_{T_0}^{T} c_p \frac{dT}{T} - R\ln\frac{P}{P_0} \tag{16}$$

If c_p is assumed constant, the physical exergy equation is as follows

$$ex_{ph} = c_p(T - T_0) - T_0 c_p ln\frac{T}{T_0} + RT_0 ln\frac{P}{P_0} \tag{17}$$

The exergy of air is calculated from

$$\begin{aligned} ex_{air} = {} & \left(c_{p,air} + \omega{*}c_{p,v}\right)(T - T_0) - T_0\left\{\left(c_{p,air} + \omega{*}c_{p,v}\right)ln\frac{T}{T_0} - (R_{air} + \omega R_v)ln\frac{P}{P_0}\right\} \\ & + T_0\left\{(R_{air} + \omega R_v)ln\left(\frac{1 + 1,6078\omega_0}{1 + 1,6078\omega}\right) + 1,6078\omega R_{air}\ln\left(\frac{\omega}{\omega_0}\right)\right\} \end{aligned} \tag{18}$$

where ω is the specific humidity ratio is.

The exergy of water

$$ex_{water} = \left(h_f - h_{g,0}\right) + v_f\left(P - P_g\right) - T_0\left(s_f - s_{g,0}\right) + T_0 R_v ln\left(\frac{P_{g,0}}{x_{v,0}P_0}\right) \tag{19}$$

2.3.4.2. Chemical exergy

The standard chemical exergy values of most of the elements are given by Szargut et al. [8]. Chemical exergy of a species at reference state is the sum of the standard Gibbs free energy of formation and standard chemical exergies of its elements

$$ex^0_{ch,j} = \Delta G^0_{for,j} + \sum v_k Ex^0_{ch,k} \tag{20}$$

where v_k is the stoichiometric constant of kth element in the species. Calculation of chemical exergy of species at a definite state is

$$ex_{ch,j} = \Delta G_{f,j}^{T} + \sum v_k Ex_{ch,k}^{0} \tag{21}$$

Following equation describes how the chemical exergy of a stream is calculated:

$$ex_{ch} = \sum x_i Ex_{ch,i} \tag{22}$$

where x_i is the mass fraction of ith element in the species.

For industrial fuels, an approximate calculation of chemical exergy might be computed by using lower heating value (LHV) and higher heating value (HHV).

$$ex_{ch} = f_l LHV = f_h HHV \tag{23}$$

where f_l and f_h are empricial coefficients for LHV and HHV, respectively.

2.3.4.3. Exergy of non-matter streams

2.3.4.3.1. Exergy of work

Exergy of work equals to its value.

$$E\dot{x}_{work} = \dot{W} \tag{24}$$

2.3.4.3.2. Exergy of electricity

The energy content of electricity is directly converted to work and the exergy of electricity

$$E\dot{x}_e = \dot{W}_e \tag{25}$$

2.3.4.3.3. Exergy of heat flows

The exergy content of a heat flow $\dot{Q}$ at a temperature is of

$$E\dot{x}_{heat} = \left(1 - \frac{T_0}{T}\right)\dot{Q} \tag{26}$$

$\dot{Q}$ and T are the heat transfer rate from the system to the surrounding and temperature of surrounding.

2.4. Exergy analysis methodology

A general balance equation for a control volume is written as follows

$$Input - Output + Generation - Consumption = Accumulation \tag{27}$$

This general balance equation is applied for mass, energy, entropy and exergy. In the general equation, input and output refer to energy or matter transfer from the surrounding to the control volume and from control volume to surrounding, respectively. Generation and consumption indicate the quantity generated and consumed in the control volume, respectively. Accumulation term defines accumulated quantity in the control volume.

2.4.1. Mass balance

In the general steady-state mass balance equation (Eq. (26)), the mass enters to the system equals to the mass exits from the system and the rate form is as follows:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{28}$$

2.4.2. Energy balance

The first law of thermodynamics deals with conservation of energy, which means that the input of energy equals the output of energy and energy conservation law is shown below for steady-state sytems

$$\sum \dot{E}_{in} = \sum \dot{E}_{out} \tag{29}$$

The second law of thermodynamics can be defined by Clausius equation as follows:

$$\sum (\dot{m}s)_{out} - \sum (\dot{m}s)_{in} = \sum_j \left(\frac{\dot{Q}}{T}\right)_j + \ \dot{s}_{gen} \tag{30}$$

where s is entropy, $\dot{Q}$ is heat transfer rate through the system boundary and $\dot{s}_{gen}$ is entropy production rate due to irreversibility. The entropy term is a physical property of substances, and it is related to availability or useful work. Reversible processes are also called as ideal processes and entropy production of these processes is assumed to be zero [6]. In reality, it can never happen, and entropy always increases and all studies about energy efficiency have focused on keeping the entropy in its minimum level.

If we combine the first (Eq. (27)) and second law of thermodynamics (Eq. (28)), the equation becomes

$$\dot{W} = \sum_n \dot{Q}_n \left(1 - \frac{T_0}{T_n}\right) + \sum \dot{m}_{in}\left(h - T_0 s + \frac{w^2}{2} + gz\right) - \sum \dot{m}_{out}\left(h - T_0 s + \frac{w^2}{2} + gz\right) - T_0 \dot{s}_{gen} \tag{31}$$

which is known as the Gouy-Stodola equation [8], and it is the base of the exergy balance equation.

2.4.3. Exergy balance

For a steady-state system, the exergy rate equation is shown below:

$$\sum \dot{Ex}_{out} - \sum \dot{Ex}_{in} = \sum \dot{Ex}_D = T_0\ \dot{s}_{gen} \tag{32}$$

If we write the equation in terms of mass, heat and work form it becomes

$$\dot{Ex}_{mass,\,in} - \dot{Ex}_{mass,\,out} + \dot{Ex}_{heat} - \dot{Ex}_{work} = \dot{Ex}_D \tag{33}$$

The exergy of the mass is defined as

$$\dot{Ex}_{mass,\,in} = \dot{m}_{in} ex_{in} \tag{34}$$

$$\dot{Ex}_{mass,\,out} = \dot{m}_{out} ex_{out} \tag{35}$$

where ex is the specific exergy, $\dot{m}$ is the mass flow rate.

2.4.4. Energy and exergy efficiency

In the conservation of energy, only quantity of energy (kinetic, potential, thermal, mechanical, internal) is taken into account, and the energy efficiency of a system is defined as the ratio of energy released by the system to energy provided to the system.

$$\eta = \frac{Energy\ released\ by\ the\ system}{Energy\ provided\ to\ the\ system} \tag{36}$$

where η refers to energy efficiency. However, the first law does not say anything about how to decrease energy consumption in the system of interest. It should be noted that different types of energies have different work ability or quality [6].

The exergetic performance of the processes is evaluated by means of different criteria. One of them is called universal efficiency which is defined as the ratio of exergy output to exergy input.

$$\varepsilon_u = \frac{Exergy\ output}{Exergy\ input} \tag{37}$$

The other one is called as functional exergy efficiency and is defined as the ratio of the exergy of the desired output to the exergy of fuel or raw material to produce desired product.

$$\varepsilon_f = \frac{Exergy\ of\ the\ desired\ output}{Exergy\ of\ raw\ material} \tag{38}$$

2.4.5. Exergetic performance indicators

For a process or system, if the exergy loss or irreversibility is minimized, the maximum improvement obtained. Van Gool [35] developed an equation for improvement potential and this relation is as follows

$$I\dot{P} = (1 - \varepsilon)(\dot{Ex}_{in} - \dot{Ex}_{out}) \tag{39}$$

where $I\dot{P}$ is the improvement potential in the rate form.

It should be noticed that if we deal with complex food processes, it is necessary to divide the processes into several units and assess exergy destruction for each unit to determine which unit operation is more inefficient in terms of thermodynamics. In addition to improvement potential, some other performance indicators such as relative irreversibility, exergetic factor, sustainability index used in exergy analysis defined as follows [36]:

$$\text{Improvement potential rate: } \dot{IP} = (1-\varepsilon)\left(\dot{Ex}_f - \dot{Ex}_p\right) \tag{40}$$

$$\text{Relative irreversibility: } RI_k = \left(\dot{Ex}_{D,k}/\dot{Ex}_{D,\,tot}\right) \times 100 \tag{41}$$

$$\text{Exergetic factor: } f_k = \left(\dot{Ex}_{f,k}/\dot{Ex}_{f,\,tot}\right) \times 100 \tag{42}$$

$$\text{Sustainability index: } SI = 1/(1-\varepsilon) \tag{43}$$

3. Applications of exergy analysis on dairy processes

Up to know, exergy analysis has been applied most of the engineering processes. In last decades, exergetic assessments of dairy processes have an increase attention. In this section, some of the examples of exergetic assessment of dairy processes are tabulated (**Table 2**) and briefly summarized.

Soufiyan et al. [31] studied exergetic performance assessment of a long-life milk processing plant. The investigated plant, shown in **Figure 2**, consisted of four main lines as steam generator, above-zero refrigeration system, milk reception, pasteurization, and standardization line, and ultra-high-temperature (UHT) milk processing unit while the main components of the steam generator line are the condensate tank, shell and tube heat exchanger, fire tube boiler with a fuel injection system and an air compressor, pressure reducing valves, pumps and steam trapping devices; the gasket plate heat exchanger, storage tanks with agitators, balance tank, deaerator with a vacuum pump, flow controller, centrifugal fat extraction apparatus, mixing devices, homogenizer, bactofuge, holding tube are the main components of the milk reception, pasteurization, and standardization line.

Study	Authors	Year
Long-life milk processing	Soufiyan MM, Aghbashlo M, Mobli H	2017
Industrial-scale yoghurt production	Jokandan MJ, Aghbashlo M, Mohtasebi SS	2015
Yoghurt drink production	Soufiyan MM and Aghbashlo M	2017
Cheese powder production	Erbay Z, Koca N, Kaymak-Ertekin F, Ucuncu M	2015
Flavored yoghurt production process	Sorgüven E and Özilgen M	2012
Milk pasteurization process	Yildirim N and Genc S	2015
Milk powder production	Yildirim N and Genc S	2017

Table 2. The studies on exergetic assessment of dairy processes in open literature.

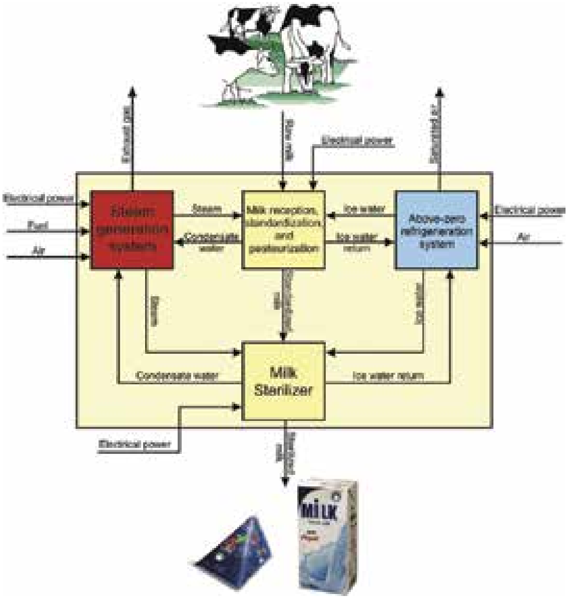

Figure 2. General description of the long-life milk processing plant of the investigated factory [31].

In the system of interest, the specific exergy destruction for production of the 1 kg-long-life milk was found to be 345.50 kJ/kg. While the highest contribution to the specific exergy destruction of the long-life milk processing (60.70%) belonged to the steam generation system, the specific exergy destruction of the milk reception, pasteurization, and standardization line, and UHT milk-processing unit were determined as 1.63 and 49.54 kJ/kg, respectively.

A detailed exergetic analysis of an industrial yoghurt production plant is conducted by Jokandan et al. (**Figure 3**) [27]. The percentile contributions for yoghurt production lines was obtained as 5.21% of the total specific exergy consumption of the pasteurized yoghurt production plant with the value of 841.34 kJ/kg to produce 3.75% fat yoghurt with 1.46 kg/s mass flow rate.

In a similar study of Soufiyan and Aghbashlo, 2017 [31] yoghurt drink manufacturing was investigated by using the same milk processing line as explained above (**Figure 2**). The specific exergy destruction of yoghurt drink manufacturing line was determined as 118 kJ/kg which is 26.7% of whole system's specific exergy destruction of 442 kJ/kg (**Figure 4**).

Erbay et al. [28] studied optimization of pilot-scale cheese powder production by using a pilot-scale spray drier and nonenzymatic browning index, free fat content, solubility index, bulk density and exergy efficiency values were determined at the optimum condition. Optimum operating conditions were provided at 174°C inlet and 68°C outlet drying temperature with 354 kPa atomization pressure of 354 kPa. The exergetic efficiency was obtained as 4.81% at the optimum operating condition as seen in **Figure 5**.

Sorguven and Ozilgen [30] investigated energy utilization, carbon dioxide emission, and exergy loss in flavored yoghurt production process (**Figure 6**). The results show that 53% of the total exergy loss occurred during the milk production and 80% of the total work input

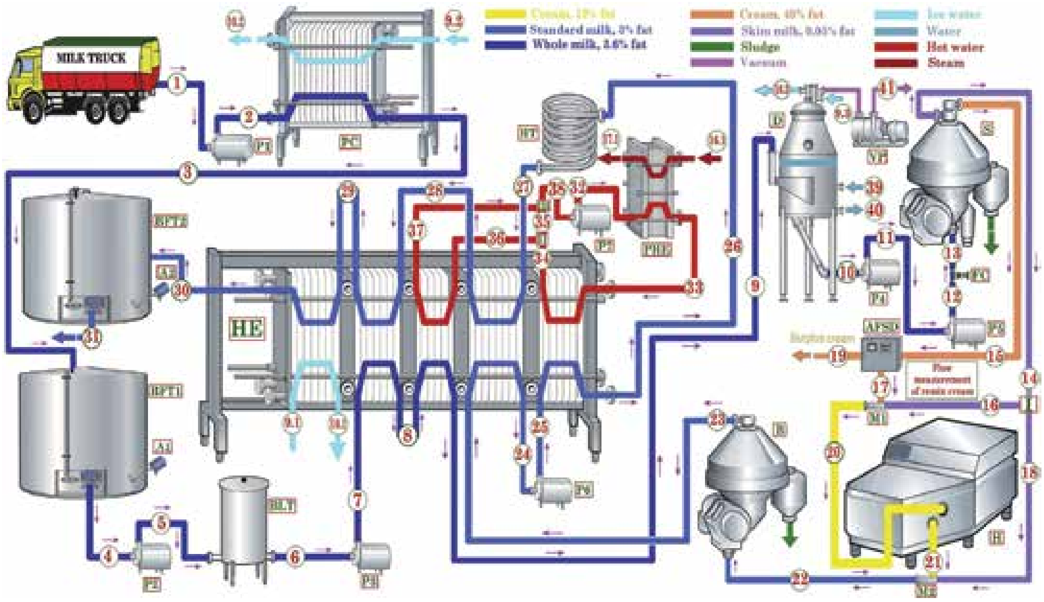

Figure 3. Schematic illustration of the yoghurt production line of the investigated dairy company [27].

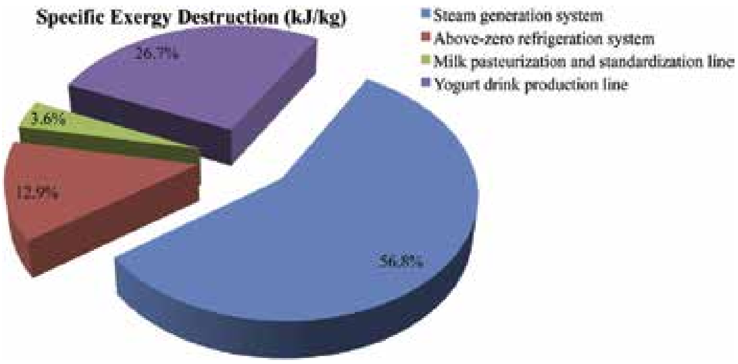

Figure 4. Specific exergy destruction fraction of yoghurt drink manufacturing factory [31].

was consumed during the plain yoghurt making process. The cumulative degree of perfection was 3.6% for the strawberry-flavored yoghurt. This value can rise up to 4.6%, if renewable energy resources like hydropower and algal biodiesel are employed instead of fossil fuels.

Thermodynamic analysis of a milk pasteurization process assisted by geothermal energy (illustrated in **Figure 7(a)**) was studied by Yildirim and Genc, 2015 [24]. In the system, a water-ammonia VAC (vapor absorption cycle), a cooling section, a pasteurizer and a regenerator were used for milk pasteurization. The universal and functional exergetic efficiencies were considered in the study. While the exergetic efficiency of the whole system was calculated as 56.81% with total exergy destruction rate of 13.66 kW, the functional exergetic efficiency of pasteurization was

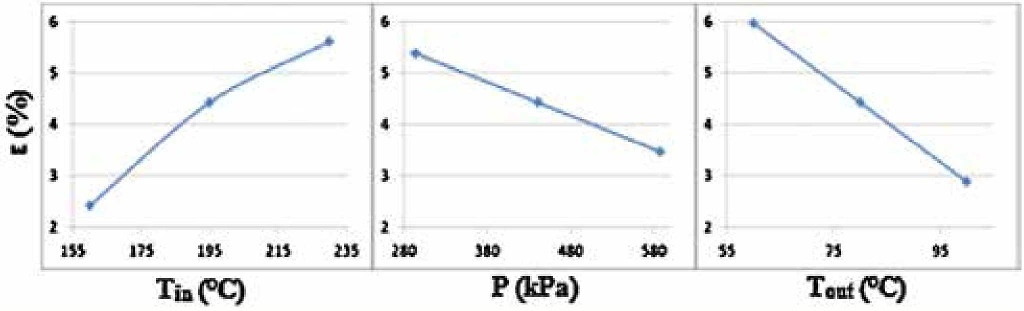

Figure 5. Exergy efficiencies at the optimum conditions [28].

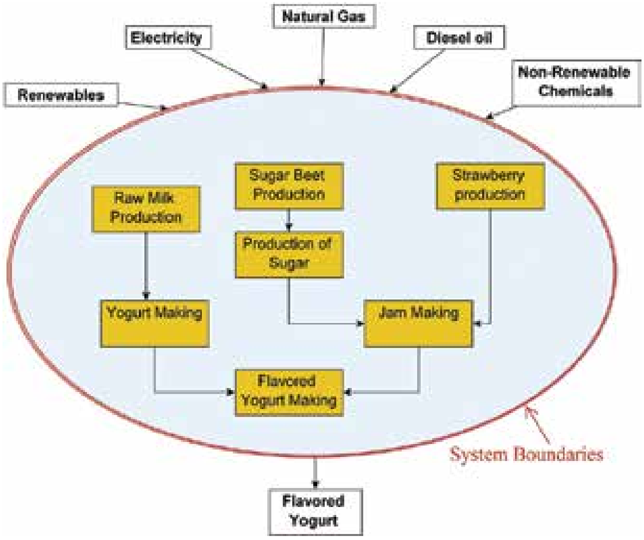

Figure 6. Schematized diagram of overall system [30].

determined as 25.8%. The effect of geothermal resource temperature and geothermal resource flow rate on the exergetic efficiency of the whole system for 1 kg/s mass flow rate of milk was investigated. The results showed that the maximum efficiency can be obtained by using lower temperature and higher flow rate of the geothermal resource (**Figure 7(b)**).

Yildirim and Genc [25] studied a thermodynamic analysis including comprehensive exergy analysis by using different performance parameters such as exergy efficiency, improvement potential rate, sustainability index, relative irreversibility and exergetic factor for the milk powder production system. The schematic diagram of the milk powder production line is illustrated in **Figure 8**. The considered system consists of an evaporator, a feed pump, a spray drier, a cooler, an economizer, a fan, a heater, and a compressor. Thermal energy needed for the production line was provided by geothermal energy.

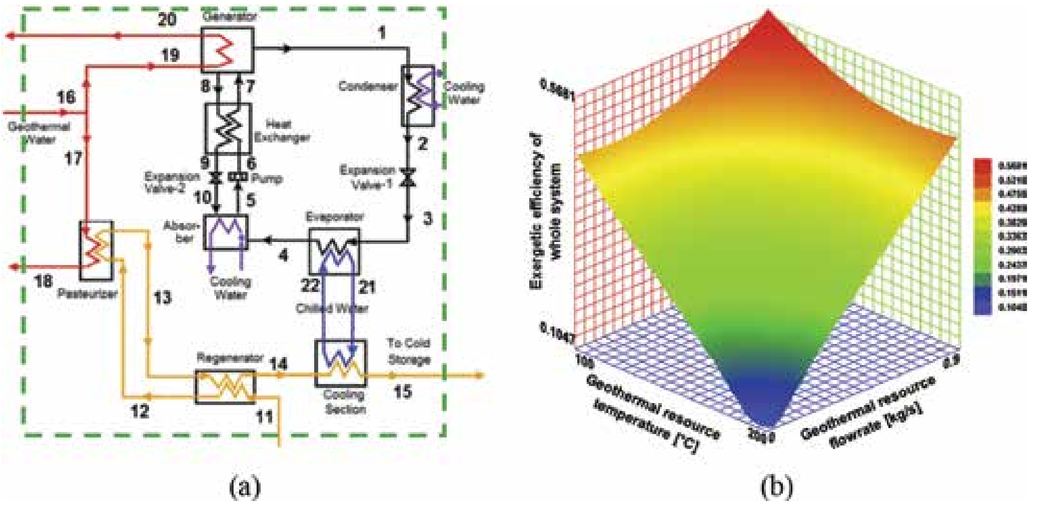

Figure 7. (a) Schematic of the system of a milk pasteurization process assisted by geothermal energy and (b) the effect of geothermal resource temperature and geothermal resource flow rate on the exergetic efficiency of the whole system [24].

The overall energy and exergy efficiencies of the whole milk powder production system were calculated as 85.4 and 57.45%, respectively. In the considered system, the evaporator has the highest (333.60 kW) exergy destruction rate and relative irreversibility (75.8%). Therefore, the evaporator has the highest improvement potential rate $I\dot{P}$ (191.1 kW). Grassman diagram is mostly used to represent exergy streams graphically. In the diagram, the width of each arrow is scaled according to their magnitude and in this way, all exergy destructions take place in the system can be seen clearly with their values. As an example, in the study of Yildirim and Genc

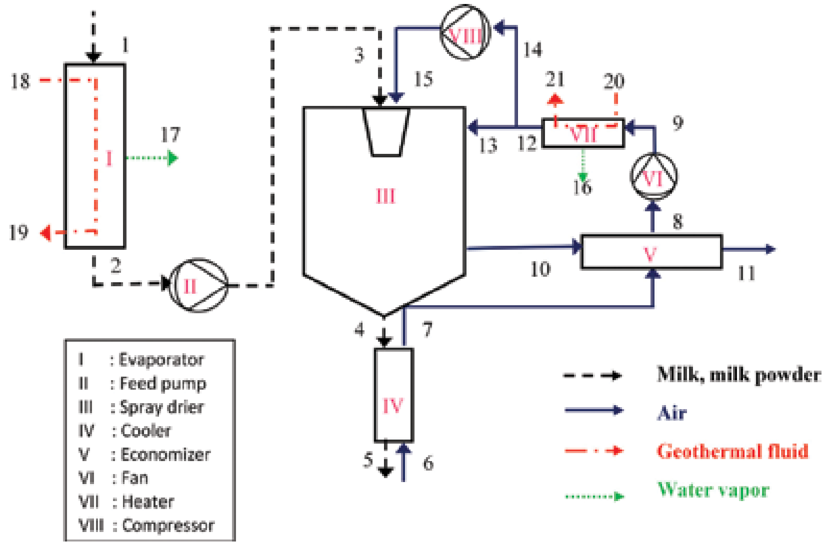

Figure 8. Schematic diagram of the milk powder production line [25].

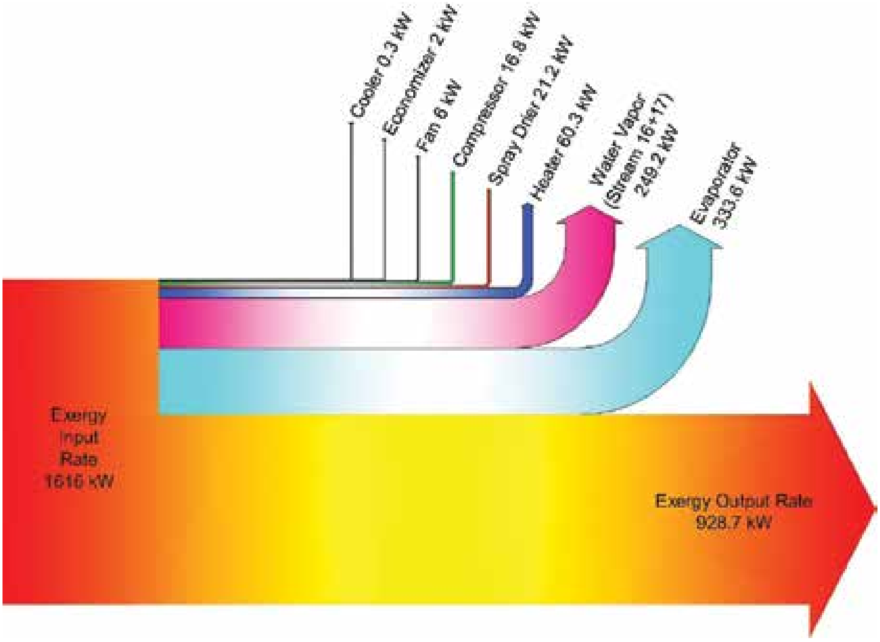

Figure 9. Grassmann diagram of the milk powder production process assisted by geothermal energy [25].

[25], exergy fluxes and destructions in the milk powder system are shown in the Grassman diagram in **Figure 9**.

4. Conclusion

The first law analysis is widely used for the assessment of many systems from living organisms to machine system. Recently, a number of exergy analysis applications have been increasing since it is a more useful tool to assess the performance of systems and support sustainable development. In this chapter, exergy analysis methodology of food processes was presented, and some important exergy analysis applications (e.g., yoghurt production, milk pasteurization process, milk powder production) of dairy industry were reviewed. Although, the exergy analysis provides to design more efficient and sustainable approaches to food industry, it has still been is at very beginning stage. Besides, thermodynamic optimization can be conducted on dairy processes to determine the best performance in terms of exergetic perspective. As a result of energy and exergy analysis, utilization of renewable energy sources becomes prominent due to high energy need of dairy industry. Furthermore, exergoeconomic analysis that integrates economics with exergy can be a useful tool to assess the effect of both economics and thermodynamics. Additionally, environmental impact of dairy industry can be determined by life cycle assessment and exergetic life cyle assessment which takes into consideration both exergy and life cycle assessment.

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Conflict of interest

There are no conflicts of interest to declare. SG is the author of the manuscript and no one else contributed to the design and writing of the manuscript.

Nomenclature

c_p	specific heat capacity (kJ/kg K)
w	mass fraction (–), velocity (m/s), specific humidity (kg H_2O/kg dry air)
ex	specific exergy (kJ/kg)
$\dot{E}$	energy rate (kW)
Ex	exergy (kJ)
$\dot{Ex}$	exergy rate (kW)
f	empricial coefficients, exergetic factor (%)
h	specific enthalpy (kJ/kg)
$\dot{IP}$	improvement potential rate (kW)
$\dot{m}$	mass flow rate (kg/s)
$\dot{Q}$	heat transfer rate (kW)
P	pressure (kPa)
R	gas constant (kJ/kg K)
RI	relative irreversibility (%)
s	specific entropy (kJ/kg K)
SI	sustainability index (–)
T	temperature (K or °C)
$\dot{W}$	rate of work or power (kW)
M	effective molar mass (kg/mol)

L	latent heat (kJ/kg)
x	mole fraction (–)

Greek letters

η	thermal (the first law) efficiency (–)
ε	exergetic (the second law) efficiency (–)
v	specific volume (m^3/kg)

Subscript

a	apparent
air	air
ch	chemical
D	destruction, destroyed
e	electricity
f	freezing, functional, fuel
for	formation
g	gas
gen	generation
heat	heat
in	input, inlet
k	location
kin	kinetic
out	output
p	product
past	pasteurizer
ph	physical
pot	potential
s	solid
tot	total
u	universal
v	vapor

water	water
work	work
0	dead/reference/environmental state

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Section 3

Thermoeconomic Analysis

Advanced Exergy Analysis of an Integrated SOFC-Adsorption Refrigeration Power System

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Abstract

In this chapter, an exergy analysis applied to a solid oxide fuel cell (SOFC)/vapor adsorption refrigeration (VAR) system is presented. The influences of four significant parameters (current density, inlet fuel temperature, fuel utilization and steam-to-carbon ratio) on the exergy efficiency of both the SOFC stack and the SOFC-VAR system are investigated. In order to do so, a mathematical model is constructed in Engineering Equation Solver (EES) software to generate the simulations. The analysis shows that the calculated exergy efficiency is around 8% lower than the energy efficiency for both cases. Moreover, it is found that most of the causes of irreversibilities in the system are due to electronic and ionic conduction in the components. It is also shown that the exergy efficiency is substantially sensitive to fuel inlet temperature, which is evidenced by a bending-over behavior. Finally, in accordance with the calculated efficiency defects, the main exergy destructions are present in the heat exchangers, the SOFC, the afterburner and the generator.

Keywords: SOFC, adsorption, exergy, efficiency defect, current density

1. Introduction

Recent research developments on alternatives to generate electricity are being directed to leading-edge technologies such as solid oxide fuel cells (SOFC). A fuel cell is considered as a highly efficient, environmentally friendly device to generate affordable energy [1, 2]. It is well-known that a solid oxide fuel cell converts the chemical energy of a fuel into electrical energy by means of an electrochemical reaction at high operating temperatures (600–1000°C). It is this type of reaction which makes it a more efficient way to produce electricity than a conventional steam engine that depends on a quite irreversible combustion reaction. Hence solid oxide fuel cells are attracting considerable attention from worldwide researchers.

At present, most of the studies are focusing on the development of multiproduct power generation systems to enhance significantly the overall efficiency of the system [3–7]. Furthermore, developments of hybrid systems are being expanded to run other power generation systems as trigeneration systems [8, 9], steam turbines [10, 11] and gasification [12]. In this context, a comparative energy and exergy analysis of an SOFC/GT waste heat to power conversion employing Kalina and Organic Rankine cycles is reported [13]. The study reports an exergy efficiency of 62.35% for the combined SOFC/GT-ORC system and 59.53% for the combined SOFC/GT-KC system. In another study, a new solar-based multi-generation system integrated with ammonia fuel cell and solid oxide fuel-cell-gas turbine combined cycle reports an energy and exergy efficiency increase of up to 19.3 and 17.8%, respectively, in comparison to single generation systems [14]. It is also an interesting study performed to determine the effect that the anode and cathode SOFC-stack recycling gas has on both the thermodynamic and thermoeconomic performance of a proposed cooling, heating and power (CCHP) system. Therein, the results show that the total energy efficiency of the trigeneration system with anode gas recycle (Tri-SOFC-AR) is 6% larger than that of a simple case [15].

Furthermore, the energy and exergy analysis is also applied to hybrid combined cooling, heating and power (CCHP) plant coupled with a molten carbonate fuel cell (MCFC) and Stirling engine [16]. In this system, the modeling and simulation show overall energy and electrical efficiencies of 71.7 and 42.28%, respectively. Micro combined power systems are also drawing particular attention from researchers. For example, a study of a micro combined cooling heating and power (CCHP) system based on high-temperature proton exchange fuel cell (PEMFC) reports an overall efficiency of 47% under winter and normal operating conditions [17].

Some researchers are also making many studies concerning the use of alcohol fuels in SOFC integrated systems. Alcohol fuels such as methanol and ethanol are being considered as promising alternative fuels since they are fluid and some of their chemical and physical properties are similar to gasoline [3, 18]. Tippawan et al. [19] investigated the influence of changing the current density, SOFC temperature, fuel utilization and SOFC anode recirculation on the efficiency of heating cogeneration, cooling cogeneration and trigeneration for an ethanol-fueled integrated SOFC system. Therein, the trigeneration exergy efficiency increased 32% in comparison to conventional power cycles.

On the other hand, other investigations are focusing particularly on the optimization of integrated SOFC systems. Hosseinpour et al. [20] performed an exergy optimization of a cogeneration system based on a methane-fed solid oxide fuel cell (SOFC) integrated with a Stirling engine. Therein, the objective function is the exergy efficiency. In accordance with the study, the optimum value for the exergy efficiency is 56.44%. In a more complex study, a multi-objective optimization of a SOFC-GT power plant is performed [21]. In this particular case, the cost function value and the exergy efficiency are the objective functions. Through application of a fuzzy multi-objective method, the optimum point for the cost function value is 0.043 (US$/s) and for the exergy efficiency is 57.7%, approximately.

In this context, the present work aims to contribute to the analysis of possible affordable hybrid systems, so an exergy analysis of a solid oxide fuel cell/vapor adsorption refrigeration system is presented herein. The objective of this chapter is to investigate the influences of four significant parameters (current density, inlet fuel temperature, fuel utilization and steam-to-carbon ratio) on the exergy efficiency of both the SOFC stack and the SOFC-VAR system. In order to do so, a detailed model is constructed with the fundamental equations that govern the operation of the components. Special attention is paid to the components where most of the input exergy is destroyed. It is important to comment that the simulation is performed using engineering equation solver software. Furthermore, both SOFC and VARS models are based on reliable data and parameters obtained from a literature review. So models are calibrated and validated comparing results with data reported by, respectively, Tao et al. [22] and Herold et al. [23]. The simulation model provides mass, energy and exergy balances for each component of the system and calculates efficiency parameters such as the energy and exergy efficiency as well as the efficiency defects.

2. Energy system description

The schematic flow diagram of the integrated SOFC-Adsorption system considered herein is depicted in **Figure 1**. The energy system consists of an SOFC stack with internal reforming of feed gas at the anode side, an afterburner, a mixer, three pre-heaters and a DC/AC inverter. Anode and cathode exit streams are fed into the afterburner, the exhaust gas is then used to preheat the supply of fuel and air. The high-grade heat yielded in the SOFC reaction is used to perform the reforming process. In order to improve the overall efficiency of the SOFC, the exhaust gas from the stack enters a LiBr-H_2O-based vapor adsorption refrigeration system (VARS) coupled to it. **Table 1** presents the physical characteristics of an intermediate temperature, anode-supported planar SOFC as reported in [24].

For the sake of simplicity, several assumptions have been considered in the present analysis. The study is carried out under thermodynamic equilibrium and steady-state conditions (**Table 2**). Kinetic and potential energy effects are negligible. The assumptions are:

A. For the solid oxide fuel cell:

1. Air consists of 79% N_2 and 21% O_2.
2. All gases are considered as ideal gases.
3. Gas mixture at the fuel channel exit is at chemical equilibrium.
4. Fuel cell is completely insulated, so there is no heat interaction with the environment.
5. Contact resistances are negligible.

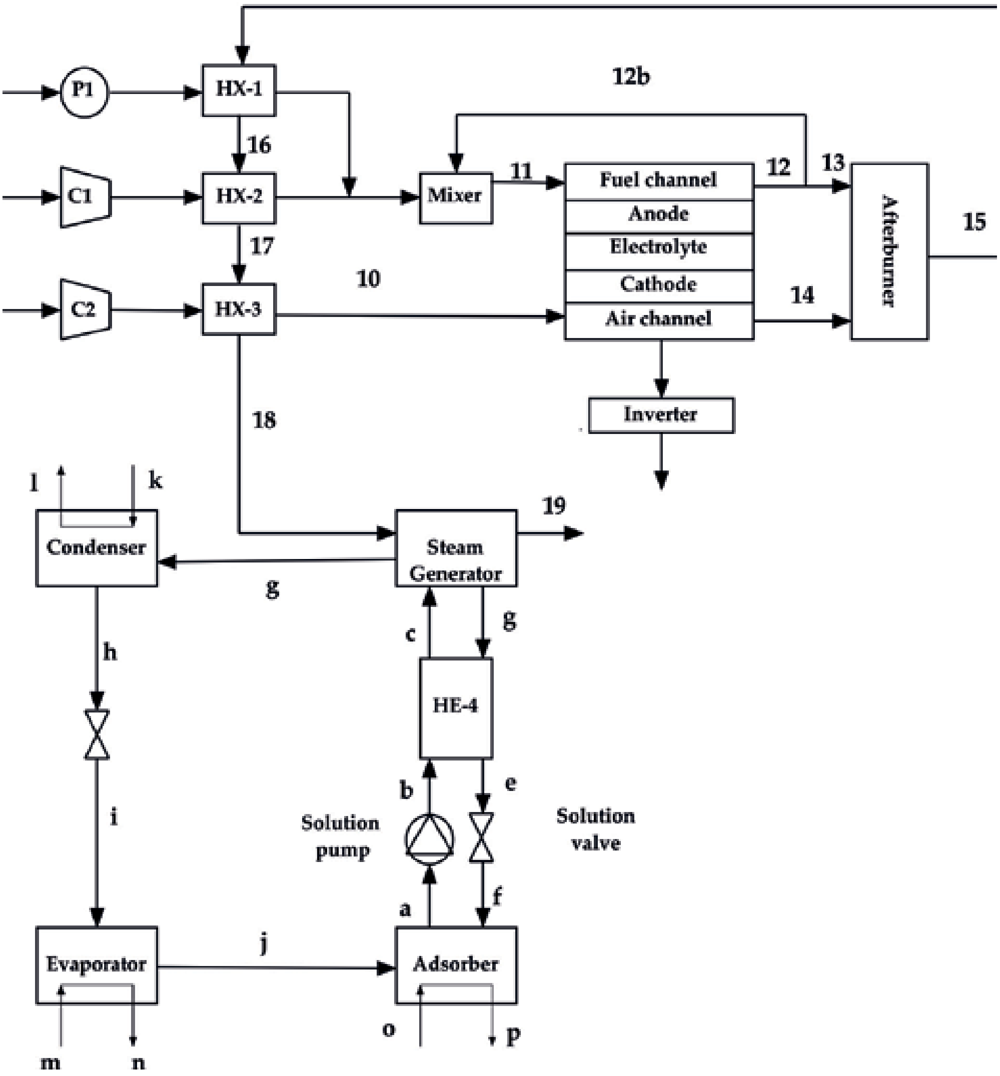

Figure 1. Schematic diagram of the integrated SOFC-VARS system.

6. Temperature at the channel inlets is the same. Also, temperature at the channel exit is the same.
7. Radiation heat transfer between gas channels and solid structure is negligible.

B. For the vapor adsorption refrigeration system [23]:

1. Water is considered as refrigerant (at states g-j).
2. States a and h are considered as saturated liquid.

3. Water is considered as saturated vapor at state j.
4. Pressure in generator and condenser are equivalent.
5. Pressure in the evaporator and absorber are equivalent.

Anode exchange current density (i_{oa})	0.65 A/cm^2
Cathode exchange current density (i_{oc})	0.25 A/cm^2
Effective gaseous diffusivity through the anode (D_{aeff})	0.2 cm^2/s
Effective gaseous diffusivity through cathode (D_{ceff})	0.05 cm^2/s
Anode thickness (L_a)	500 μm
Cathode thickness (L_c)	50 μm
Electrolyte thickness (L_e)	10 μm
Number of cells	11,000
Active surface area	0.01 m^2

Table 1. Characteristics of the SOFC as reported in [24].

Input data	
Temperature difference between SOFC inlet and outlet	100 K
Fuel cell inlet temperature	1000 K
Fuel utilization factor	85%
Steam-to-carbon ratio	2.5
Fuel cell pressure drop	2%
Heat exchangers pressure drop	3%
Afterburner pressure drop	5%

Table 2. Operating conditions of the SOFC stack.

3. SOFC mathematical modeling

3.1. Electrochemical model

Modeling of the electrochemical part can be as complicate as the study requires it, and most of the current literature provides basic models developed under the zero-dimensional assumption [6, 8, 25]. The main idea of such modeling is to have mathematical equations that mimic the connection between the chemical energy of the fuel and the electrical power. So, the mechanisms of reaction involved are:

For the steam reforming reaction:

$$CH_4 + H_2O \rightarrow 3H_2 + CO \tag{1}$$

For the shifting reaction:

$$CO + H_2O \rightarrow H_2 + CO_2 \tag{2}$$

Thus the net electrochemical reaction of the fuel cell is given as:

$$H_2 + \frac{1}{2}O_2 \rightarrow H_20 \tag{3}$$

Reforming and shifting reactions are carried out within the fuel cell stack, so the energy required for the reaction is directly supplied by the fuel cell as heat. The real velocity at which both chemical and electrochemical reactions are carried out are based on the following equilibrium reactions:

For the real reforming reaction:

$$X_r[CH_4 + H_2O \rightarrow 3H_2 + CO] \tag{4}$$

For the real shifting reaction:

$$Y_r[CO + H_2O \rightarrow H_2 + CO_2] \tag{5}$$

For the real net electrochemical reaction:

$$Z_r\left[H_2 + \frac{1}{2}O2 \rightarrow H_20\right] \tag{6}$$

where X_r, Y_r and Z_r represent the conversion ratios during the reactions.

The equilibrium constants for the reforming and shifting reactions can be formulated as a function of the operating temperature as follows:

$$Log_{10}K_{\mathrm{r,s}} = A_{r,\mathrm{s}}T^4 + B_{r,\mathrm{s}}T^3 + C_{r,\mathrm{s}}T^2 + D_{r,\mathrm{s}}T + E_{r,\mathrm{s}} \tag{7}$$

The values of the constants are given in **Table 3** as suggested in [26].

	Reforming	**Shifting**
A	-2.63121×10^{-11}	$5.4730\ 1 \times 10^{-12}$
B	1.24065×10^{-7}	-2.57479×10^{-8}
C	-2.25322×10^{-4}	4.63742×10^{-5}
D	1.95028×10^{-1}	-3.91500×10^{-2}
E	-66.1396	13.2097

Table 3. Constant values for the equilibrium constant equation.

The equilibrium constants can also be determined as a function of the molar fraction of each species as now described.

For the reforming reaction:

$$Ln(K_r) = \frac{\left(y^{CO}{}_{11}\right)\left(y^{H_2}{}_{11}{}^{3}\right)}{\left(y^{H_2O}{}_{11}\right)\left(y^{CH_4}{}_{11}\right)} * \left[\frac{P}{P_0}\right]^2 \tag{8}$$

For the shifting reaction:

$$Ln(K_s) = \frac{\left(y^{H_2}{}_{11}\right)\left(y^{CO_2}{}_{11}\right)}{\left(y^{CO}{}_{11}\right)\left(y^{H_2O}{}_{11}\right)} \tag{9}$$

Here, each equilibrium molar fraction of species i is represented by y^i_{eq}, and it can be written as a ratio between the equilibrium number of moles of species i and the total number of moles at equilibrium.

$$y^i_{eq} = \frac{\dot{n}^i_{eq}}{\dot{n}_{eq}} \tag{10}$$

On the other hand, the equations used in the modeling for calculating the maximum voltage achievable by the solid oxide fuel cell are:

$$V_{\text{SOFC}} = V_N - V_{\text{Loss}} \tag{11}$$

where V_N represent the Nerts voltage and V_{Loss} stands for the voltage losses. It is important to recall that after a SOFC delivers electrical current, its components exhibit a resistance which results in voltage losses. These voltage losses are generally classified as ohmic, activation and concentration polarization losses. Accordingly,

$$V_N = -\frac{\Delta g^o_{rxn}}{2F} + \frac{\overline{R}T_{o,SOFC}}{2F}\ln\left(\frac{a_{11}^{H_2} a_{14}^{O_2^{1/2}}}{a_{11}^{H_2O}}\right) \tag{12}$$

$$V_{Loss} = \Delta V_{ohm} + \Delta V_{act} + \Delta V_{conc} \tag{13}$$

For simplicity, the total ohmic losses can be evaluated through the equation

$$\Delta V_{ohm} = i \cdot r \tag{14}$$

$$r = \delta \cdot \rho \tag{15}$$

$$\rho = \xi e^{\alpha/T} \tag{16}$$

where r is the area-specific resistance which depends on material thickness, δ, and on the SOFC operating temperature because of the resistivity exponential dependence. For further details of the model see [1, 7, 26].

As for the concentration losses, they are worked out using the Fick's Law of diffusion and the definition of the limiting current density i_{L} (corresponding to a surface concentration value of zero) [27]:

$$i = \frac{nFD(C_B - C_A)}{3.6\delta} \tag{17}$$

$$i_L = \frac{nFDC_B}{3.6\delta} \tag{18}$$

hereafter

$$\Delta V_{conc} = \frac{RT}{nF}\ln\left(\frac{C_s}{C_B}\right) = \frac{RT}{nF}\ln\left(1 - \frac{i}{i_L}\right) \tag{19}$$

Finally, the activation losses can be described by the Butler-Volmer equation [1]:

$$i = i_o\left[Exp\left(\alpha\frac{nF}{RT}V_{act}\right) - Exp\left(-(1-\alpha)\frac{nF}{RT}V_{act}\right)\right] \tag{20}$$

where i_o is referred to as the exchange current density. So applying Eq. (20) to both anode and cathode half reactions, it becomes

$$i_{o,anode} = \gamma_{anode}\left(\frac{P_{H_2}}{P_{ref}}\right)\left(\frac{P_{H_2O}}{P_{ref}}\right)Exp\left(-\frac{V_{act,anode}}{RT}\right) \tag{21}$$

$$i_{o,anode} = \gamma_{Cathode}\left(\frac{P_{O_2}}{P_{ref}}\right)^{0.25}Exp\left(-\frac{V_{act,cathode}}{RT}\right) \tag{22}$$

Accordingly, the activation losses can be calculated as follows

$$\Delta V_{act} = A\ln\left(\frac{i}{i_o}\right) \tag{23}$$

$$A = \frac{RT}{n\alpha F} \tag{24}$$

However, Eq. (23) is only valid as long as the current density is higher than the exchange current density, see [2, 3].

Accordingly, the current density can be worked out as:

$$j = \frac{n_e Fz}{A_c} \tag{25}$$

where A_c is referred to as the cell activation area. Then, the total electrical power generated directly from the SOFC stack is given by

$$\dot{W}_{SOFC,\,stack} = NV_c jA_c \tag{26}$$

whereas the net electrical power results from subtracting the power consumed by other components, that is,

$$\dot{W}_{SOFC,\,net} = \dot{W}_{SOFC}\eta_{inv} - \dot{W}_{comp,\,i} \tag{27}$$

where η_{inv} is the inverter efficiency.

3.2. Thermodynamic model

In order to model the distribution of energy and exergy in the SOFC-VARS system, it is important to set both the boundary of the entire system and control volumes for each component as depicted in **Figure 1**. Both energy and exergy analyses are performed at steady-state condition. Kinetic and potential energy changes are negligible.

3.2.1. General energy balance

In accordance with the first law of thermodynamics, the energy balance for any system can be written as:

$$\frac{dE_{CV}}{dt} = \sum_j \dot{Q}_j - \sum_j \dot{W}_j + \sum_i \dot{m}e - \sum_o \dot{m}e \tag{28}$$

Based on Eq. (28) and **Figure 1**, the energy balance for the SOFC and VARS system is provided, respectively, in **Tables 4** and **5**. And, **Table 6** provides the energy performance parameters to be evaluated.

Component	Energy balance	Eq.
Pump 1	$\dot{W} = \dot{m}_1(h_2 - h_1)$	(29)
Fuel compressor	$\dot{W} = \dot{m}_2(h_5 - h_4)$	(30)
Air compressor	$\dot{W} = \dot{m}_3(h_8 - 7)$	(31)
Heat exchanger 1	$\dot{m}_2 h_2 + \dot{m}_{15} h_{15} = \dot{m}_3 h_3 + \dot{m}_{16} h_{16}$	(32)
Heat exchanger 2	$\dot{m}_5 h_5 + \dot{m}_{16} h_{16} = \dot{m}_6 h_6 + \dot{m}_{17} h_{17}$	(33)
Heat exchanger 3	$\dot{m}_8 h_8 + \dot{m}_{17} h_{17} = \dot{m}_9 h_9 + \dot{m}_{18} h_{18}$	(34)
Mixing chamber	$\dot{m}_3 h_3 + \dot{m}_6 h_6 + \dot{m}_{12b} h_{12b} = \dot{m}_{11} h_{11}$	(35)
SOFC	$\dot{m}_3 h_3 + \dot{m}_6 h_6 + \dot{m}_{10} h_{10} = \dot{m}_{13} h_{13} + \dot{m}_{14} h_{14} + \dot{W}_{SOFC}$	(36)
After burner	$\dot{m}_{13} h_{13} + \dot{m}_{14} h_{14} = \dot{m}_{15} h_{15}$	(37)

Table 4. Energy balance for the SOFC`s components.

Component	**Energy balance**	**Eq.**
Solution pump	$\dot{W}_{\text{pump}} = \dot{m}_{14}(h_b - h_a)$	(38)
Solution heat exchanger	$\dot{Q}_{HX} = \dot{m}_{15}(h_c - h_b) = \dot{m}_d(h_d - h_e)$	(39)
Steam generator	$\dot{Q}_{SG} = \dot{m}_d h_d + \dot{m}_g h_g - \dot{m}_c h_c = \dot{m}_{18}(h_{18} - h_{19})$	(40)
Solution valve	$h_e = h_f h_e = h_f$	(41)
Condenser	$\dot{Q}_{Cond} = \dot{m}_g\left(h_g - h_h\right) = \dot{m}_l(h_l - h_k)$	(42)
Refrigerant valve	$h_h = h_i$	(43)
Evaporator	$\dot{Q}_{evap} = \dot{m}_i\left(h_j - h_i\right) = \dot{m}_m(h_m - h_n)$	(44)
Absorber	$\dot{Q}_{abs} = \dot{m}_f h_f + \dot{m}_j h_j - \dot{m}_a h_a = \dot{m}_o\left(h_p - h_0\right)$	(45)

Table 5. Energy balance for the VARS`s components.

Parameter	**Definition**	**Eq.**
SOFC stack AC electrical power	$\dot{W}_{AC} = \dot{W}_{SOFC}/\eta_{inv}$	(46)
SOFC net electrical power	$\dot{W}_{net} = \dot{W}_{AC} - \dot{W}_{P1} - \dot{W}_{P2} - \dot{W}_{C1} - \dot{W}_{C2}$ P = pump; C = compressor	(47)
Fuel cell efficiency	$\eta_{FC} = \frac{\dot{W}_{AC}}{\dot{m}_{fuel}LHV}$	(48)
SOFC stack net efficiency	$\eta_{SOFC} = \frac{\dot{W}_{net}}{\dot{m}_{fuel}LHV}$	(49)
Cogeneration efficiency	$\eta_{FC} = \frac{\dot{W}_{AC}+\dot{Q}_{evap}}{\dot{m}_{fuel}LHV}$	(50)
Coefficient of performance	$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{gen}}$	(51)

Table 6. General parameters of performance.

3.2.2. General exergy balance

Provided that the SOFC presents both chemical and electrochemical reactions, it is required to consider two exergy contributions, namely, physical and chemical exergy. Hence the general exergy balance for a given control volume is:

$$\dot{I} = \sum_j \left(1 - \frac{T_o}{T_j}\right)\dot{Q}_j - \dot{W}_{CV} + \sum_{inlet} \dot{E}x_{inlet} - \sum_{outlet} \dot{E}x_{outlet} \tag{52}$$

where $\dot{I}$ is referred to as the exergy destruction ratio, see [28].

The physical and chemical exergy are evaluated, respectively, using the following equations [7]:

$$\dot{E}x_{ph} = \sum_i \dot{n}_i\left[\left(\overline{h}_i - \overline{h}_o\right) - T_o(\overline{s}_i - \overline{s}_o)\right] \tag{53}$$

$$\dot{E}x_{ch} = \dot{n}\left[\sum\nolimits_i y_i \overline{e}_{x,i}^{ch,o} + \overline{R}T_o \sum\nolimits_i y_i \ln\left(y_i\right)\right] \tag{54}$$

where $\overline{e}_{x,i}^{ch,o}$ is the standard chemical exergy as proposed by Szargut et al. [29]; y_i refers to the molar fraction of each species.

For the particular case of the VARS subsystem, the exergy of the Li-Br solution can be calculated using the model proposed by Palacios-Berech [30]. The model calculates the chemical exergy of the dissolution as a function of the H_2O and LiBr activities, the molality and the osmotic coefficient as described in **Table 7**.

where a_i and b_i are constants whose values are provided in **Table 8** [31].

The exergy balances for SOFC and VARS subsystems are given respectively in **Tables 9** and **10**.

In this work, the exergy analysis of the system is simplified using the general definition of exergy efficiency which is referred to as the ratio between the exergy rate of the product and the exergy rate of the fuel. Consequently, it is possible to write the exergy efficiency of the SOFC stack as:

$$\eta_{SOFC} = \frac{\dot{W}_{SOFC}}{\dot{E}x_{fuel}} \tag{61}$$

Parameter	Exergy analysis	Eq.
Molality in saturated state	$m_{sat} = \frac{x_{LiBr,\,sat}}{\left(1-x_{LiBr,\,sat}\right)\overline{M}_{LiBr}}$	(55)
Molality at any state	$m_{sat} = \frac{x_i}{(1-x_i)\overline{M}_{LiBr}}$	(56)
H_2O activity	$\ln\left(a_{H_2O}\right) = -\varnothing \cdot v \cdot m \cdot \overline{M}_{H_2O}$	(57)
Osmotic coefficient	$\phi = 1 + \sum_{i=1}^{6} a_i \cdot m^{\frac{i}{2}} + \frac{P}{2v} \sum_{i=1}^{2} i \cdot b_i \cdot m^{i/2}$	(58)
LiBr activity	$a_{LiBr} = -v\left[\ln(m) + \sum_{i=1}^{6} \frac{i+2}{i}\left(a_i + i\frac{pb_i}{2v}\right) \cdot m^{i/2}\right]_{m}^{m_{sat}}$	(59)
Chemical exergy	$\dot{E}_{ch} = \frac{RT_o}{M_{sol}}\left(y_{H_2O}\ln\left(a_{H_2O}\right) + y_{LiBr}\ln\left(a_{LiBr}\right)\right)$	(60)

Table 7. Parameters to calculate the chemical exergy of the Li-Br solution.

	j = 0	j = 1	j = 2
a_{1j}	$-2.19631551 \times 10^{1}$	4.9372316×10^{3}	$-6.55484056 \times 10^{5}$
a_{2j}	-3.810475×10^{3}	2.611535×10^{6}	-3.6699691×10^{8}
a_{3j}	1.228085×10^{5}	-7.718792×10^{7}	1.039856×10^{10}
a_{4j}	-1.471674×10^{6}	9.195285×10^{8}	-1.189450×10^{11}
a_{5j}	7.765821×10^{6}	-4.937567×10^{9}	6.317555×10^{11}
a_{6j}	-1.511892×10^{7}	9.839974×10^{9}	-1.2737×10^{12}
b_{1j}	3.07410×10^{-4}	-1.86321×10^{-1}	2.738714×10^{1}
b_{2j}	-4.080794×10^{-4}	2.16081×10^{-1}	-2.5175971×10^{1}

Table 8. Value of constants a_i and b_i as suggested in [8].

whereas the exergy efficiency of cogeneration is defined in this work as:

$$\eta_{SOFC} = \frac{\dot{W}_{net,\,SOFC} + \dot{Ex}^{Q}_{evap}}{\dot{Ex}_{fuel}} \tag{62}$$

Component	Exergy balance	Eq.
Pump 1	$\dot{E}_{D,P1} = \dot{W}_{P,1} + \dot{Ex}_1 - \dot{Ex}_2$	(55)
Fuel compressor	$\dot{E}_{D,C1} = \dot{W}_{comp1} + \dot{Ex}_4 - \dot{Ex}_5$	(56)
Air compressor	$\dot{E}_{D,C2} = \dot{W}_{comp2} + \dot{Ex}_8 - \dot{Ex}_9$	(57)
Heat exchanger 1	$\dot{E}_{D,HX1} = \dot{Ex}_2 + \dot{Ex}_{15} - \dot{Ex}_3 - \dot{Ex}_{16}$	(58)
Heat exchanger 2	$\dot{E}_{D,HX2} = \dot{Ex}_5 + \dot{Ex}_{16} - \dot{Ex}_6 - \dot{Ex}_{17}$	(59)
Heat exchanger 3	$\dot{E}_{D,HX3} = \dot{Ex}_9 + \dot{Ex}_{17} - \dot{Ex}_{10} - \dot{Ex}_{18}$	(60)
Mixing chamber	$\dot{E}_{D,MC} = \dot{Ex}_3 + \dot{Ex}_6 + \dot{Ex}_{12b} - \dot{Ex}_{11}$	(61)
SOFC stack	$\dot{E}_{D,\mathrm{sOFC}} = \dot{Ex}_{10} + \dot{Ex}_{11} - \dot{Ex}_{12} - \dot{Ex}_{14} - \dot{W}_{\mathrm{sOFC}}$	(62)
Afterburner	$\dot{E}_{D,\mathrm{AB}} = \dot{Ex}_{14} + \dot{Ex}_{13} - \dot{Ex}_{15}$	(63)

Table 9. Exergy balance equations for the SOFC′s components.

Component	Exergy balance	Eq.
Solution pump	$\dot{E}_{D,P\mathrm{sol}} = \dot{W}_{b,\,\mathrm{sol}} + \dot{Ex}_a - \dot{Ex}_b$	(55)
Solution heat exchanger	$\dot{E}_{D,HX\mathrm{sol}} = \dot{Ex}_d + \dot{Ex}_b - \dot{Ex}_c - \dot{Ex}_e$	(56)
Steam generator	$\dot{E}_{D,SG} = \dot{Ex}_{18} + \dot{Ex}_c - \dot{Ex}_{19} - \dot{Ex}_d - \dot{Ex}_g$	(57)
Condenser	$\dot{E}_{D,cond} = \dot{Ex}_g + \dot{Ex}_k - \dot{Ex}_h - \dot{Ex}_l$	(58)
Evaporator	$\dot{E}_{D,evap} = \dot{Ex}_i + \dot{Ex}_m - \dot{Ex}_j - \dot{Ex}_n$	(59)
Absorber	$\dot{E}_{D,\mathrm{Abs}} = \dot{Ex}_j + \dot{Ex}_f + \dot{Ex}_o - \dot{Ex}_a - \dot{Ex}_p$	(60)

Table 10. Exergy balance equations for the VARS′ components.

4. Validation of SOFC and VARS models

Any mathematical model is not useful if this has not been previously validated with either experimental data or previous works. Hence validation of both SOFC and VARS models is carried out in this section. For the case of the SOFC model, this is validated with a previous work introduced by Tao et al. [22]. As for the VARS model, it is validated with data taken from Herold et al. [23].

From **Figure 2**, it can be deduced that the model developed here has a good fit along a broad range of current density, but shows a slight deviation at the end of the curve (at higher current densities) which can be considered to be negligible (error less than 3%). Likewise, the VARS

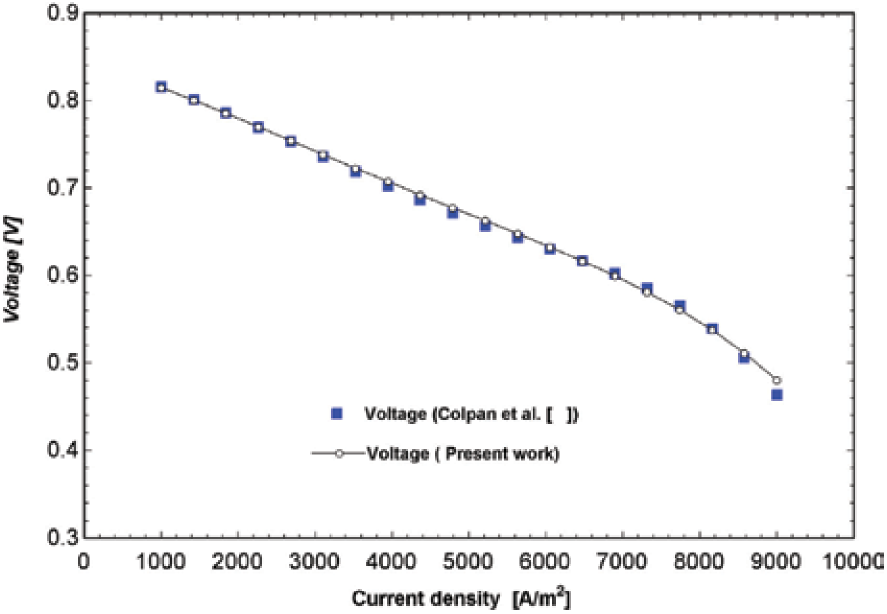

Figure 2. Characteristic polarization curve.

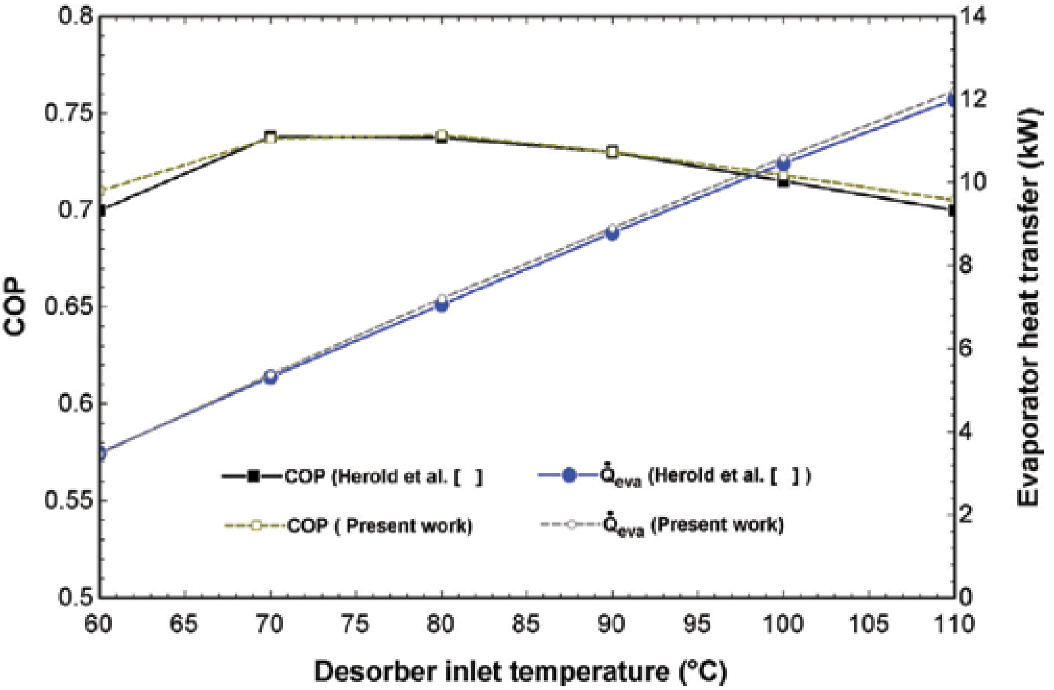

Figure 3. Characteristic curves of a vapor-adsorption refrigeration system.

model used herein is seen to exhibit good agreement with previous works, **Figure 3**. In this particular case, the validation is worked out by comparison of two important parameters used to evaluate the performance of an adsorption refrigeration system, namely, the coefficient of performance (COP) and the rate of heat transfer in the evaporator as a function of operation temperature of the generator.

5. Results and discussion

For the actual analysis, the current density, operation temperature, fuel utilization factor and steam-to-carbon ratio are considered as decision variables. The thermodynamic performance of the SOFC and the cogeneration system, i.e. its exergy efficiency, is then obtained by varying the decision variables over an acceptable operation range. So the variations of the exergetic efficiencies with such decision variables are explained to understand their nature.

5.1. Current density

One of the important parameters used to characterize the performance of a fuel cell stack is the current density when plotted versus the cell voltage, known as the polarization curve as shown in **Figure 4**. Hence it is important to study its effect on both the SOFC and the global exergy efficiencies.

The calculated variations of both energy (η) and exergy (ψ) efficiencies of the SOFC stack with current density under a constant $U_f = 0.85$ and T = 1000 K are depicted in **Figure 4**. It clearly shows the dependence of efficiency on current density.

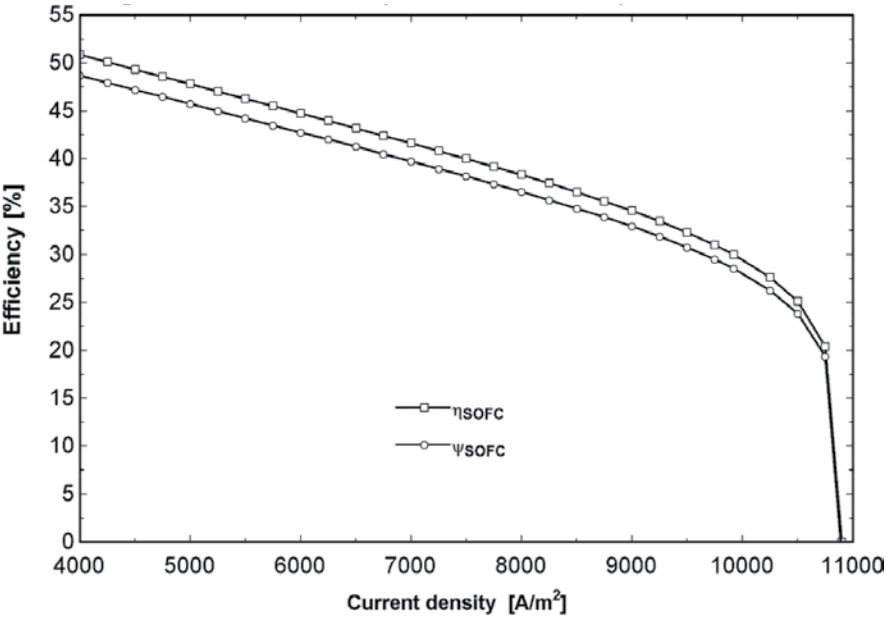

Figure 4. Variation of the SOFC energy and exergy efficiencies with respect to current density.

As expected, the calculated values for the exergy efficiency are slightly lower than those worked out for the energy efficiency. The reason is because the energy analysis does not take into account irreversibilities generated into the SOFC stack, so it assumes that more useful energy is available. Another important implication of **Figure 4** is that lowering current density increases both efficiencies as reported in literature [8], which is due largely to the reduction of voltage losses at lower current densities in accordance with literature [1].

The effect of current density on both efficiencies is further expanded to the whole system (cogeneration system) as evidenced in **Figure 5**. Moreover, it is observed that the first law efficiency of the cogeneration system is 15% higher than the first law efficiency of the SOFC attack alone. Likewise, the exergy efficiency maintains such a percentage difference. This is expected because the sensible heat of the stack gas is captured and converted into useful thermal energy to drive the adsorption refrigeration system. So cooling is available as second product. Furthermore, it is noteworthy to mention that the overall exergy efficiency is similar to other previous works [8, 32].

Figure 6 shows the results of the calculated exergy efficiencies for both SOFC stack and cogeneration system as a function of the fuel utilization factor. It is very clear that lowering the operating temperature at a given U_F, the exergy efficiency increases. This applies to both the SOFC stack and the cogeneration system. For the particular case of the cogeneration system, lowering temperature from 1000 to 900 K at UF = 0.85, the exergy efficiency increases by roughly 15%. Whereas, for the SOFC stack, its exergy efficiency increases by only 10%. This behavior stems from the fact that at lower temperatures the effect of leakage current is less significant, which causes the exergy efficiency to rise. It is, however, possible that at lower temperatures the polarization losses increase too. To some extent, it is desirable to operate at

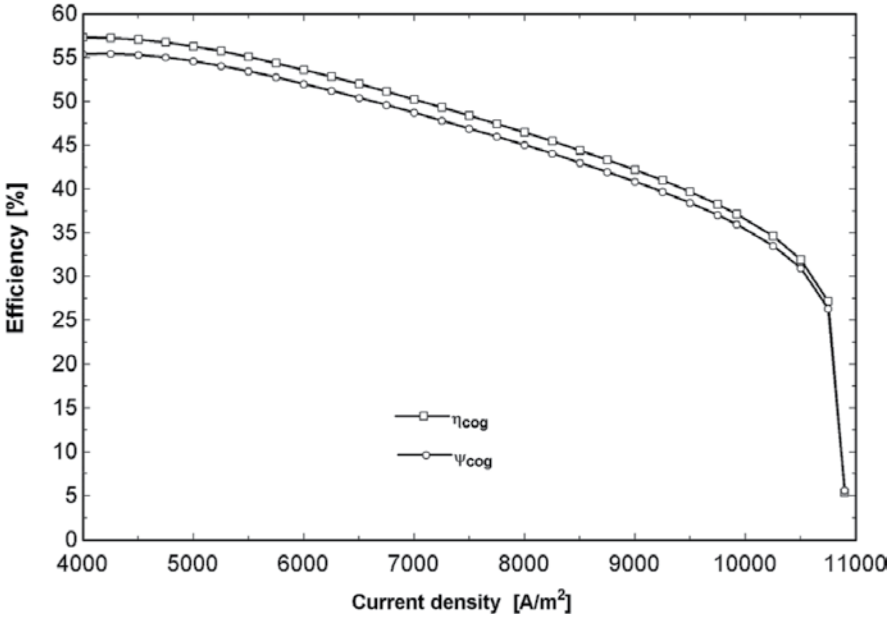

Figure 5. Variation of the cogeneration energy and exergy efficiencies with respect to current density.

U_F values lower than 0.85 and low temperatures as efficiently as possible. Consistent with **Figure 6**, the effect of temperature on exergy efficiency can be better studied when the temperature is expanded to a wider range as depicted in **Figure 7**.

In reference to the cogeneration system, it strongly suggests that the SOFC has to operate within a range between 850 and 950 K as to boost the global exergy efficiency. A noticeable trend is that lowering U_F at a given temperature, the exergy efficiency of the SOFC increases in contrast to **Figure 6**. The reason is because in spite of increasing U_F, which is assumed to increase the useful external current, there are other types of irreversibilities caused mainly by electronic and ionic conduction throughout the SOFC components (i.e. leakage currents [1]) that determine these atypical bending-over exergy efficiency curves. For the sake of comparison, a previous work [8] reports lower exergy efficiencies, for the cogeneration system, at different inlet temperature. The reason is because of the different U_f values used in this work.

On the other hand, the steam-to-carbon ratio (SC) is one of the key parameters in the operation of a SOFC that is worth analyzing. In particular, the SC defines whether carbon deposition at cell anode is built up, which causes that more heat is generated and less H_2 is consumed in the electrochemical reaction. In this respect, **Figure 8** shows that at higher SC ratios, the exergy efficiency of the SOFC stack and of the cogeneration system slightly decreases as a result of the less chemical energy converted into electrical energy as previously explained.

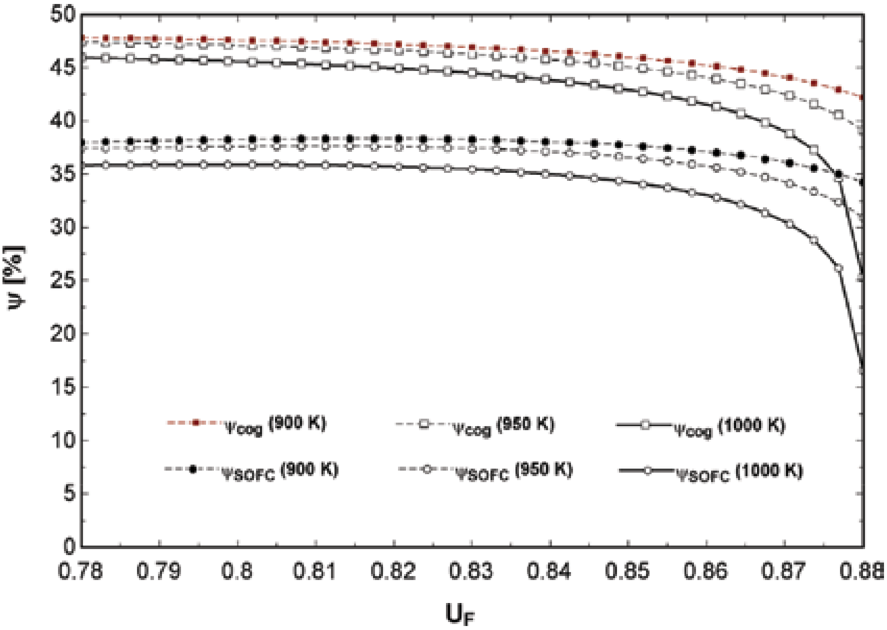

Figure 6. Variation of exergy efficiency with U_F at three different inlet fuel temperatures.

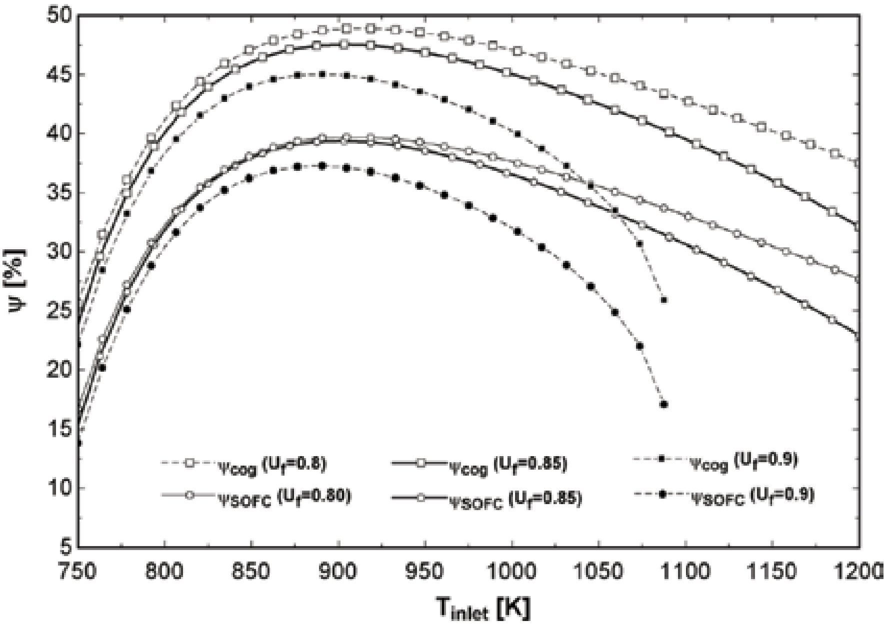

Figure 7. Variation of exergy efficiency with inlet fuel temperature at three different levels of U_F.

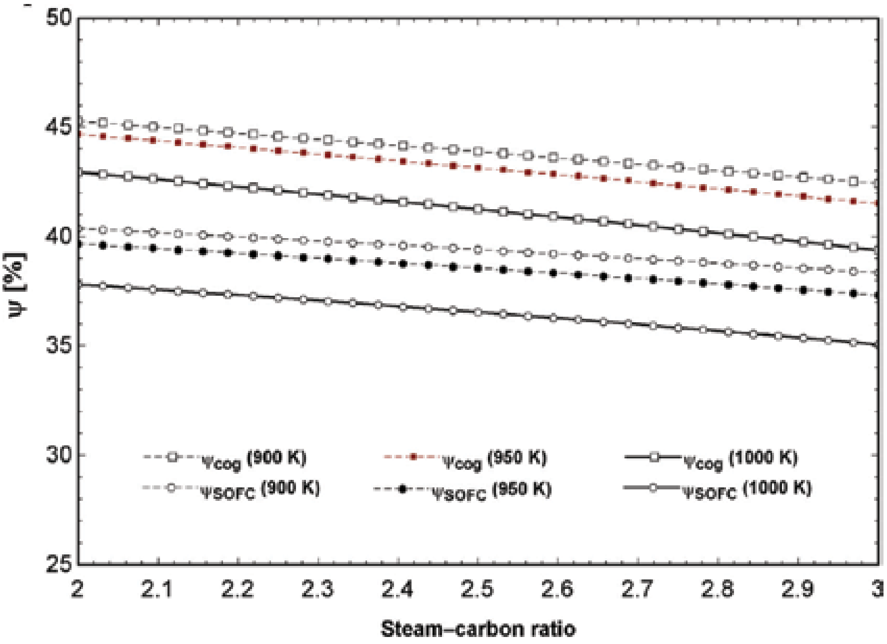

Figure 8. Effect of the steam-to-carbon ratio on exergy efficiency at three different inlet fuel temperatures.

Furthermore, a noticeable trend is that lowering temperature at a given SC ratio, both exergy efficiencies increases in agreement to **Figure 7**. In order to show the effect of U_F on exergy efficiency and at given values of SC ratio, **Figure 9** is presented. It is worth observing that exergy efficiency is slightly sensitive to SC ratio at U_F lower than 0.85. In contrast, at higher U_F values than 0.85, the effect is considerably more noticeable. It is also interesting the bending-over behavior appearing at $U_F = 0.9$, which is the result of other voltage losses as explained above. To be consistent, **Figure 6** shows that at higher values of U_F the exergy efficiency bends over as occurs in **Figure 7**.

Finally, **Figure 10** depicts the calculated efficiency defect, δ, for the most representative components of the total system (i.e. SOFC-VARS system). It is worth mentioning that the efficiency defect represents the portion of exergy or useful energy that is destroyed in each component [28]. For any case, it is noticeable that the component where the most exergy is destroyed is the heat exchanger 3 (coinciding with a previous work [8]), which is located at the end of the SOFC stack, the reason is because this is the largest heat exchanger and controls the heat that is sent to the generator of the adsorption refrigeration system.

Another noticeable observation is that increasing current density the efficiency defect of the SOFC, afterburner and heat exchanger increases, respectively, 10, 30 and 35% roughly (**Figure 10**). Moreover, it is observed from this figure that the other components are not considerably affected with respect to the current density of the SOFC. This explains why the exergy efficiency is lower than the energy efficiency discussed in **Figure 4**.

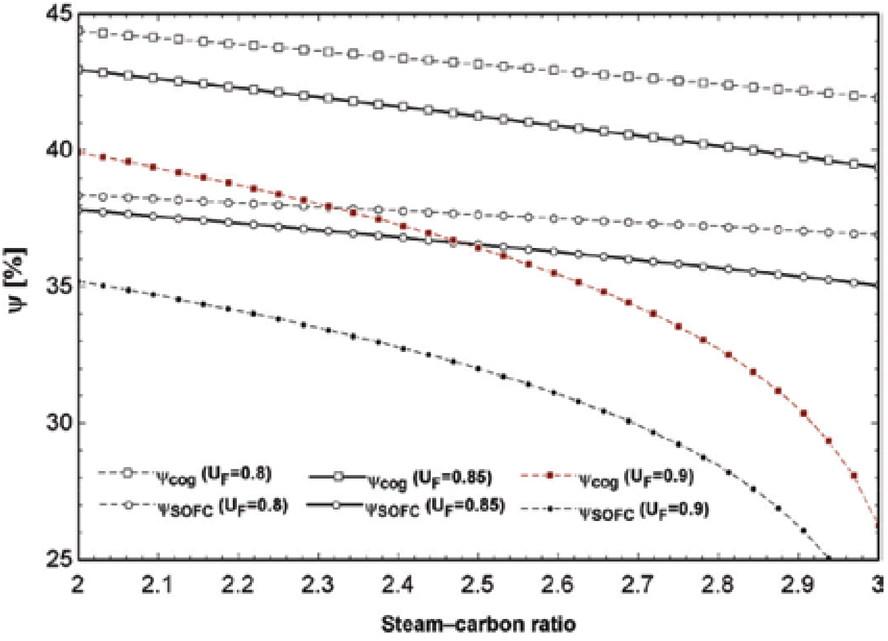

Figure 9. Effect of the steam-to-carbon ratio on exergy efficiency at three different levels of U_F ratios.

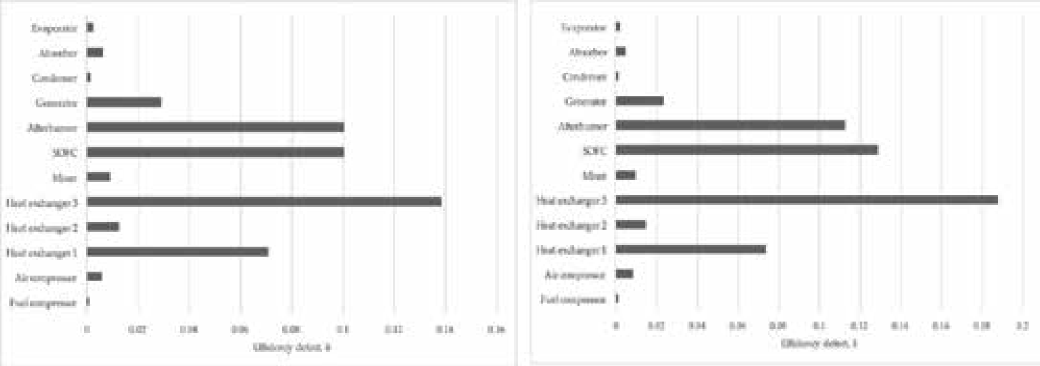

Figure 10. Efficiency defect, δ, for the total system: a) UF = 0.85, T = 1000 K and i = 500 A/m^2, b) UF = 0.85, T = 1000 K and i = 800 A/m^2.

6. Conclusions

A solid oxide fuel cell/adsorption refrigeration system for electricity and cooling generation is evaluated in terms of exergy. From a thermodynamic standpoint, a combined system is a highly efficient way of making use of heat which would otherwise be lost during the production of electricity and converts it into useful thermal energy so as to boot a vapor adsorption refrigeration system as described herein. All the mathematical models are thoroughly described in order to provide a robust and thorough exergy analysis of the system. Hence the following conclusions are worked out:

1. It is interesting to mention that applying a first law analysis it is not sufficient to evaluate the amount of usable energy that is destroyed throughout the system as evidenced in **Figure 4**.
2. The first law efficiency and second law efficiency of the SOFC stack and of the cogeneration system are affected with the SOFC current density.
3. The effect of the fuel utilization factor (U_F) on the exergy efficiency of the cogeneration and the SOFC is not substantial at U_F lower than 0.85.
4. The analysis of the effect that fuel inlet temperature has on the exergy efficiency of both the SOFC and the SOFC-VARS system demonstrates a bending-over behavior that becomes more pronounced at higher U_F values. The significance of this behavior results from the irreversibilities caused by other mechanisms such as the electronic and ionic conduction in the SOFC.
5. The exergy efficiency of both the SOFC and SOFC-VARS system is slightly sensitive to steam-to-carbon ratio, notwithstanding the temperature.
6. The effect of steam-to-carbon ratio on exergy efficiency at different values of U_F is noticeably more pronounced at higher U_F. This suggests that a more detailed analysis has to be carried out to unveil the root causes.

7. The components where most of the input exergy is destroyed are the heat exchangers, the SOFC, the afterburner and the generator. This is an advantage of the exergy analysis since it permits to pinpoint the main components where useful energy is destroyed. So efforts to improve the total system efficiency have to be targeted at these components.

Finally, it only remains to say it would be interesting to know if this model is economical and environmentally attractive and it is a project underway.

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New Exergetic Methodology to Promote Improvements in nZEB

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Additional information is available at the end of the chapter

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Abstract

The benefits obtained through the application of exergy concept in buildings are currently known, since they contribute to the proper use of energy as well as to a better adequacy of the different energy qualities taking part in a facility. Besides, an exergy analysis supports the identification of both the economic and environmental cost formation in every phase of the energy transformation chain. Those type of studies are known as exergoeconomic and exergoenvironmental analyses. In this work, a nearly zero energy buildings (nZEB) single-family dwelling is analyzed, where heating and DHW exergy demands are hourly calculated. A full exergetic analysis of its building envelope and thermal facility is carried out and exergoeconomic and exergoenvironmental analyses are applied. The aim of this study is to show the enormous possibilities for the energy efficiency improvement that still exist, which cannot be appreciated through a common energetic analysis (being the facility's energetic efficiency of 81% and exergetic one of 13%). In addition, the results of this study indicate the location and the correct assessment of the real inefficiencies.

Keywords: building envelope, building thermal facility, exergoeconomic cost, exergoenvironmental cost, improvements in nZEB

1. Introduction

The energy consumption in buildings has increased rapidly in recent years, among other things, due to the rise in population, higher requirements of healthy, comfortable indoor environments and so on. The situation in Europe indicates that tertiary buildings (residential and services) are responsible for 40% of the final energy consumption and for the 50% of the

CO_2 emissions. In Spain, these buildings account for a smaller 28% of the national global consumption, being 18% used in dwellings and the remainder 10% in services, [1]. In any case, as it is one of the main energy consumers, there is still a great potential for energy improvement in the building sector that should be performed in the coming years.

This context encouraged new European energy saving policies such as the energy efficiency directive (EED) and the energy performance of buildings directive (EPBD). The last update of the EPBD in 2010 established that, from 2020 onwards, all new buildings must be nearly zero energy buildings (nZEB). Therefore, this pressures the necessity to increase greatly buildings energy efficiency [2].

In this context, the Passivhaus standard is considered a very low-energy demand construction often used as a reference for nZEB buildings. That standard is based on an exhaustive procedure, during the project design and construction, which procures buildings with a thoroughly low-energy requirement. Nevertheless, energy is used as the base parameter of building design and, as it will be justified later on, some incongruences can arise within such analysis. One of the aims of this paper is, precisely, to demonstrate the huge potential to improve the energy performance of this type of buildings.

As it is well-known, different types of energy have diverse abilities to transform into other forms. The quality of energy (or what is the same: exergy) identifies the idea of convertibility disparity and, because of that, reflects that the same amount of energy can have different quality according to its ability to be transformed into other forms. In general, among all possible forms of energy the most common reference is *work*, in other words, the quality of energy is expressed by its ability to become useful work.

In this way, exergy is the parameter that reflects the idea of utility since it expresses the capacity of an energy to become work [3]. Some energy forms can be completely converted into work (e.g. electrical energy); consequently, energy is identified with exergy. However, there are other forms of energy (for instance, heat), that only one part can be converted into work; therefore, only a fraction of that heat flow is an exergy flow. Thus, exergy allows to quantitative assess the different levels of energy quality.

In buildings, the energy demands have different qualities and the resources used to supply them are manifold. Electricity is used for lighting and appliances, whereas other types of high-quality energy are used for indoor heating, cooling and domestic hot water (DHW), such as natural gas or fossil fuels. The electric demand is high-quality energy in contrast with the heating, cooling and DHW demands. These last ones are low-quality demands because their objective is to keep the indoor air temperature few degrees above or below the ambient temperature. Accordingly, there is no matching between the generation energy quality and the one for conditioning, as clearly exposed through the exergy analysis presented in [4]. As a consequence, important exergy destructions (quality losses) take place, which are much higher than the energy losses. **Figure 1** illustrates the proper way to use energy according to the supplying source.

The low-ex building, in contrast to nZEB, takes advantage of the different energy qualities for the different types of energy demands. These buildings reduce the energy losses and exergy destructions. Thus, an increase of the system efficiency means reducing irreversible losses, so,

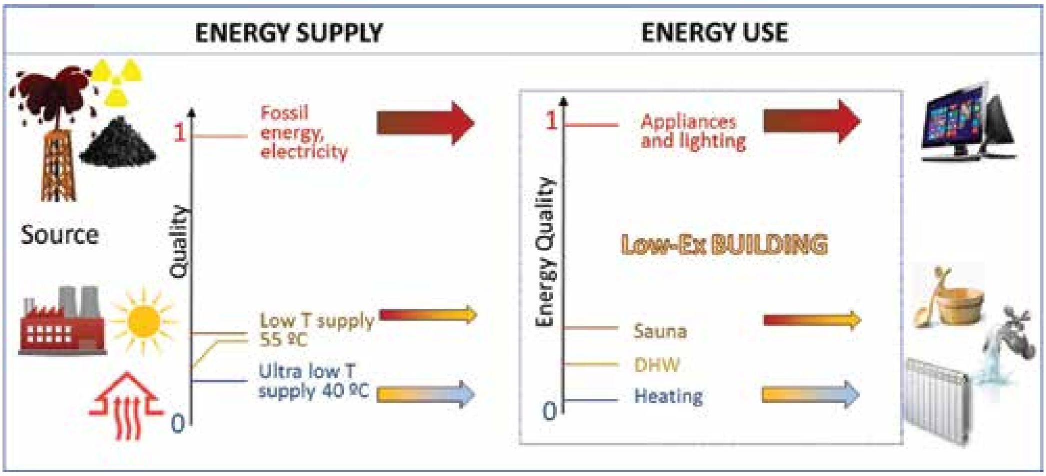

Figure 1. Energy types according to their quality.

ultimately, using energy more efficiently. A work based on building systems exergetic performance is found in [5].

This suggests that in order to reduce energy consumption in buildings as well as to use energy more efficiently, the energetic studies based on the First Law of Thermodynamics should be complemented with exergetic studies, which take into account also the Second Law of Thermodynamics. These analyses should be carried out for both building envelope and its facilities. The objective must be to optimize all the stages of the process: starting with the building design, proceeding with the construction and commissioning phases and improving the control and even the maintenance.

The main target of this study is to show a new method that comes up with the real performance values of buildings systems. In such way, structures as nZEB can further develop and optimize to achieve the common desired energy-saving goal.

The article is organized into six different sections as follows: Section 1 contains a brief introduction, Section 2 explains the way to calculate energy and exergy demands in buildings; in Section 3, the facility's exergetic study is outlined by considering the exergoeconomic and exergoenvironmental analysis. A real case study data and results appear in Section 4, where the envelope as well as the facilities is studied energetically and exegetically. Section 5 deals with the findings discussion and, finally, the last part contains the research main conclusions.

2. Energy and exergy heating demands

As already expressed, exergy measures the maximum theoretical useful work that can be obtained from an energy until it achieves the balance with the ambient [6]. Therefore, it is dependent on the environment conditions. This exergy fluctuation according to the environmental temperature variation was analyzed in several studies, such as in building services [7, 8] or in a ground-coupled heat pump [9].

Regarding buildings, the energy demand, $\dot{Q}_{heat}$, is the required amount of energy in order to keep the indoor environment in thermal comfort conditions for the users. Similarly, the exergy demand, $\dot{E}_{heat}$, is the required amount of exergy to maintain the place in comfort conditions; that is, the exergy content of the previously defined energy demand. It can be said also that exergy demand is the minimum work required to ensure the energy demand.

The supplied energy for covering the demand must be the one with the minimum quality required; otherwise, exergy destruction would come up. This happens, for example, when a heating system supplying hot water at 80°C is used to maintain the indoor air at 21°C [10]. Therefore, in an ideal situation, the minimum exergy required to satisfy the comfort conditions should be provided. Any excess exergy supplied will result in utility losses, called exergy destructions, between the heating (or cooling) facility and the demand point.

Moreover, unlike energy, exergy is not conserved but it is destroyed owing to the irreversibility of the process. So a destruction term ($\dot{E}_{D,k}$) appears, when an exergetic balance is applied in k^{th} component, by comparing the input ($\dot{E}_{in,k}$) and output ($\dot{E}_{out,k}$) exergy flows [11]:

$$\dot{E}_{in,k} = \dot{E}_{out,k} + \dot{E}_{D,k} \tag{1}$$

Accordingly, if this equation is applied to the building envelope requirement, two unknowns will appear, namely the exergy demand and the exergy destruction. Therefore, the exergy demand calculation requires knowing first the energy demand values and, later, assessing the exergy estimations.

There are two methods for exergy demand calculation: simplified and detailed. Although the first one is mostly utilized, as in [12], the second one will be used instead. This, developed in Annex 49 [13], differs from the simplified one basically because it separates the exergy demand associated with ventilation from the rest. None of them takes into account the chemical exergy and neither considers the small differences between the heat convection exergy and the radiation exergy exchanged between surfaces with small temperature differences.

Referring to the heating period, the total demand is equal to the total losses (transmission through the inertial walls, ventilation and infiltration) minus the gains (solar and internal). Hence, it is calculated as follows:

$$\dot{Q}_{demand} = \left(\dot{Q}_{trans} + \dot{Q}_{vent} + \dot{Q}_{inf}\right) - \left(\dot{Q}_{g_{solar}} + \dot{Q}_{g_{int}}\right) \tag{2}$$

The quality factor (the relationship between exergy and energy, E/Q) of the *internal energy* of a system at T_{op} is smaller than the quality factor associated with *heat energy* at the same temperature T_{op}. In this way, in order to determine the exergy demand, it will be necessary to evaluate, first of all, which part of the demand is needed to warm up (or cool down) the ventilation air and, after that, provide the remaining heat in form of heat at the operating temperature.

To sum up, exergy demand calculation should be carried on in two steps: firstly, the ventilation exergy should be accounted (the exergy needed to temperate the air coming from the outside and mixed with the inside); and secondly, if more exergy demand exists, that must be supplied as heat (or cold) (the exergy needed to temperate the room at the operating temperature T_{op}).

When the energy balance is made, the total energy demand $\dot{Q}_{demand}$ is compared first with the ventilation loss. If this is smaller than the total demand, the ventilation air must be warmed up to the operative temperature T_{op}. That implies a minimum contribution of exergy, which can be calculated with the expression:

$$\dot{E}_{vent} = \dot{Q}_{vent} \cdot \left[1 - \frac{T_0}{\left(T_{op} - T_0\right)} \cdot ln\left(\frac{T_{op}}{T_0}\right)\right] \tag{3}$$

where $\dot{Q}_{vent}$ is the heat that must be provided to temperate the room air, obtained as:

$$\dot{Q}_{vent} = \dot{m}_{vent} \cdot c_P \cdot \left(T_{op} - T_0\right) \tag{4}$$

The difference between the total demand and this $\dot{Q}_{vent}$ must be supplied as heat, at T_{op} temperature, so that the complementary exergy provided is:

$$\dot{E}_Q = \left(1 - \frac{T_0}{T_{op}}\right) \cdot \left(\dot{Q}_{demand} - \dot{Q}_{vent}\right) \tag{5}$$

In case that the total demand is less than the ventilation losses ($\dot{Q}_{demand} < \dot{Q}_{vent}$), the air does not need to be warmed up to T_{op}. This means that no additional heat is required, since it is compensated with the internal and solar gains. In such situation, the temperature of the air should be tempered at:

$$\Delta T_{vent} = \frac{\dot{Q}_{demand}}{\dot{Q}_{vent}} \cdot \left(T_{op} - T_0\right) \tag{6}$$

and, therefore, the required total exergy is:

$$\dot{E}_{vent} = \dot{Q}_{vent} \cdot \left[1 - \frac{T_0}{\Delta T_{vent}} \cdot ln\left(\frac{\Delta T_{vent} + T_0}{T_0}\right)\right] \tag{7}$$

By comparison, two circumstances can happen in cooling: in the case where $T_0 > T_{op}$, all the natural energy flows represent unwanted gains, so that $\dot{Q}_{demand} > \dot{Q}_{vent}$ will constantly be fulfilled. Hence, the ventilation air will always require to be cooled down to the T_{op} temperature. Conversely, in the situation where $T_0 < T_{op}$, the need for cooling (energy output) will not represent an exergy demand, but rather a cession of unwanted exergy. This exergy is acquired by internal gains and could be somehow collected and taken advantage of it as heat at T_{op}.

In any case, the exergy demand is verified to be about 10% of the energy demand, which obviously depends on T_0 and T_{op}.

3. Building facility study

3.1. Exergoeconomic analysis

Exergoeconomics (or thermoeconomics) is a science that combines thermodynamic and economic analyses by applying the concept of *cost of exergy*. It is an effective tool since it allows determining the production costs of an energy system; as it is the case of [14] where a sugar production process is exegetically analyzed to find out the unit exergy cost of the turbine. In addition, it allows the calculation of the intermediate costs of different flows as well.

The determination of all these streams costs is useful to make trade-off economic analyses of the subsystem components. Hence, thermoeconomics application is a key stage for designing the building thermal systems as a whole [15] or its particular components individually.

As it was said before, on one side, full systems can be studied, such as solar energy based heating systems [16], HVAC systems [17], air conditioning systems [18, 19], absorption cooling systems [20], etc. On the other side, generating and intermediate engines can be analyzed, namely micro-trigeneration machines [21], ground and air source heat pumps [22–24], heat exchangers [25], thermal energy storage modules [26] and so on.

The procedure created for the cost study is based on the exergetic cost theory (ECT) which provides the extra equations needed (apart from the exergy balance one, Eq. (1)) for solving the unknown costs of every flow. In order to apply that ECT, a *functional model* of the system must be set up from its physical model. That last physical model is used to determine simply the entering and outgoing flows of a component, or what is the same, it serves to define the typical Eq. (1) balance.

On the other hand, instead of distinguishing the flows between the ins and outs, the functional analysis classifies them among fuel (F), product (P) and residues (R) [27]. This satisfies the statement that the exergy of resources must be equal to the exergy of products plus residues plus irreversibities, as follows:

$$F = P + R + I \tag{8}$$

Once the flows are grouped according to their purpose, F_k represents the required resources for the development of the k^{th} component objective; and P_k reflects the flows that constitute the production objective. If the fuel and product are evaluated using energy parameters, the ratio between P_k and F_k would represent the energy efficiency η_k; whereas if it is defined with exergy values, the ratio would refer to the exergetic efficiency ε_k. If no exergy losses are assumed, the difference between F_k and P_k reveal the exergy destruction, $B_{D,k}$ and then:

$$\varepsilon_k = \frac{P_k}{F_k} = 1 - \frac{E_{D,k}}{F_k} \tag{9}$$

Although the exergy of a flow is a thermodynamic property that depends on its state and composition, the cost (and the environmental impact) is a function of the specific process followed by that flow production. Consequently, the same flow can have different exergy costs (or environmental impacts) according to the procedure used for its creation [28].

For this reason, the exergy cost of a flow, which is the amount of exergy required to produce it, incorporates the accumulated irreversibilities until arriving to it. Consequently, the exergy cost increases during the energy transformation chain; this fact is a result of the exergetic destructions gathered during the formation path. Consequently, this parameter shows the direct and indirect influence of the equipment interconnections, as well as the justification of different costs in each flow.

Once the productive structure of the system is specified, by following the guidelines detailed in [29], the unit exergy cost of the fuels and products of each component, $k^*_{F,k}$ and $k^*_{P,k}$ $(-)$, can be calculated. Similarly, the exergoeconomic cost of fuels and products of each component, $c_{F,k}$ and $c_{P,k}$ (€/kWh), can be obtained. These values are associated with the unitary exergoeconomic costs of the i external resources, $c_{e,i}$ $(€/kWh)$, and the inversion, maintenance and other operating costs of every component Z_k (€/kWh). Accordingly, the total costs of fuels and products, $C_{F,k}$ and $C_{P,k}$ (€), are obtained simply by multiplying the unitary exergoeconomic cost by the corresponding exergy of fuel or product.

In such way, exergoeconomics allow to assign monetary costs to the different flows, as well as to the irreversibilities. Similarly, it assesses the consumed resources costs, either in terms of energy or in economic parameters. The destroyed exergy is considered inside this resources cost due to the encountered inefficiencies. Therefore, this information helps knowing in which way the resources can be adapted to save energy more effectively. Likewise, intermediate monetary costs express the economic effects of the inefficiencies and enable to improve the performance, and, hence, the cost of the processes. Then, this methodology provides information that cannot be obtained with the conventional energy study.

3.2. Exergoenvironmental analysis

The exergoenvironmental analysis is based on a modification of the previous analysis as it evaluates the ecological impact instead of the cost problem [30]. That is, the consumed resources effect is given in terms of environmental impact. Once again, exergy is the fundamental basis since it represents the appropriate parameter to allocate both costs and environmental impacts, over the components of any energy conversion process.

The exergoenvironmental analysis proposed in [31] consists of the following three steps: first, a detailed exergy study is made; then, the environmental impact values of each component are calculated. These impacts are determined by applying the appropriate method to quantifying the environmental impact, which in many cases, the Eco-indicator 99 is used together with the life cycle analysis (LCA) method; in the last step, the environmental impact related to every component product is measured. In addition, the exergoenvironmental variables, which are analogous to the exergoeconomics, are evaluated.

As already said, the variables in the exergoenvironmental analysis maintain a similarity with those obtained by means of the exergoeconomic analysis: the specific environmental impact of fuels and products ($b_{F,k}$ and $b_{P,k}$), like to the specific cost ($c_{F,k}$ and $c_{P,k}$), consider the relative position of each component and its interconnections with the rest of the equipment. These environmental values are connected with the unitary exergoenvironmental impacts of the i external resources, $b_{e,i}$ (pts/kWh), and the impacts associated with the equipment that consider the construction stage, maintenance and other operation as well as the disposal stages of every k^{th} component Y_k (pts/kWh).

The increase of those specific environmental impacts between the fuel and the product of the component ($b_{P,k} - b_{F,k}$) represents the environmental impact related to that element. In the same way, the specific cost difference ($c_{P,k} - c_{F,k}$) represents the costs of irreversibilities due to the inefficiencies and technological limitations. Finally, the specific impact of fuel ($b_{F,k}$) equally to its cost ($c_{F,k}$), takes into account the accumulated impacts until the creation of that fuel.

3.3. Application in buildings thermal facilities

There are many works containing exergy analysis in industrial applications, such as in PEM fuel cells [32], in a sugar plant [33] or in a drying plant [34]. Even being scarcer, the concepts associated with exergy in buildings are becoming known so, as aforementioned, it is already common to consider some low-energy buildings as low-ex buildings.

Exergy analyses help showing the low effective performance of many building services: for instance, a DHW production can have an exergetic efficiency of 3.2–10.8% [35]; a heating system ranges from 2.5 to 7.4% [36] or a common air conditioning system is about 3.4% [37]. More information of building systems exergetic particularities can be found in [38].

Despite this considerable interest of exergetic application, the implementation of exergoenvironmental or even exergoeconomics analyses in buildings is still under research and it is rarely used in the daily practice. Besides, there are still many methodological aspects that must be solved.

For the adequate management of a building facility, the appropriate sensors need to be installed in both the envelope and the facility. The building is a dynamic system itself because it is directly dependent on the changing outside conditions and the user's unpredictable requirements. Therefore, it must be studied dynamically (or quasi-statically).

By an appropriate monitoring system, the exergy destruction taking place in the building thermal envelope and the energy supply system of a building can be quantified.

Through the application of the exergoeconomics and exergoenvironmental analysis, the costs and impacts can be calculated, that is, €/kWh of heating, €/kWh of DHW and so on. The knowledge of these costs provides the bases for thermoeconomic operation diagnosis [39], which allows identifying the causes of the efficiency degradation of the building energy supply system. Moreover, the assessment of its effects in terms of additional fuel consumption can be also acquired. The understanding of costs helps, in turn, with the optimization of control strategies that will guarantee lower energy consumption (or economic or CO_2 emissions), always satisfying the user demands.

4. Case study—data and results

The theory explained above was applied to a nZEB in order to become more familiar with the method and understand better the exergy use in buildings. It is a single-family dwelling (**Figure 2**) with 176 m^2 of net floor area, located in Álava (northern Spain), in a D1 climatic zone.

The thermal facility consists of a biomass stove (2.4–9 kW) for the heating coverage. The domestic hot water (DHW) is obtained by the combination of a solar panel (with a 2.3 m^2 module) that preheats the water and an air-water heat pump (3.6 kW) with an internal 300 l DHW storage, see **Figure 3**. The house includes ventilation with heat recovery system and

Figure 2. Single-family house located in Álava.

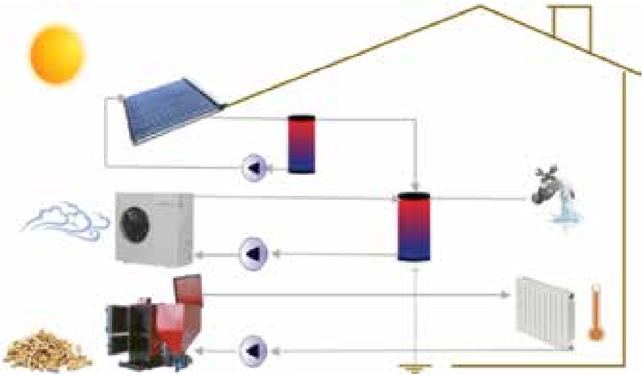

Figure 3. Heating and DHW facility of the case study.

summer bypass and there is no active cooling. The annual heating demand is less than 15 kWh/m^2·y and it has the Passivhaus certification [40].

4.1. Energy and exergy demands

First, a dynamic thermal modeling of the single-family house is performed using the EnergyPlus software. In this case, the model was calibrated by the data obtained during one-year monitoring [41]. By comparing the results obtained from the model with the monitored data, solar gains, ventilation and transmission losses were adjusted. In this way, the different components of the heating demand were hourly obtained. This is carried on according to the balance represented in Eq. (2) where the incoming flows are compensated with the outgoing flows.

The exergy heating demand has been calculated following the detailed exergy demand calculation method, as previously explained. **Figure 4** shows the accumulated annual energy balance per m^2 and, in contrast, **Figure 5** depicts the exergy balance.

It is worth noting the enormous difference among these values, as well as the fact that the exergy destruction term is about the 21% of the total exergy demand. Therefore, considerable improvements can be done to avoid those destructions.

In addition to all these, **Figure 6** shows the monthly energy gains and losses and **Figure 7** presents those gains and losses in exergy values. In both cases, the outdoor air temperature (T_{ext}) and the indoor operative temperature (T_{op}) profiles were added so that the profile of the demand could be better understood.

It should be noted that the scale of the kWh/month in the exergy graph (**Figure 7**) is five times smaller.

In order to show the hourly behavior, in **Figure 8** the heating energy and exergy demands are hourly presented. A typical winter day was chosen for that, namely February 15th. In this plot,

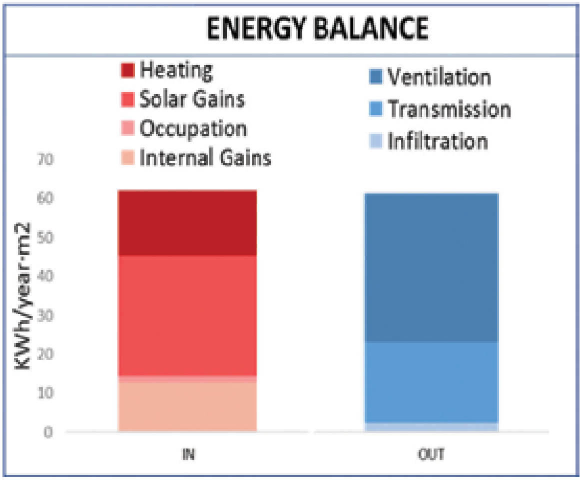

Figure 4. Energetic balance of the building.

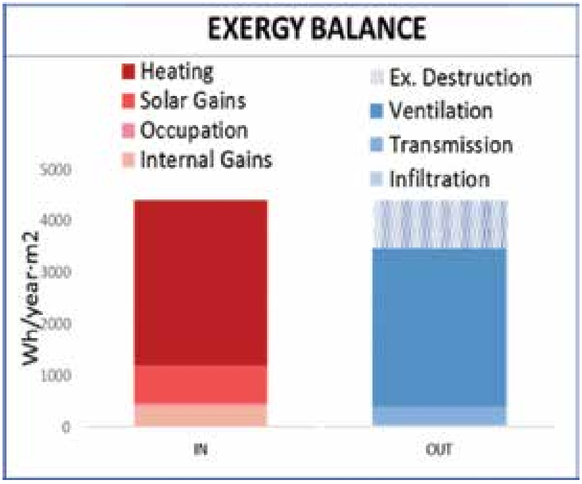

Figure 5. Exergetic balance of the building.

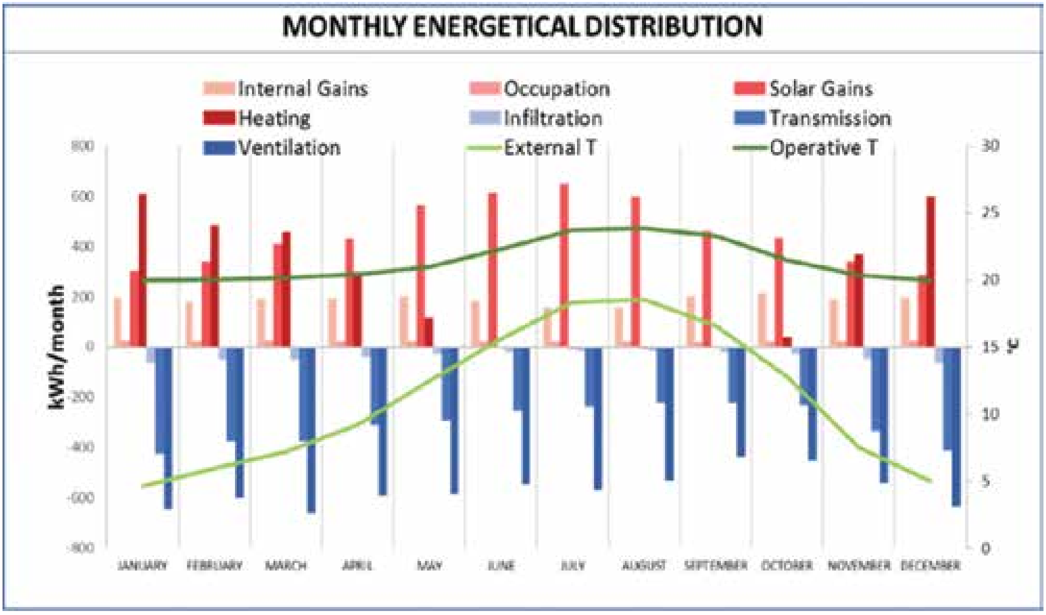

Figure 6. Energy gains and losses every month.

T_{ext} as well as the exergetic factor (the ratio between the exergy and the energy demand) were added.

In a similar way, the DHW demand is obtained based on a standard hourly profile defined by IEA-SHC Task26 software [42]. The annual DHW energy demand is 459 kWh/y·pers and the exergy demand is 46 kWh/y·pers. Then, **Figure 9** represents the energy and exergy DHW

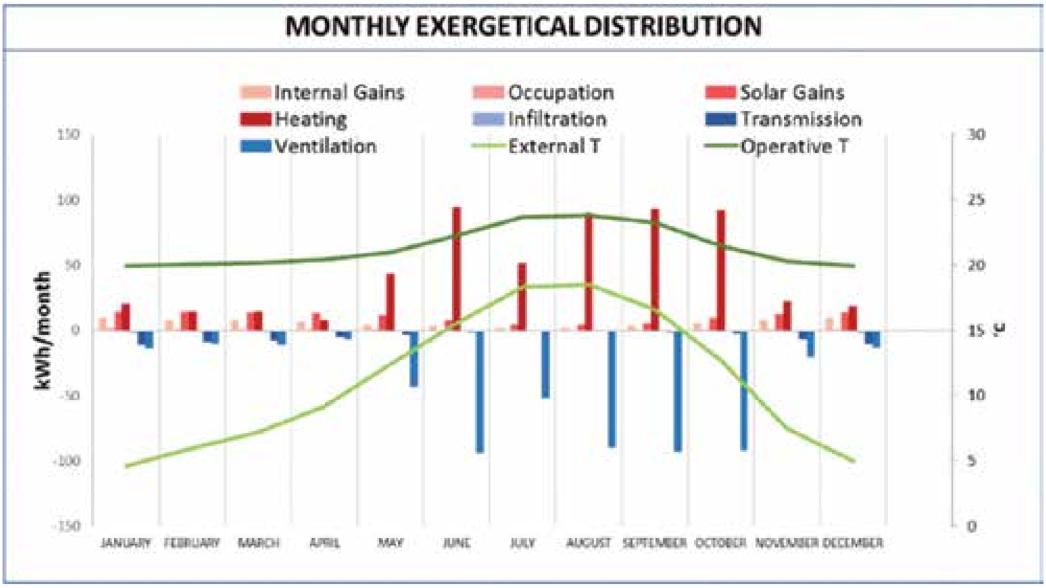

Figure 7. Exergy gains and losses every month.

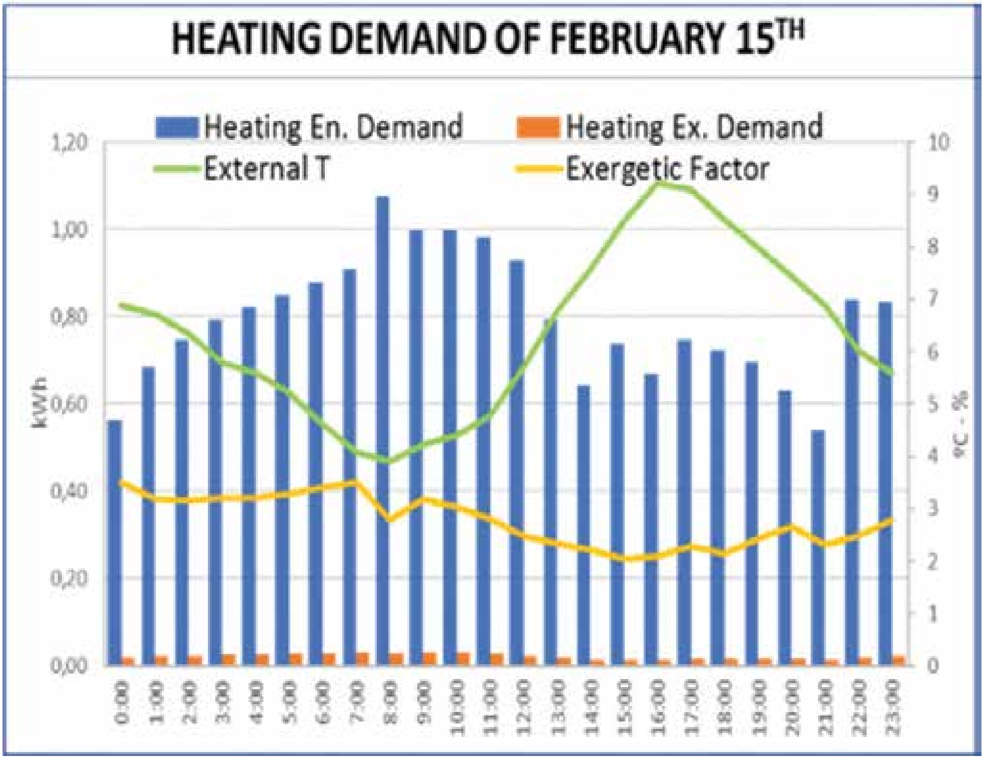

Figure 8. Q and E heating demand for winter typical day.

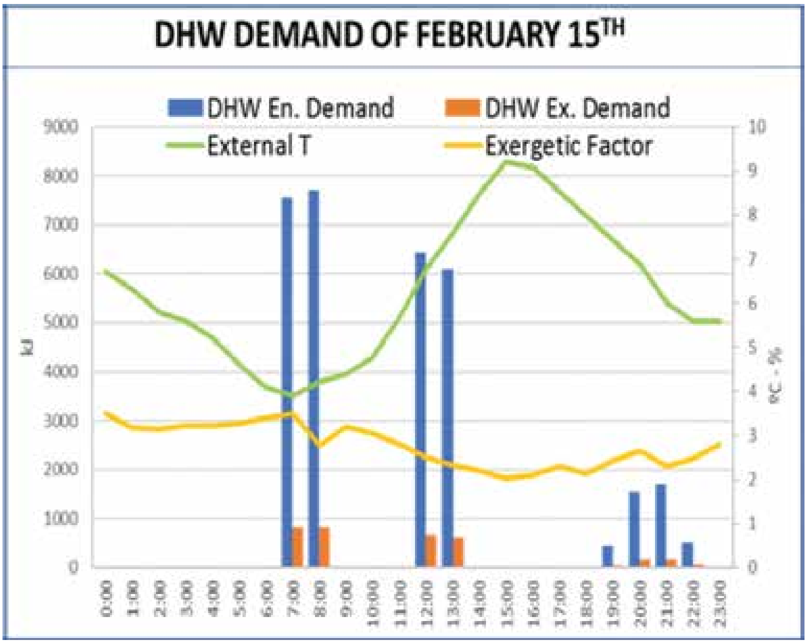

Figure 9. Q and E DHW demand for winter typical day.

demands for the same day February 15th as well as the external temperature and the exergetic factor for DHW.

4.2. Study of the facility

The facility portrayed in **Figure 3** of the previous section was simulated by means of TRNSYS v17. As it is shown, there are two principal circuits: the heating and the DHW branch. The heating demand is mainly covered by a biomass boiler whereas the DHW is provided by a combination of solar panels and an air-to-water heat pump. This circuit incorporates also two tanks: one for the solar income and the other one for the DHW storage.

The various components appearing in the case study were simulated using simplified models available in the TRNSYS software library, which try to represent their real performance as faithfully as possible. In addition, the calculated DHW and the real heating demands were simultaneously inserted in that simulation. The modeling was implemented for a period of a year with one-hour time step.

Once the simulation was performed, the thermodynamic data of every flow was extracted and the energy and exergy hourly values were registered. Hence, considering the *Q* and *E* yearly accumulated values **Figure 10** was constructed. In that **Figure 10** five stages were taken into account, moving on from the resource acquisition to the product fulfilling: primary energy, generation, distribution, storage and demand.

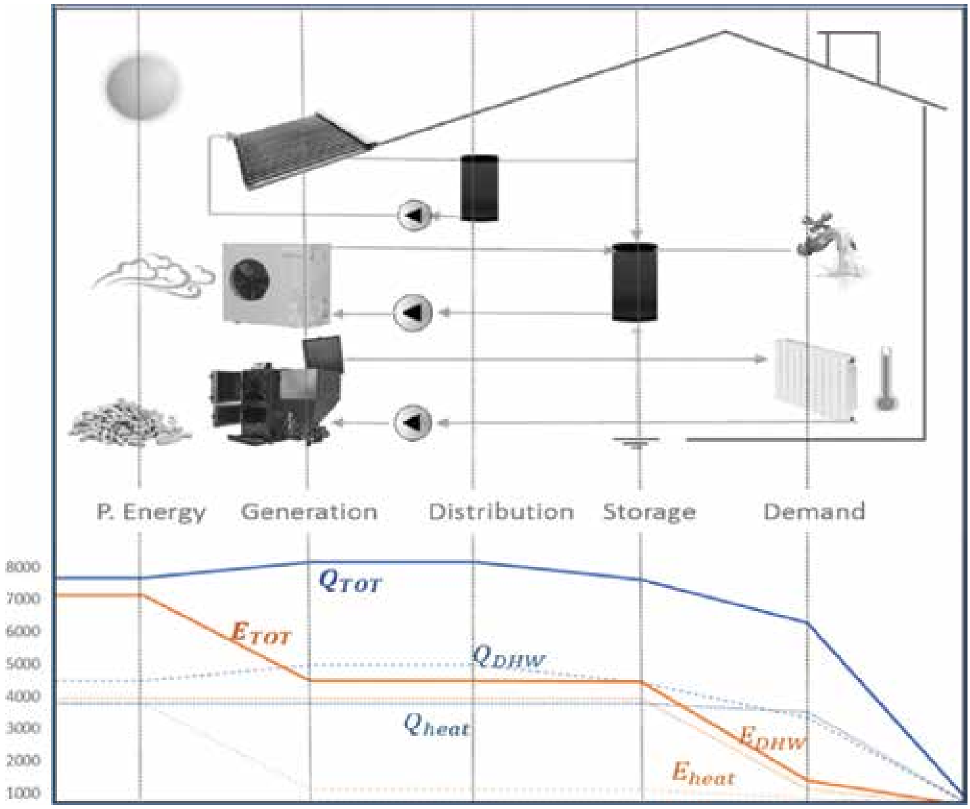

Figure 10. Energy and exergy transformation chain during the thermal facility.

The blue line illustrates the energy transformation chain whereas the orange one symbolized the exergetic one. Likewise, the full lines allude to the total results (Q_{TOT} and E_{TOT}), the dotted lines represent the DHW (Q_{DHW} and E_{DHW}) circuit, while the blinking ones refer to the heating branch (Q_{Heat} and E_{Heat}). This is a decisive graph in order to understand the quality of the energy used to provide the demand; as it can be observed, the exergy curve is notably lower than the energy one.

It may appear strange the opposite direction the Q_{TOT} and E_{TOT} lines have between the Primary Energy and Generation transformation phase, as the first one goes upwards and the second's tencency is downward. This is because, in this stage, the heat pump's coefficient of performance (COP) is considered. That value is determined by the ratio between the heat extracted from the condenser of the unit and the energy usage of the compressor. For instance, a COP value of three means that the consumption of 1 kW of electrical energy releases 3 kW of heat at the condenser. Therefore, it is always a factor higher than the unit.

However, if the quality factor of the energy is considered (i.e. the exergy amount), the value would radically decrease. This happens because electricity is pure exergy and only a part of heat energy can become in useful work, so that, by definition, the exergetic efficiency will be always less than one. In this example, whereas the yearly average COP of the heat pump is 2.85, the exergetic efficiency is $\varepsilon_{HP} = 0.28$.

Something similar occurs with storage units: as the only losses considered in energy analysis are the thermal losses, their efficiencies are close to the ideality of 100% (as it is the case of adiabatic tanks). Nevertheless, the irreversibilities occurring due to the mixing of flows at different temperatures are not contemplated there, as it happens in the cases of cold water flow mixing with the tank's hot water. In this example, the yearly average energy efficiency of the DHW storage tank is $\eta_{T1} = 81\%$ and the exergetic efficiency is $\varepsilon_{T1} = 54\%$.

All in all, the overall energy performance of the whole facility is $\eta_{TOT} = 81\%$, whereas the exergetic efficiency is $\varepsilon_{TOT} = 13\%$, as the energy conversion irreversibilities are now accounted. Likewise, the simple observation of the exergy profile E_{TOT} indicates where the greatest exergy destructions occur: during the transformation from primary energy to the warming up of the circuit and from the tank output to user's final demand. Similarly, the large-scale differences between exergy and energy are shown.

4.3. Cost and environmental impact analyses

As mentioned before, the exergoeconomic analysis was carried out in order to account the intermediate and output product flow costs. As the facility is very simple and there are only a few units, the final demand results have been presented directly.

The heating unitary (referred to energy) average cost appears to be $c_P^{heat} = 6.89\,\frac{c€}{kWh}$, whereas the DHW unitary (referred to energy) cost is $c_{P,HP}^{DHW} = 21.81\,\frac{c€}{kWh}$ when the demand is exclusively generated by means of the heat pump and it is diminished to $c_{P,S}^{DHW} = 17.99\,\frac{c€}{kWh}$ thanks to the solar thermal collectors.

To obtain these results, the acquisition, amortization and maintenance costs of every equipment have been taken into account. Furthermore, an effective annual rate of 0.05 and 20 years of useful life were considered so that results in a recovery factor of 0.08. If those fixed costs were not contemplated, the solar collector contribution would be null, or what is the same for free.

In spite of the heat pump's irreversibilities, heating production is less expensive than DHW generation since fewer equipment are connected between the primary energy consumption and the demand side. After all, as mentioned before, the cost is related to its formation process, and consequently, lower irreversibilities are accumulated along the heating path.

Discussing about environmental impact, the following considerations can be forwarded: first, it must be noted that performing a LCA for each component of the system requires a learning time and a knowledge that are not the objective of this work. Besides, the results obtained in other studies show that the environmental impact the units have in the final products is really small. For these reasons, the exergoenvironmental analysis has been carried out considering only the CO_2 emissions associated exclusively with the input resources, $b_{e,i}$ (CO_2/kWh) and not with the equipment that make up the energy facility.

Following those simplifications, the analysis was carried out and the average results were obtained. For instance, the environmental impact related to the heating branch is $b_P^{heat} = 0\frac{kg_{CO_2}}{kWh}$. This null effect is because biomass is considered as a renewable source with a neutral CO_2 emission balance. On the other way round, the DHW environmental impact corresponding to the heat pump generation is $b_{P,HP}^{DHW} = 0.65\frac{kg_{CO_2}}{kWh}$ while the DHW coming from the solar panel is zero as the sun is a sustainable source.

5. Results and discussion

The building sector is responsible for almost a third of the total energy consumption over the world and this justifies the great concern for the improvement of their energetic efficiency. For this reason, great advances have been prompted in energy regulations during the last decade; moreover, the Directive 2010/31/UE already lays down a broad definition of a nZEB and establishes the 31st of December of 2020 as the deadline for making all new buildings nZEB. Nevertheless, it allows the Member States to draw up national plans for increasing the number of nZEB and, undoubtedly, this is the first step toward the positive energy building. In that context, many authors have worked in the optimization and design of thermal systems for buildings, but unfortunately, most of them were done from a purely energetic point of view.

One of the aims of this paper is to show the differences between the energy and exergy performance of the whole energy supply chain of buildings. For that, the whole way from the primary energy until the demand covering is considered. In this regard, energy and exergy analysis are performed to a recognized a nZEB situated in Álava, Spain, (*with* Passivhaus certificate) following the First and the Second Law of Thermodynamics through a yearly dynamic analysis.

The yearly heating demand amounts $Q_{heat} = 2.96\,GWh/y$ and the yearly DHW demand is $Q_{DHW} = 2.76\,GWh/y$. Translated into exergy values those demands are $E_{heat} = 0.56\,GWh/y$ and $E_{DHW} = 0.27\,GWh/y$, respectively, being evident the low-quality factor of both. The energetic performance of the heating circuit is $\eta_{heat} = 93\%$ and the exergetic efficiency $\varepsilon_{heat} = 17\%$. Meanwhile, the DHW generation circuit efficiencies are $\eta_{DHW} = 71\%$ and $\varepsilon_{DHW} = 0.09\%$.

As a result, even being a nZEB, energy saving enhancements can be accomplished as long as the energy quality is considered in both the building thermal envelope and the thermal facility. A reduction of the exergy resource consumption implies that less high-quality energy is needed and, thus, low-quality energy sources can be used instead (such as residual heat) to cover the demand.

Besides, one of the outcomes of the exergoeconomics and exergoenvironmental analyses are the allocation of costs and environmental impacts, to final products and intermediate flows, based on physical criteria. In this way, the heating and DHW final specific costs depending on their cost formation were obtained and shown in Section 4.3.

Thus, the use of exergy as a base variable supports both the energy efficiency (focusing on the required energy reduction) and the enhancement of the use of renewable energy, both in economic and environmental terms. The European directives related to energy efficiency in buildings should contemplate, therefore, the exergy as an additional basis of study.

6. Conclusions

The conventional energy studies are based on the First Law of Thermodynamics. This type of analysis is confined to a simple energy accounting, which quantifies the energy inputs and outputs of a system and particularly of a building. In this way, the energy given to the processes through fuels, electricity, flows of matter and so on, must appear into the final products or by-products. Under this perspective, energy losses are the not used heat flows. Thus, the analysis based on the First Law suggests that the loss of efficiency of an equipment or a process is a consequence of those waste heats.

There are currently different ways to state the energetic efficiency of a system or component based in this First Law and none of them takes into account the quality of energy. Thus, with those efficiency definitions the same weight can be assigned to different forms of energy, regardless of their quality. Correspondingly, this conveys some drawbacks, for example, the fact that the performance of the Carnot engine is the Carnot factor instead of the unit (which is what one expects for the perfect engine); or even the point that the heat pumps efficiency is expressed through the COP (an index always greater than unity) and so on. Furthermore, large thermoelectric plants, which are regarded among the most efficient energy conversion systems, have low performances (between ~ 40 and 55%); while typical individual hot water boilers, which are thermodynamically much less efficient devices, appear to have higher performances (~90%), a fact that seems contradictory.

By contrast, the exergy-based efficiencies describe better the way in which the resources are used and provide a clearer guidance about the possible improvements. Both, the exergy destruction and the entropy production, are valid measures for the irreversibility of a process. However, the use of entropy makes difficult to assign a meaning to the loss due to the encountered irreversibilities. On the other side, the exergy method allows assessing directly the real losses of a process, that is, it evaluates the decrease in the available work because of the process transformation irreversibilities. Accordingly, the irreversibilities measure the system inefficiency and the exergy method quantifies them and enables identifying their location.

Notwithstanding these advantages, the exergetic method does not allow determining the effect of each equipment irreversibilities over the required additional consumption; it does not permit determining the impact of an improper operation due to a particular equipment. To achieve that objective, exergoeconomics was developed, which is a science that combines the Second Law with economic concepts. It develops the concept of exergetic cost, which reflects the required exergy to produce any flow, calculated from its formation process. Consequently, this exergetic cost is the weighting factor for accounting every irreversibility regarding the global resource consumption. To determine those exergetic costs, in addition to the exergy flows, the productive structure of the installation must be defined [27].

Based on the same exergetic fundaments, Exergoenvironmics was evolved which accounts for the irreversibility formation in terms of environmental impacts. These types of analyses are especially suitable for larger scale installations, such as district heating systems. Both Exergoeconomics and Exergoenvironmics can also be used in energy audits, since they allow detecting the places where losses occur and quantifies their costs and environmental impacts. This fact makes easier to propose profitable improvements. Likewise, they can be applied to the design and synthesis of energy plants as they provide the designer with information about the cost formation process, the interactions among thermodynamics, economics, environmental impacts and the interactions among the plant components. This information is especially useful for the design of energy supply systems.

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Abbreviations

EED	energy efficiency directive
EPBD	energy performance of buildings directive
nZEB	nearly zero energy buildings
DHW	domestic hot water
ECT	exergetic cost theory

Nomenclature

$\dot{Q}_{demand}$ (kW)	energy demand of the building
$\dot{Q}_{trans}$,$\dot{Q}_{vent}$ $\dot{Q}_{inf}$ (kW)	transmission, ventilation and infiltration losses
$\dot{Q}_{g_{solar}}$, $\dot{Q}_{g_{int}}$ (kW)	solar and internal gains

T_0, T_{op} (K)	ambient temperature and building operative temperature
$\dot{m}_{vent}, \dot{Q}_{vent}, \dot{E}_{vent}$ (kg/h) (kW)	ventilation mass flow rate, energy and exergy
$\dot{E}_{in,k}, \dot{E}_{out,k}$ (kW)	input and output exergy flow of kth component
$\dot{E}_{D,k}$ (kW)	exergy destruction of kth component
F_k, P_k, R_k, I_k (kWh)	fuel, product, residues and irreversibities of kth component
$\eta_k\ \varepsilon_k$ (%)	energy and exergy efficiency of kth component
$k^*_{F,k}\ k^*_{P,k}$ $(-)$	unit exergy cost of the fuels and products of kth component
$c_{F,k}\ c_{P,k}$ (€/kWh)	unit exergoeconomic cost of fuels and products of kth component
$c_{e,i}\ Z_k$ (€/kWh)	unit external resources cost and inversion, maintenance and other operating costs of kth component
$b_{F,k}\ b_{P,k}$ (pts/kWh)	specific environmental impact of fuels and products of kth component
$b_{e,i}\ Y_k$ (pts/kWh)	unit external resources exergoenvironmental impact and construction stage, maintenance, operation and disposal impacts of kth component

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Application of Exergy Analysis to Energy Systems

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Additional information is available at the end of the chapter

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Abstract

Exergy analysis is a practical approach to evaluate the merit of energy conversion or distribution processes and systems. With the aid of an energy analysis, the performance of an energy conversion system cannot be evaluated efficiently and precisely. But, an exergy analysis complements and enhances an energy analysis. Exergy analysis involves the application of exergy concepts, balances, and efficiencies to evaluate and improve energy and other systems. Many scientists suggest that processes or sytems can be well evaluated and improved using exergy analysis in addition to or in place of energy analysis. Application of exergy analysis has given us more beneficial opportunities through a big part of a wide range of processes and systems particularly for the evaluation of energy systems and technologies as well as an environmental impact in all existing thermal and nuclear power plants. Conventional energy technologies, especially for power generation plants, have made numerous energy and exergy analyses and have produced beneficial results. Also, the use of energy and exergy analyses for advanced nuclear energy technologies can be expected to provide meaningful insights into performance that can assist in achieving optimal design concepts. Finally, explaining the analysis of thermal and nuclear power plant systems deals with exergetic approach.

Keywords: energy analysis, exergy analysis, energy conversion systems, power plant

1. Introduction

Thermodynamics permits the behavior, performance, and efficiency to be described for systems for the conversion of energy from one form to another. Conventional thermodynamic analysis is based primarily on the first law of thermodynamics, which states the principle of conservation of energy. An energy analysis of an energy conversion system is essentially an accounting of the energies entering and exiting. The exiting energy can be broken down into products and wastes. Efficiencies are often evaluated as ratios of energy quantities and are often used to assess and compare various systems. The thermodynamic losses that occur within a system often are not accurately identified and assessed with energy analysis.

Exergy is defined as the maximum amount of work that can be produced by a stream or system as it is brought into equilibrium with a reference environment, and it can be thought of as a measure of the usefulness or quality of energy. Exergy is consumed during real processes due to irreversibilities and conserved during ideal processes. The exergy analysis nomenclature used here follows that proposed by Kotas et al. [1].

Exergy analysis is a powerful tool for developing, evaluating, and improving an energy conversion system. The growing energy supply and demand have created an interest toward the plant equipment efficiency and the optimization of existing thermal power plants. At present, most of the power plants are going to be designed by the energetic performance criterion which is based on the first law of thermodynamics. Energy losses taking place in a system can be easily determined by using exergy analysis.

The exergy concept has gained considerable interest in the thermodynamic analysis of thermal processes and plant systems since it has been seen that the first law analysis has been insufficient from an energy performance standpoint. The system energy balance is not sufficient for the possible finding of the system imperfections.

There is a new technology for high-temperature air combustion and ultrahigh-temperature combined cycle. In this case, it is necessary to study the exergy analysis on combustion and thermodynamic processes, because ordinary energy analysis does not have any evaluation supported at its temperature level. In this particular field of engineering, it is difficult to use the ambient temperature energy of air and water, which are widely available. In a thermodynamic cycle, it is necessary to consider the power generation, which includes many kinds of effective and invalid items. The exergy analysis must be introduced to analyze power generation and heat pump cycles against energy analysis. Recently, a large number of studies based on exergy analysis have been carried out by many researchers all over the world in various system applications [2].

The benefits of exergy analysis are numerous compared to energy analysis. Some of the more significant ones follow below:

- Exergy efficiencies are always the measures of the approach to true ideality and provide more meaningful information when assessing the performance of energy systems. Also, exergy losses clearly identify the locations, causes, and sources of deviations from ideality in a system.
- Exergy methods can help evaluate the thermodynamic values of the product energy forms in complex systems with multiple products (e.g., cogeneration and trigeneration plants).
- Exergy-based methods can be used to improve economical and environmental assessments.
- Exergy can improve understanding of terms like energy conservation and energy crisis.
- Exergy methods can help in optimization activities.

There has been an increasing interest in using energy and exergy modeling techniques for energy utilization assessments in order to attain energy and financial savings [3].

All energy conversion systems have to be analyzed in terms of energetic, economic, and environmental aspects. Exergy-based analyses are very convenient methods for assessing the performance of energy conversion systems. Exergy is the maximum work that can be obtained from a system. Exergy-based analyses help determine the irreversibilities (entropy generation) and how a source can be used effectively. There are a few studies on advanced exergy-based analyses of power-generating systems in the open literature [4–13].

Rosen [4] presented energy-based and exergy-based comparisons of coal-fired and nuclear electrical-generating stations. Overall energy and exergy efficiencies, respectively, are 37 and 36% for the coal-fired process and 30 and 30% for the nuclear process. The losses in both plants exhibit many common characteristics. Naterer et al. [5] analyzed the coal-fired thermal power plant with measured boiler and türbine losses. Tsatsaronis and Moung-Ho [6] were the first to develop the concepts of avoidable and unavoidable exergy destruction, which were used to determine the potential of improving the thermodynamic performance and cost-effectiveness of a system. Morosuk and Tsatsaronis [7] applied advanced exergy analysis to a simple gas turbine cycle to assess its performance and discussed their calculation methods in detail. Khaliq and Kaushik [8] studied thermodynamic methodology for the performance evaluation of combustion gas turbine cogeneration system with reheat. The effects of process steam pressure and temperature used in the design of heat recovery steam generator, and reheat on energetic and exergetic efficiencies has been investigated. Koroneos et al. [9] discussed the exergy analysis of solar energy, wind power, and geothermal energy, and renewable energy sources are compared with the nonrenewable energy sources on the basis of efficiency. Ivar et al. [10] studied the exergy analysis of a natural-gas fired power plant with CO_2 capture. It indicates that maximum exergy is destroyed during the combustion and steam generation process, which represents over 80% of the total exergy destruction in the overall system. Petrakopoulou et al. [11] studied a combined power plant using advanced exergy and conventional analyses and demonstrated the superiority of the former. They reported that an advanced exergy analysis provided a wide range of optimization strategies and potential improvements. Taner [12] studied the energy and exergy analysis of a sugar factory model in Turkey and investigated within a general context to provide energy saving by reducing energy and exergy losses in the sugar production process. In another study, Taner [13] discussed the performance of a proton exchange membrane (PEM) fuel cell in terms of its pressure and voltage parameters and researched by experimental optimization to improve the performance, efficiency, and development of modeling and simulations of PEM fuel cells.

2. Comprehensive energy and exergy analyses

In general, open systems have mass, heat, and work interactions, and closed systems have heat and work interactions. The mass conservation equation can be written for an open system (control volume) as follows:

$$\sum_{i} m_i - \sum_{e} m_e = 0 \tag{1}$$

Energy, being subject to a conservation law (neglecting nuclear reactions), can be neither generated nor consumed. For a nonsteady flow process in a system during a finite time interval, energy balance can be written as follows [14]:

$$\text{Energy input} - \text{Energy output} = \text{Energy accumulation} \tag{2}$$

The first law energy balance equation for the control volume system is

$$\sum_{i} (h + ke + pe)_i m_i - \sum_{e} (h + ke + pe)_e m_e + \sum_{r} Q_r - W = 0 \tag{3}$$

where Q_r is the heat transfer into the system across r region on the system boundary; W is the work (including all forms of work) transferred out of the system; m_i and m_e *denote*, *respectively*, the rate of mass input and exits; and h, ke, and pe are the specific values of enthalpy, kinetic energy, and potential energy, respectively [14].

For a nonsteady flow process in a system, exergy balance can be written as follows [14]:

$$\text{Exergy input} - \text{Exergy output} - \text{Exergy consumption} = \text{Exergy accumulation} \tag{4}$$

Exergy is consumed due to irreversibilities. Exergy consumption is proportional to entropy creation. The main important difference between energy and exergy: energy is conserved, while exergy, a measure of energy quality or work potential, can be consumed.

The general exergy balance for the above system can also be expressed as [15]:

$$Ex_i - Ex_e = Ex_{heat} - Ex_{work} + Ex_{mass,i} - Ex_{mass,e} = Ex_{dest} = I \tag{5}$$

Assuming that flows are one-dimensional and the input and output terms in Eq. (5) are net quantities, the following may be written:

$$\sum_{i} m_i ex_i - \sum_{e} m_e ex_e + \sum_{r} Ex_{Qr} - Ex_W - I = 0 \tag{6}$$

The first two terms on the left-hand side of Eq. (6) represent the net input rate of exergy associated with matter, the third term the net input rate of exergy associated with heat, the fourth term the net output rate of exergy associated with work, and the fifth term the rate of irreversibility.

The thermal exergy transfer (Ex_{Qr}) at a constant temperature of T_r and the work exergy (Ex_W) may be calculated from the following equations:

$$Ex_{Qr} = Q_r \left(1 - \frac{T_r}{T_0}\right) \tag{7}$$

$$Ex_W = W \tag{8}$$

The rate of irreversibility, I, equals to exergy loss as

$$I = T_0 S_{gen} \tag{9}$$

This statement equals to the exergy destruction in the real process, where the subscript 0 indicates properties at the reference environment of P_0 and T_0. The specific exergy of a mass flow with negligible potential and kinetic energy changes may be written as

$$ex = \left[(h - h_0) - T_0(s - s_0) + \left[\sum\nolimits_j \left(\mu_{j0} - \mu_{j00}\right) x_j\right]\right. \tag{10}$$

where x_j denotes the mass fraction of species j, s is the specific entropy, μ_{j0} is chemical potentials for each of the j components, and the subscript 0 refers to a quantity evaluated with respect to a reference environment. It is assumed to be reversible processes in which its temperature T_0, pressure P_0, and the chemical potentials, μ_{j00}, for each of the j components presented remain constant. The reference environment is in stable equilibrium [14].

The energy or the first law efficiency η of a system or system component is defined as the ratio of energy output to the energy input of system or system component, i.e.,

$$\eta = \left(\text{Desired output energy}/\text{Input energy supplied}\right) \tag{11}$$

The second law efficiency is defined as

$$\zeta = \left(\text{Desired output}/\text{Maximum possible output}\right) \tag{12}$$

Energy-related systems commonly have been designed and evaluated using the first law heat balances. But such calculations neglect the variation in the quality of the energy throughout a system. As a result, losses and inefficiencies are not evaluated realistically.

An exergy analysis should be conducted only after the validity of the mass, and energy balances has been confirmed. The thermodynamic analysis should be evaluated that an exergy balance is obtained by combining the corresponding energy and entropy balances [16].

A simple procedure for performing a comprehensive exergy analysis of such a system involves the following steps:

- Subdivision of the process under consideration into each system component (process or subprocess), depending on the depth of detail and understanding desired from the analysis.
- Calculation of the conventional mass and energy balances on the process and definition of all basic quantities (e.g., work and heat) and properties (e.g., temperature and pressure).
- Based on the nature of the process, the acceptable degree of analysis complexity and accuracy, and select a reference environment model.
- Calculation of the energy and exergy values, relate to the selected reference environment model.

- Calculation of the exergy balances including the determination of exergy consumptions.
- Select the efficiency definitions, depending on the measures of merit desired, and evaluate the efficiencies.
- Detailed evaluation of each system component based on the results, and draw appropriate conclusions and recommendations relating to such issues as design changes, retrofit plant modifications, etc.

3. Application of exergy analysis to energy systems

The types of applications of exergy methods to energy systems are extremely varied and include the following subgrouped as (a) utility, (b) industrial, (c) residential-commercial, and (d) transportation. Application of the exergy analysis may be applied predominantly among industrial energy systems.

The energy-utilization efficiency of a macrosystem, such as a region or country, can be assessed beneficially using exergy analysis. This section illustrates how the efficiency of energy utilization in a particular macrosystem, region, or country can be examined using both energy and exergy analyses.

The relation between energy resources and sectors of a macrosystem can be modeled as in **Figure 1**. The energy resources considered are coal, petroleum, natural gas, nuclear energy, and hydraulic energy. Renewable energy resources (e.g., solar energy, hydropower, biomass, geothermal, wind) can be neglected since the quantity is minor relative to other resources.

The utility sector includes processes for electricity generation using both conventional sources, such as fossil and nuclear power plants, and alternative sources, such as solar energy,

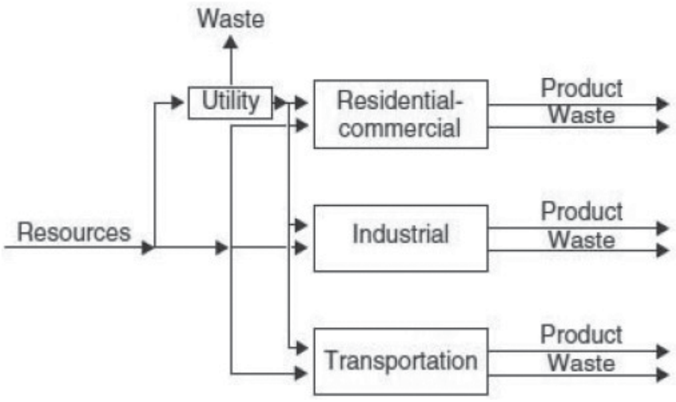

Figure 1. The diagram of the energy flows in a macrosystem.

hydropower, biomass, geothermal, wind, etc. It can be expected that performing an exergy analysis will be meaningful for performance comparisons, assessments, and improvement for all conventional and alternative power plants with a low efficiencies. Energy and exergy utilization efficiencies for utility sector, η and ζ, can be calculated from the given above equations, respectively.

Generally, the performance of thermal power plants is evaluated through energetic performance criteria based on the first law of thermodynamics, including electrical power and thermal efficiency. In recent decades, the exergetic performance based on the second law of thermodynamics has found a useful method in the design, evaluation, optimization, and improvement of thermal power plants. The exergetic performance analysis cannot only determines magnitudes, location, and causes of irreversibilities in the plants but also provides more meaningful assessment of plant individual component efficiency [17]. Lately, many researchers have concentrated their attention on exergy analysis of thermal power plants to optimize energy quality.

The industrial sector (petrochemical, chemical, and metallurgical processes, heating and cooling systems, etc.) is the most complex for determining overall efficiency and effectiveness values due to the profusion of different uses of energy. Exergy methods are used in many industries because they provide powerful tools for analyzing, assessing, designing, improving, and optimizing systems and processes [14].

In this sector, the energy consumption may be grouped as (1) mechanical drive, (2) process steam, (3) direct heat, (4) space heating, and (5) others, such as lighting, electrolytic processes, and miscellaneous applications. Relations used to calculate energy and exergy efficiencies of process steam, direct heat, and space heating are similar, expecting to the process steam, direct heat, and space heating temperatures [18].

Residential-commercial sector includes residential, commercial public, and agriculture activities (lighting, water heating, space heating, cooking, refrigerating, air conditioning, etc.), while it uses various energy carriers. Utilization of renewable energy may be spread, such as from sunlight to heat water, from geothermal to heat water and space, and from biowaste for general use.

The only specific application for which energy and exergy consumptions have been estimated in the transportation sector is transportation of fuel, because the use of energy sources as raw material is considered to be non-energy use.

Figure 2 shows the distribution of the world electricity use by sector over the period of 2015–2040. Electricity use increases the most in industrial, residential, and commercial buildings over the period of 2015–2040 [19].

Net electricity generation in the world increases an average 1.0%/year from 2015 to 2040, so it is very important to carry out an energy analysis of power generation plants with low efficiency.

One of the areas is environmental impact in which applications of exergy are expanding. The effect of energy resource utilization on the environment should be addressed by way of using exergy. Although the exergy of an energy form or a substance is a measure of its

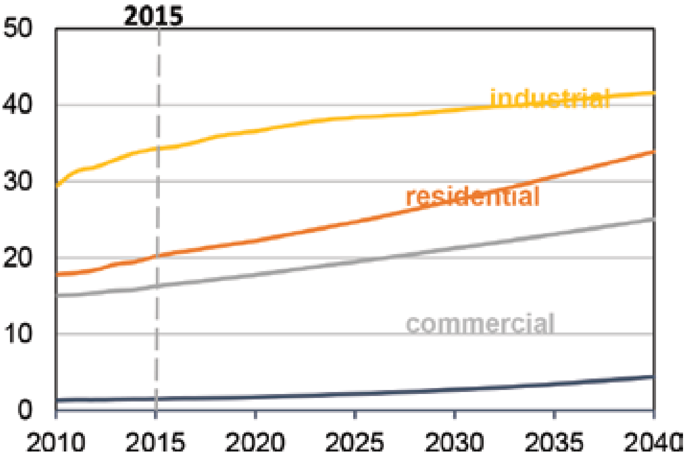

Figure 2. World electricity use by sector (quadrillion Btu).

usefulness, exergy is also a measure of its potential to cause change. Another area in which applications of exergy are increasing is that of economics. In the analysis and design of energy systems, techniques are often used that combine especially thermodynamics with economics to achieve optimal designs.

Currently, 80% of electricity in the world is approximately produced from fossil fuels (coal, petroleum, fuel oil, natural gas) and fired thermal power plants, while 20% of the electricity generation from renewable energy sources such as hydraulic, nuclear, wind, solar, geothermal, and biogas [20]. In recent decades, the exergetic performance analysis is found as a useful method in the design, evaluation, optimization, and improvement of nuclear and thermal power plants. The energetic and exergetic performance analyses are carried out for the existing coal-fired and nuclear power plants to identify and enhance the performance criteria.

3.1. Energy and exergy analyses of nuclear power plant

A detailed flow diagram for a nuclear power plant is shown in **Figure 3**. The diagram is divided into four main sections:

i. Steam generation: Heat is produced and used to generate and reheat steam. In the nuclear power plant, four natural circulation steam generators each produce steam.

ii. Power production: The steam produced in the steam generation section is passed through turbine generators that are attached to a transformer. Extraction steam from several points on the turbines preheats feedwater in several low-pressure and high-pressure heat exchangers and one deaerator.

iii. Condensation: Cooling water from the lake or sea condenses the steam exhausted from the turbines. The flow rate of cooling water is adjusted so that a specified temperature increase in the cooling water is achieved across the condenser.

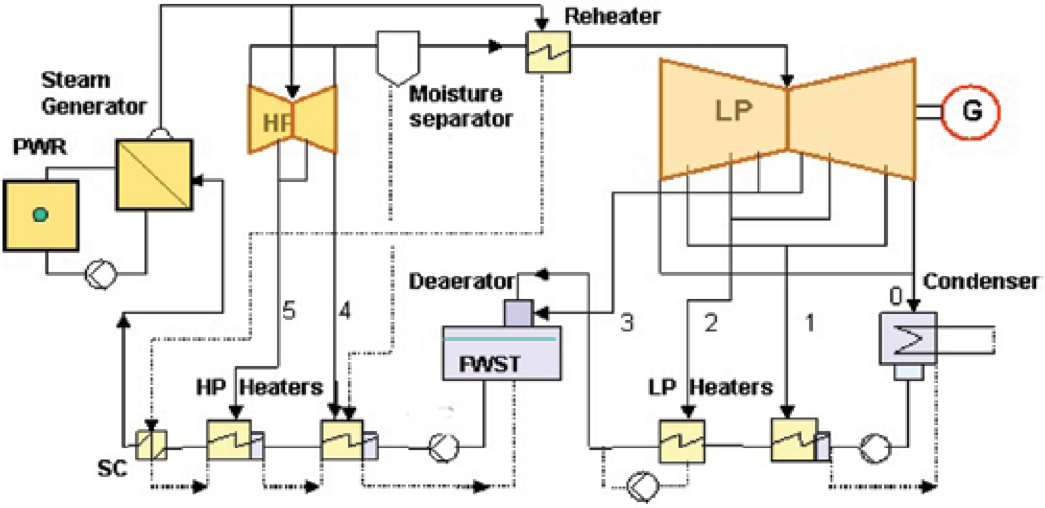

Figure 3. The simplified flow diagram of nuclear power plant.

iv. Preheating: The temperature and pressure of the feedwater are increased in a series of pumps and feedwater-heater heat exchangers.

A continuous mass flow diagram for one unit of the power plant modeled in this study includes the main components such as high- and low-pressure turbines, a reactor, pumps, a dearetor, a steam generator, a condenser, low- and high-pressure feed water heaters (**Figure 3**). The thermodynamic models are based on fundamental mass and energy balances. Using the energy and mass balance equations for each component in the power plant model, it is possible to compute energy and exergy contents in terms of turbine power outputs, pump power consumptions, energy and exergy flows at each node of the plants, first and second component efficiencies, and component irreversibilities in the plants [21].

There are three types of energy transfer, namely, works, heat transfer, and energy associated with mass transfer in open system. The first law of thermodynamics or energy balance for the steady flow process is given:

$$\dot{Q} - W_{cv} = \sum_e \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e \right) - \sum_i \dot{m}_i \left(h_i + \frac{\dot{V}_i^2}{2} + gz_i \right) \tag{13}$$

where Q is the heat transfer to system, W_{cv} the net work developed by the system, V is the bulk velocity of the working fluid, z is the altitude of the stream above the sea level, and g is the specific gravitational force.

Exergy flow equation for each part of the power plant is

$$\dot{E}x = \dot{m}[(h - h_0) - T_0(s - s_0)] \tag{14}$$

The second law can be expressed as

$$\dot{S}_g = \sum_e \dot{m}_e s_e - \sum_i \dot{m}_i s_i + \frac{\dot{Q}_s}{T_0} \tag{15}$$

where $\dot{S}_g$ represents the entropy generation rate for the process due to the irreversibilities. Two terms, namely, $\sum_i \dot{m}_i s_i$ and $\sum_e \dot{m}_e s_e$, are considered as the entropy transfer. $\dot{Q}_s = -\dot{Q}$ denotes the heat transfer rate for the instantaneous temperature T_0. The last term, $\dot{Q}_{sur}/T_0$, stands for the entropy transfer rate. The heat transfer above has been neglected as well as the kinetic, potential energies of the stream; one arrives at [21]

$$\dot{W}_{cv} = \dot{W}_u \cong \sum_i \dot{m}_i (h_i - T_0 s_i) - \sum_e \dot{m}_e (h_e - T_0 s_e) - T_0 \dot{S}_g \tag{16}$$

The reversible work can be obtained by adjusting the entropy generation term $\dot{S}_g = 0$:

$$\dot{W}_{rev} = \dot{W}_{u,\max} = \sum_i \dot{m}_i (h_i - T_0 s_i) - \sum_e \dot{m}_e (h_e - T_0 s_e) = \dot{m}[(h_i - h_e) - T_0(s_i - s_e)] \tag{17}$$

The reversible work can be obtained as

$$\dot{\mathrm{W}}_{\mathrm{u,\max}} = \sum_{\mathrm{i}} \dot{\mathrm{m}}_{\mathrm{i}} \mathrm{ex}_{\mathrm{i}} - \sum_{\mathrm{e}} \dot{\mathrm{m}}_{\mathrm{e}} \mathrm{ex}_{\mathrm{e}} = \sum_{\mathrm{i}} \dot{\mathrm{E}}\mathrm{x}_{\mathrm{i}} - \sum_{\mathrm{e}} \dot{\mathrm{E}}\mathrm{x}_{\mathrm{e}} \tag{18}$$

The rate of irreversibility $\left(\dot{I}\right)$ equals to exergy loss as

$$\dot{I} = \dot{W}_{u,\max} - \dot{W}_u = T_0 \dot{S}_g \tag{19}$$

This statement equals to the exergy destruction in the real process. The equations of energy and exergy can be applied for the nuclear power plants in order to find out the irreversibility rates for the processes in the reactor.

Energy and exergy balance for the adiabatic turbine system can be written as

$$\dot{W}_t = \left(\dot{W}_u\right)_t = \sum_i \dot{m}_i h_i - \sum_e \dot{m}_e h_e \tag{20}$$

The irreversibility rate as a measure of the exergy loss is

$$\dot{I}_t = \sum_i \dot{E}x_i - \sum_e \dot{E}x_e - \dot{W}_t = T_0 \left(\sum_e \dot{m}_e s_e - \sum_i \dot{m}_i s_i \right) \tag{21}$$

The isentropic efficiency of turbine is

$$\eta_t = \frac{\left(\dot{W}_u\right)_t}{\left(W_{u,\max}\right)_t} = \frac{W_t^{\cdot}}{\left(\dot{W}_{u,\max}\right)_t} \tag{22}$$

and the exergy efficiency of turbine is

$$\zeta_t = \frac{\dot{W}_t + \sum_e \dot{E}x_e}{\sum_i \dot{E}x_i} \tag{23}$$

The internal power input for the pump can be given as

$$\dot{W}_p = \dot{m}_i h_i - \dot{m}_e h_e \tag{24}$$

The rate of irreversibility of pump is

$$\dot{I}_p = \sum_i \dot{E}x_i - \sum_e \dot{E}x_e - \dot{W}_p \tag{25}$$

The isentropic efficiency of pump is

$$\eta_{p=} \frac{\left(\dot{W}_{u,rev}\right)_p}{\left(W_u\right)_p} = \frac{\left(\dot{W}_{u,rev}\right)_p}{\dot{W}_p} \tag{26}$$

and the second law efficiency is

$$\zeta_t = \frac{\dot{W}_t + \sum_e \dot{E}x_e}{\sum_i \dot{E}x_i} \tag{27}$$

The reversible work for the heat transfer equals to the rate of irreversibilities in the heaters, coolers, reheaters, condensers, and steam generators. The expression is

$$\dot{I}_{HT} = \dot{W}_{rev} - \dot{W}_u = \dot{W}_{rev} = \sum_i \dot{E}x_i - \sum_e \dot{E}x_e \tag{28}$$

Because $\dot{\mathrm{W}}_{\mathrm{u}} = 0$, there is no useful work produced during the heat transfer. The second law efficiency can be defined as

$$\zeta_{HT} = \frac{\sum_e \dot{E}x_e}{\sum_i \dot{E}x_i} \tag{29}$$

3.2. Energy and exergy analyses of coal-fired power plant

Coal-based thermal power plant generally operates on Rankine cycle as shown in **Figure 4** [22]. Similarly, there are four main sections: steam generation, power production, condensation, and preheating.

The thermodynamic models are based on fundamental mass, energy, and exergy balance equations. The first law of thermodynamics or energy balance is

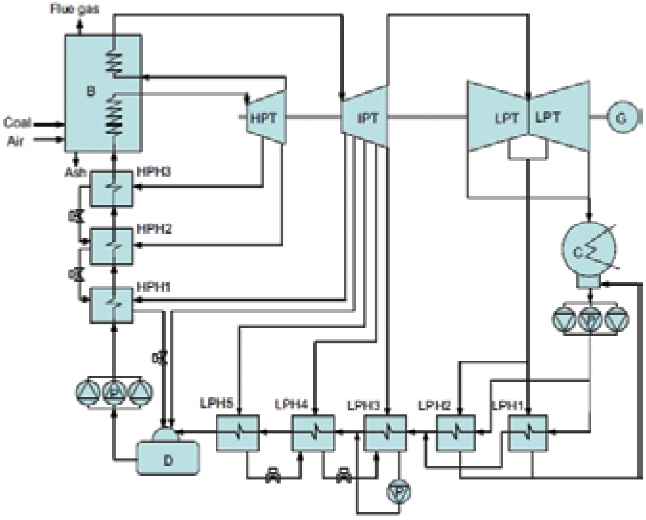

Figure 4. The simplified diagram of coal-fired thermal power plant.

$$\dot{Q} - W_{cv} = \sum_e \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e \right) - \sum_i \dot{m}_i \left(\frac{h_i + \dot{V}_i^2}{2 + gz_i} \right) \tag{30}$$

The energy of the first law efficiency of a system and/or system component is defined as the ratio of energy output to the energy input to system/component.

Entropy equation is analyzed with the following equation:

$$\dot{S}_g = \sum_e \dot{m}_e s_e - \sum_i \dot{m}_i s_i + \frac{\dot{Q}_s}{T_0} \tag{31}$$

where s_e, s_i, and S_g are the specific entropies at the inlet, outlet, and generation of the system, respectively. Balance equation for exergy is analyzed with following equation [22]:

$$\sum_{\mathrm{i}} \dot{\mathrm{m}}_{\mathrm{i}} \mathrm{ex}_{\mathrm{i}} - \sum_{\mathrm{e}} \dot{\mathrm{m}}_{\mathrm{e}} \mathrm{ex}_{\mathrm{e}} - \dot{\mathrm{E}}\mathrm{x}_{\mathrm{Des}} - \dot{\mathrm{E}}\mathrm{x}_{Heat} = \Delta \dot{\mathrm{E}}\mathrm{x}_{sys} \tag{32}$$

where ex_{i} and ex_{e} are the specific exergy at the inlet or outlet of the system components, respectively. Also, $\mathrm{Ex}_{\mathrm{Des}}$ and $\mathrm{Ex}_{\mathrm{heat}}$ are destructed exergy rate and exergy rate due to heat transfer, respectively.

$$\mathrm{ex} = \mathrm{ex}_{ph} - \mathrm{ex}_{\mathrm{ch}} \tag{33}$$

where $\mathrm{ex}_{\mathrm{ph}}$ and $\mathrm{ex}_{\mathrm{ch}}$ are the physical exergy and chemical exergy, respectively. The physical and chemical exergies can be defined as follows:

$$\mathrm{ex}_{ph} = (h_i - h_e) - T_0(s_i - s_e) \tag{34}$$

$$\mathrm{ex}_{ch} = \sum x_i ex_{ch}^0 + RT \sum x_i \ln (x_i) \tag{35}$$

The energy and exergy efficiencies of all system can be defined as the ratio of total useful output to the system input. General definitions for energy and exergy efficiencies are

$$\eta_{en} = \frac{En_e^{\cdot}}{En_i^{\cdot}} \tag{36}$$

$$\eta_{ex} = \frac{Ex_e^{\cdot}}{Ex_i^{\cdot}} \tag{37}$$

where subscripts En and Ex stand for energy and exergy per unit time, respectively. For the whole thermal power plant, the exergy efficiency is

$$\eta_{Ex} = \frac{\dot{W}_{Nett}}{m_{coal}.ex_{coal}} \tag{38}$$

The other important exergetic performance criterion defined in this study is the amount of exergy loss rate per unit power output, and it can be written as the following equation:

$$\zeta = \frac{\dot{E}x_D}{\dot{W}_{Nett}} \tag{39}$$

4. Conclusion

A major contribution of exergy analysis to the evaluation of a system is provided through a thermoeconomic evaluation that considers not only the inefficiencies but also the costs associated with these inefficiencies and compares the latter with the investment expenditures to reduce inefficiencies. In general, the major driving force for all advances in thermal and nuclear power plants is thermal efficiency. Thermal efficiencies of modern power plants are varied from average 30% to 45%. Also, an exergy analysis identifies and evaluates the thermodynamic inefficiencies of these plants. The costs associated with these inefficiencies should be analyzed by comparing these costs with the investment expenditures needed to reduce thermodynamic inefficiencies.

In the analyses, the developed model for each power plant using the mass, energy and exergy balance equations, and system and component performance criteria such as thermal efficiency, exergy efficiency, and exergy destruction have been determined and compared with each other. Using the energy analyses of the energy loss in the condenser seems higher, but the largest exergy losses occur in the reactor pressure vessel with the highest exergy destruction in the nuclear power plant. According to the analysis, the main exergy loss of the nuclear power plant stem from the reactor pressure vessel [21]. Indeed, it is the worse component in the whole NPP due to the irreversibility of the energy transformation. This result means that the NPP has the greatest potential to improve the plant efficiency in new pressure vessel components. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the reactor pressure vessel followed by the steam generator and then the turbine [21]. The first law analysis shows major energy loss that has been found to occur in condenser, while already its exergetic value is significantly low in coal-fired thermal plant. Also, the second law (exergy) analysis shows that the largest exergy losses occur in the combustion chamber or boiler [22].

Finally, there is a need for further work to develop a methodology for optimizing the power plant and to relate the overall efficiency to the rational efficiencies of its components or subsystems to include in some significant new conceptual developments and beneficial interpretations. An exergy method of optimization gives logical solution improving the power production opportunities in thermal and nuclear power plants.

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Exergetic Costs for Thermal Systems

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Additional information is available at the end of the chapter

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Abstract

Exergy costing to estimate the unit cost of products from various power plants and refrigeration system is discussed based on modified-productive structure analysis (MOPSA) method. MOPSA method provides explicit equations from which quick estimation of the unit cost of products produced in various power plants is possible. The unit cost of electricity generated by the gas-turbine power plant is proportional to the fuel cost and inversely proportional to the exergetic efficiency of the plant and is affected by the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of fuel. On the other hand, the unit cost of electricity from the organic Rankine cycle power plant with heat source as fuel is proportional to the unit cost of heat and the ratio of the monetary flow rate of non-fuel items to the generated electric power, independently. For refrigeration system, the unit cost of heat is proportional to the consumed electricity and inversely proportional to the coefficient of performance of the system, and is affected by the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of consumed electricity.

Keywords: exergy, thermoeconomics, unit exergy cost, power plant, refrigeration system

1. Introduction

Exergy analysis is an effective tool to accurately predict the thermodynamic performance of any energy system and the efficiency of the system components and to quantify the entropy generation of the components [1–3]. By this way, the location of irreversibilities in the system is determined. Furthermore, thermoeconomic analysis provides an opportunity to estimate the unit cost of products such as electricity and/or steam from thermal systems [4, 5] and quantifies monetary loss due to irreversibility for the components in the system [6]. Also, thermoeconomic analysis provides a tool for optimum design and operation of complex thermal systems such as cogeneration power plant [7] and efficient integration of new and renewable energy systems [8]. Recently, performance evaluation of various plants such as sugar plant [9], drying plant [10], and geothermal plant [11] has been done using exergy and thermoeconomic analyses. In this chapter, a procedure to obtain the unit cost of products from the power plants and refrigeration system is presented by using modified-productive structure analysis (MOPSA) method. The power plants considered in this chapter are gas-turbine power plant and organic Rankine cycle power plant. These systems generate electricity as a product by consuming the heat resultant from combustion of fuel and by obtaining heat from any hot stream as fuel, respectively. In addition, MOPSA method is applied to an air-cooled air conditioning system, which removes heat like a product while the consumed electricity is considered as fuel. Explicit equations to estimate the unit cost of electricity generated by the gas-turbine power plant and organic Rankine cycle plant, and the unit cost of heat for the refrigeration system are obtained and the results are presented.

2. A thermoeconomic method: modified productive structure analysis (MOPSA)

2.1. Exergy-balance and cost balance equations

A general exergy-balance equation that can be applied to any component of thermal systems may be formulated by utilizing the first and second law of thermodynamics [12]. Including the exergy losses due to heat transfer through the non-adiabatic components, and with decomposing the material stream into thermal and mechanical exergy streams, the general exergy-balance equation may be written as [6]

$$\dot{E}_x^{CHE} + \left(\sum_{inlet}\dot{E}_x^T - \sum_{outlet}\dot{E}_x^T\right) + \left(\sum_{inlet}\dot{E}_x^P - \sum_{outlet}\dot{E}_x^P\right) + T_o\left(\sum_{inlet}\dot{S}_i - \sum_{outlet}\dot{S}_i + \dot{Q}_{cv}/T_o\right) = \dot{E}_x^W \quad (1)$$

The fourth term in Eq. (1) is called the neg-entropy which represents the negative value of the rate of lost work due to entropy generation, which can be obtained from the second law of thermodynamics. The term $\dot{E}_x^{CHE}$ in Eq. (1) denotes the rate of exergy flow of fuel, and $\dot{Q}_{cv}$ in the fourth term denotes heat transfer interaction between a component and the environment, which can be obtained from the first law of thermodynamics.

$$\dot{Q}_{cv} + \sum_{in}\dot{H}_i = \sum_{out}\dot{H}_i + \dot{W}_{cv} \quad (2)$$

However, the quantity $\dot{Q}_{cv}$ for each component, which is usually not measured, may be obtained from the corresponding exergy-balance equation with the known values of the entropy flow rate at inlet and outlet.

Exergy, which is the ability to produce work, can be defined as the differences between the states of a stream or matter at any given particular temperature and pressure and the state of the same stream at a reference state. The exergy stream per unit mass is calculated by the following equation:

$$e_x = h(T,P) - h_{ref}\left(T_{ref}, P_{ref}\right) - T_o\left[s(T,P) - s_{ref}\left(T_{ref}, P_{ref}\right)\right] \tag{3}$$

where T is temperature, P is pressure, and the subscript ref denotes reference values. The exergy stream per unit mass can be divided into its thermal (T) and mechanical (P) components as follows [3]:

$$e_x = e_x^T + e_x^P \tag{4}$$

and

$$e_x^T = \left[h(T,P) - h\left(T_{ref}, P\right)\right] - T_o\left[s(T,P) - s\left(T_{ref}, P\right)\right] \tag{5}$$

$$e_x^P = \left[h\left(T_{ref}, P\right) - h_{ref}\left(T_{ref}, P_{ref}\right)\right] - T_o\left[s\left(T_{ref}, P\right) - s_{ref}\left(T_{ref}, P_{ref}\right)\right] \tag{6}$$

Assigning a unit exergy cost to every exergy stream, the cost-balance equation corresponding to the exergy-balance equation for any component in a thermal system [13] may be written as

$$\begin{aligned} &\dot{E}_x^{CHE} C_0 + \left(\sum_{inlet} \dot{E}_{x,i}^{T} - \sum_{outlet} \dot{E}_{x,i}^{T}\right) C_T + \left(\sum_{inlet} \dot{E}_{x,i}^{P} - \sum_{outlet} \dot{E}_{x,i}^{P}\right) C_P \\ &+ T_0\left(\sum_{inlet} \dot{S}_i - \sum_{outlet} \dot{S}_j + \dot{Q}_{cv}/T_o\right) C_S + \dot{Z}_k = \dot{E}_x^{W} C_W, \end{aligned} \tag{7}$$

The term $\dot{Z}_k$ includes all financial charges associated with owning and operating the kth component in the thermal system. We call the thermoeconomic analysis based on Eqs. (1) and (7) as modified-productive structure analysis (MOPSA) method because the cost-balance equation in Eq. (7) yields the productive structure of the thermal systems, as suggested and developed by Lozano and Valero [5] and Torres et al. [14]. MOPSA has been proved as very useful and powerful method in the exergy and thermoeconomic analysis of large and complex thermal systems such as a geothermal district heating system for buildings [15] and a high-temperature gas-cooled reactor coupled to a steam methane reforming plant [16]. Furthermore, the MOPSA can provide the interaction between the components in the power plant through the entropy flows [17] and a reliable diagnosis tool to find faulty components in power plants [18].

2.2. Levelized cost of system components

All costs due to owning and operating a plant depend on the type of financing, the required capital, the expected life of components, and the operating hours of the system. The annualized (levelized) cost method of Moran [1] was used to estimate the capital cost of components in this study. The amortization cost for a particular plant component is given by

$$PW = C_i - S_n PWF(i, n) \tag{8}$$

The present worth of the component is converted to annualized cost by using the capital recovery factor CRF(i, n):

$$\dot{C}(\$/year) = PW \cdot CRF(i, n) \tag{9}$$

The capital cost rate of the kth component of the thermal system can be obtained by dividing the levelized cost by annual operating hours δ.

$$\dot{Z}_k = \phi_k \dot{C}_k / 3600\delta \tag{10}$$

The maintenance cost is taken into consideration through the factor ϕ_k. It is noted that the operating hours of thermal systems is largely dependent on the energy demand patterns of end users [19].

3. Gas-turbine power plant

A schematic of a 300 MW gas-turbine power plant considered in this chapter is shown in **Figure 1**. The system includes five components: air compressor (1), combustor (2), gas turbines (3), fuel preheater (5), and fuel injector (6). A typical mass flow rate of fuel to the combustor at full load condition is 8.75 kg/s and the air–fuel mass ratio is about 50.0. Thermal and mechanical exergy flow rates and entropy flow rate at various state points shown in

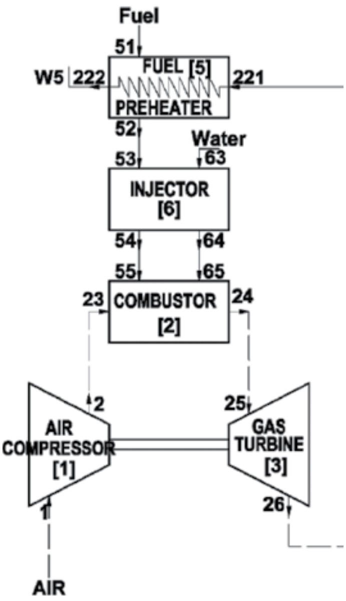

Figure 1. Schematic of a gas-turbine power plant.

States	$\dot{m}$ (kg/s)	P (MPa)	T $(°C)$	$\dot{E}_x^T$ (MW)	$\dot{E}_x^P$ (MW)	$\dot{S}$ (MW/K)
1	862.722	0.103	15.000	0.000	−0.558	0.121
2	862.722	1.025	323.589	88.176	164.572	0.193
23	862.722	1.025	323.589	88.176	164.572	0.193
24	891.056	1.025	1130.775	702.452	173.550	1.201
25	891.056	1.025	1130.775	702.452	173.550	1.201
26	891.056	0.107	592.700	261.996	2.661	1.262
51	17.500	0.103	15.000	0.000	0.018	0.001
52	17.500	0.103	185.000	1.563	0.018	0.018
53	17.500	0.103	185.000	1.563	0.018	0.018
54	17.500	1.025	415.314	7.735	5.337	0.021
55	17.500	1.025	415.314	7.735	5.337	0.021
63	10.833	0.103	(1.000)	6.064	0.000	0.004
64	10.833	1.025	418.176	12.338	0.010	0.006
65	10.883	1.025	418.176	12.338	0.010	0.006
221	11.111	3.540	220.100	2.417	0.038	0.028
222	11.111	3.540	72.941	0.239	0.038	0.011

Table 1. Property values and thermal, and mechanical exergy flows and entropy production rates at various state points in the gas-turbine power plant at 100% load condition.

Figure 1 are presented in **Table 1**. These flow rates were calculated based on the values of measured properties such as pressure, temperature, and mass flow rate at various state points.

3.1. Exergy-balance equation for gas-turbine power plant

The following exergy-balance equations can be obtained by applying the general exergy-balance equation given in Eq. (1) to each component in the gas-turbine power plant.

Air compressor

$$\left(\dot{E}_{x,1}^{T}-\dot{E}_{x,2}^{T}\right)+\left(\dot{E}_{x,1}^{P}-\dot{E}_{x,2}^{P}\right)+T_o\left[\left(\dot{S}_1-\dot{S}_2\right)+\dot{Q}_{[1]}/T_o\right]=\dot{E}_{x,[1]}^{W} \tag{11}$$

Combustor

$$\begin{aligned}\dot{E}_x^{CHE}&+\left(\dot{E}_{x,23}^{T}+\dot{E}_{x,55}^{T}+\dot{E}_{x,65}^{T}-\dot{E}_{x,24}^{T}\right)+\left(\dot{E}_{x,23}^{P}+\dot{E}_{x,55}^{P}+\dot{E}_{x,65}^{P}-\dot{E}_{x,24}^{P}\right)\\&+T_o\left[\dot{S}_{23}+\dot{S}_{55}+\dot{S}_{65}-\dot{S}_{24}+\dot{Q}_{[2]}/T_o\right]=0\end{aligned} \tag{12}$$

Turbine

$$\left(\dot{E}_{x,25}^{T}-\dot{E}_{x,26}^{T}\right)+\left(\dot{E}_{x,25}^{P}-\dot{E}_{26}^{P}\right)+T_o\left(\dot{S}_{25}-\dot{S}_{26}+\dot{Q}_{[3]}/T_o\right)=\dot{E}_{x,[3]}^{W} \tag{13}$$

Fuel preheater

$$\begin{aligned}&\left(\dot{E}^{T}_{x,51}-\dot{E}^{T}_{x,52}\right)+\left(\dot{E}^{P}_{x,51}-\dot{E}^{P}_{x,52}\right)+\left(\dot{E}^{T}_{x,221}-\dot{E}^{T}_{x,222}\right)+\left(\dot{E}^{P}_{x,221}-\dot{E}^{P}_{x,222}\right)\\&\quad+T_o\left[\dot{S}_{51}-\dot{S}_{52}+\dot{S}_{221}-\dot{S}_{222}+\dot{Q}_{[5]}/T_o\right]=0\end{aligned}\tag{14}$$

Steam injector

$$\begin{aligned}&\left(\dot{E}^{T}_{x,53}-\dot{E}^{T}_{x,54}+\dot{E}^{T}_{x,63}-\dot{E}^{T}_{x,64}\right)+\left(\dot{E}^{P}_{x,53}-\dot{E}^{P}_{x,54}+\dot{E}^{P}_{x,63}-\dot{E}^{P}_{x,64}\right)\\&\quad+T_o\left(\dot{S}_{53}-\dot{S}_{54}+\dot{S}_{63}-\dot{S}_{64}+\dot{Q}_{[6]}/T_o\right)=\dot{E}^{W}_{x,[6]}\end{aligned}\tag{15}$$

Boundary

$$\begin{aligned}&\left(\dot{E}^{T}_{x,1}+\dot{E}^{T}_{x,51}+\dot{E}^{T}_{x,63}-\dot{E}^{T}_{x,26}\right)+\left(\dot{E}^{P}_{x,1}+\dot{E}^{P}_{x,51}+\dot{E}^{P}_{x,63}-\dot{E}^{P}_{x,26}\right)+\left(\dot{E}^{T}_{x,221}-\dot{E}^{T}_{x,222}\right)\\&+\left(\dot{E}^{P}_{x,221}-\dot{E}^{P}_{x,222}\right)+T_o\left(\dot{S}_{1}+\dot{S}_{51}+\dot{S}_{63}-\dot{S}_{26}+\dot{S}_{221}-\dot{S}_{222}+\dot{Q}_{[boun]}/T_o\right)=0\end{aligned}\tag{16}$$

The net flow rates of the various exergies crossing the boundary of each component in the gas-turbine power plant at 100% load condition are shown in **Table 2**. Positives values of exergies indicate the exergy flow rate of "products," while negative values represent the exergy flow rate of "resources" or "fuel." The irreversibility rate due to entropy production in each component acts as a product in the exergy-balance equation. The sum of exergy flow rates of products and resources equals to zero for each component and the overall system; this zero sum indicates that perfect exergy balances are satisfied.

3.2. Cost-balance equation for gas-turbine power system

When the cost-balance equation is applied to a component, a new unit cost must be assigned to the component's principle product, whose unit cost is expressed as Gothic letter. After a unit

Component	Net exergy flow rates (MW)				Irreversibility rate (MW)
	$\dot{E}^{W}_{(k)}$	$\dot{E}^{CHE}_{x}$	$\dot{E}^{T}_{x}$	$\dot{E}^{P}_{x}$	
Compressor	−274.04	0.00	88.18	165.13	20.73
Combustor	0.00	−881.22	594.20	3.63	283.39
Gas turbine	593.74	0.00	−440.46	−170.89	17.61
Fuel preheater	0.00	0.00	−0.61	0.00	0.61
Steam injector	−18.68	0.00	11.91	5.33	1.44
Boundary	0.00	0.00	−253.22	−3.20	
Total	301.02	−881.22	253.22	3.20	323.78

Table 2. Exergy balances of each component in the gas-turbine power plant at 100% load condition.

cost is assigned to the principal product of each component, the cost-balance equations corresponding to the exergy-balance equations are as follows:

Air compressor

$$\left(\dot{E}^{T}_{x,1} - \dot{E}^{T}_{x,2}\right)C_T + \left(\dot{E}^{P}_{x,1} - \dot{E}^{P}_{x,2}\right)\mathbf{C_{1P}} + T_o\left[\left(\dot{S}_1 - \dot{S}_2\right) + \dot{Q}_{[1]}/T_o\right]C_S + \dot{Z}_{[1]} = \dot{E}^{W}_{[1]}C_W \quad (17)$$

Combustor

$$\begin{aligned}&\dot{E}^{CHE}_{x}C_o + \left(\dot{E}^{T}_{x,23} + \dot{E}^{T}_{x,55} + \dot{E}^{T}_{x,65} - \dot{E}^{T}_{x,24}\right)\mathbf{C_{2T}} + \left(\dot{E}^{P}_{x,23} + \dot{E}^{P}_{x,55} + \dot{E}^{P}_{x,65} - \dot{E}^{P}_{x,24}\right)C_P \\ &\quad + T_o\left[\dot{S}_{23} + \dot{S}_{55} + \dot{S}_{65} - \dot{S}_{24} + \dot{Q}_{[2]}/T_o\right]C_S + \dot{Z}_{[2]} = 0\end{aligned} \quad (18)$$

Turbine

$$\left(\dot{E}^{T}_{x,25} - \dot{E}^{T}_{x,26}\right)C_T + \left(\dot{E}^{P}_{x,25} - \dot{E}^{P}_{26}\right)C_P + T_o\left(\dot{S}_{25} - \dot{S}_{26} + \dot{Q}_{[3]}/T_o\right)C_S + \dot{Z}_{[3]} = \dot{E}^{W}_{[3]}\mathbf{C_W} \quad (19)$$

Fuel preheater

$$\begin{aligned}&\left(\dot{E}^{T}_{x,51} - \dot{E}^{T}_{x,52} + \dot{E}^{T}_{x,221} - \dot{E}^{T}_{x,222}\right)\mathbf{C_{5T}} + \left(\dot{E}^{P}_{x,51} - \dot{E}^{P}_{x,52} + \dot{E}^{P}_{x,221} - \dot{E}^{P}_{x,222}\right)C_P \\ &\quad + T_o\left[\dot{S}_{51} - \dot{S}_{52} + \dot{S}_{221} - \dot{S}_{222} + \dot{Q}_{[5]}/T_o\right]C_P + \dot{Z}_{[5]} = 0\end{aligned} \quad (20)$$

Steam injector

$$\begin{aligned}&\left(\dot{E}^{T}_{x,53} - \dot{E}^{T}_{x,54} + \dot{E}^{T}_{x,63} - \dot{E}^{T}_{x,64}\right)C_T + \left(\dot{E}^{P}_{x,53} - \dot{E}^{P}_{x,54} + \dot{E}^{P}_{x,63} - \dot{E}^{P}_{x,64}\right)\mathbf{C_{6P}} \\ &\quad + T_o\left(\dot{S}_{53} - \dot{S}_{54} + \dot{S}_{63} - \dot{S}_{64} + \dot{Q}_{[6]}/T_o\right)C_S + \dot{Z}_{[6]} = \dot{E}^{W}_{x,[6]}C_W\end{aligned} \quad (21)$$

Applying the general cost-balance equation to the system components, five cost-balance equations are derived. However, these equations present eight unknown unit exergy costs, which are C_T, C_S, C_W, C_{1P}, C_{2T}, C_P, C_{5T}, and C_{6P}. To calculate the value of these unknown unit exergy costs, three more cost-balance equations are required. These additional equations can be obtained from the thermal and mechanical junctions and boundary of the plant.

Thermal exergy junction

$$\begin{aligned}&\left(\dot{E}^{T}_{x,23} + \dot{E}^{T}_{x,55} + \dot{E}^{T}_{x,65} - \dot{E}^{T}_{x,24} + \dot{E}^{T}_{x,51} - \dot{E}^{T}_{x,52} + \dot{E}^{T}_{x,221} - \dot{E}^{T}_{x,222}\right)\mathbf{C_T} \\ &= \left(\dot{E}^{T}_{x,23} + \dot{E}^{T}_{x,55} + \dot{E}^{T}_{x,65} - \dot{E}^{T}_{x,24}\right)C_{2T} + \left(\dot{E}^{T}_{x,51} - \dot{E}^{T}_{x,52} + \dot{E}^{T}_{x,221} - \dot{E}^{T}_{x,222}\right)C_{5T}\end{aligned} \quad (22)$$

Mechanical exergy junction

$$\begin{aligned}&\left(\dot{E}^{P}_{x,1} - \dot{E}^{P}_{x,2} + \dot{E}^{P}_{x,53} - \dot{E}^{P}_{x,54} + \dot{E}^{P}_{x,63} - \dot{E}^{P}_{x,64}\right)\mathbf{C_P} = \left(\dot{E}^{P}_{x,1} - \dot{E}^{P}_{x,2}\right)C_{1P} \\ &\quad + \left(\dot{E}^{P}_{x,53} - \dot{E}^{P}_{x,54} + \dot{E}^{P}_{x,63} - \dot{E}^{P}_{x,64}\right)C_{6P}\end{aligned} \quad (23)$$

Boundary

$$\begin{aligned}&\left(\dot{E}^{T}_{x,1}+\dot{E}^{T}_{x,51}+\dot{E}^{T}_{x,63}-\dot{E}^{T}_{x,26}+\dot{E}^{T}_{x,221}-\dot{E}^{T}_{x,222}\right)C_T+\left(\dot{E}^{P}_{x,1}+\dot{E}^{P}_{x,51}+\dot{E}^{P}_{x,63}-\dot{E}^{P}_{x,26}+\dot{E}^{P}_{x,221}-\dot{E}^{P}_{x,222}\right)C_P\\&+T_o\left(\dot{S}_1+\dot{S}_{51}+\dot{S}_{63}-\dot{S}_{26}+\dot{S}_{221}-\dot{S}_{222}+\dot{Q}_{[boun]}/T_o\right)\mathbf{C_S}+\dot{Z}_{[boun]}=0\end{aligned} \tag{24}$$

In **Table 3**, initial investments, the annuities including the maintenance cost, and the corresponding monetary flow rates for each component are given. The cost flow rates corresponding to a component's exergy flow rates at 100% load condition are given in **Table 4**. The same sign convention for the cost flow rates related to products and resources was used as the case of exergy balances shown in **Table 2**. The lost cost due to the entropy production in a component is consumed cost. The fact that the sum of the cost flow rates of each component in the plant becomes zero, as verified in **Table 4**, shows that all the cost balances for the components are satisfied.

The overall cost-balance equation for the power system is simply obtained by summing Eqs. (17)–(24).

$$\dot{E}^{CHE}_{x}C_o+\sum_{i=1}^{n}\dot{Z}_{[i]}=\left(\dot{E}^{W}_{x,[1]}+\dot{E}^{W}_{x,[3]}+\dot{E}^{W}_{x,[6]}\right)C_W \tag{25}$$

From the above equation, the unit cost of electricity for the gas-turbine power system is given as [1]

$$C_W=\frac{C_o}{\eta_e}\left[1+\frac{\sum Z_{[i]}}{C_o\dot{E}^{CHE}_{x}}\right] \tag{26}$$

The production cost depends on fuel cost and the exergetic efficiency of the system, and is affected by the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of fuel. With the unit cost of fuel, C_o = 5.0 \$/GJ, an exergetic efficiency of the gas-turbine power

Component	Initial investment cost (US$10^6)	Annualized cost (×US$10^3/year)	Monetary flow rate (US$/h)
Compressor	36.976	4744.997	628.712
Combustor	2.169	278.340	36.880
Gas turbine	29.213	3748.799	496.716
Fuel preheater	7.487	960.780	127.303
Steam injector	14.787	1897.562	251.427
Total	90.542	11,630.478	1531.038

Table 3. Initial investments, annualized costs, and corresponding monetary flow rates of each component in the gas-turbine power plant.

Component	$\dot{C}_W$ (US$/h)	$\dot{C}_o$ (US$/h)	$\dot{C}_T$ (US$/h)	$\dot{C}_P$ (US$/h)	$\dot{C}_S$ (US$/h)	$\dot{Z}$ (US$/h)
Compressor	−17732.47	0.00	4217.91	15,071.00	−927.63	−628.71
Combustor	0.00	−15861.96	28238.85	341.28	−12681.19	−36.88
Gas turbine	38419.49	0.00	−21068.79	−16066.19	−787.79	−496.72
Fuel preheater	0.00	0.00	154.92	0.00	−27.52	−127.30
Steam injector	−1208.41	0.00	569.65	954.76	−64.44	−251.43
Boundary	0.00	0.00	−12112.54	−300.85	14488.57	−2075.18
Total	19478.61	−15861.96	0.00	0.00	0.00	−3616.22

Table 4. Cost flow rates of various exergies and neg-entropy of each component in the gas-turbine power plant at 100% load condition.

plant, 0.341, and a value of the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of fuel, 0.22, the unit cost of electricity estimated from Eq. (26) is approximately 17.97 $/GJ. However, one should solve Eqs. (17)–(24) simultaneously to obtain the unit cost of electricity and the lost cost flow rate occurred in each component.

4. Organic Rankine cycle power plant using heat as fuel

A schematic of the 20-kW ocean thermal energy conversion (OTEC) plant [20] operated by organic Rankine cycle, which is considered to apply MOPSA method, is illustrated in **Figure 2**. Five main components exist in the system: the evaporator (1), turbine (2), condenser (3), receiver tank (4) and pump (5). The refrigerant stream is heated by a heat source in the evaporator, and then the refrigerant stream is divided into two streams. A portion of this stream is passed through the throttling valve and reaches the receiver tank, while the remaining part of the refrigerant stream leaving from evaporator is sent to turbine. A portion of the stream flowing to turbine is throttled and bypassed to turbine outlet. The "pipes" are introduced into the analysis as a component to consider the heat and pressure losses in the pipes and the exergy removal during the throttling processes. Refrigerant of R32 is used as a working fluid in the organic Rankine cycle. At the full load condition, the mass flow rate of the refrigerant is 3.62 kg/s. The warm sea water having mass flow rate of 86.99 kg/s is used as a heat source for the plant, while the cold sea water having mass flow rate of 44.85 kg/s is used as a heat sink for the plant. The reference temperature and pressure for the refrigerant R32 are −40°C and 177.60 kPa, respectively. For water, the reference point was taken as 0.01°C, the triple point of water.

4.1. Exergy-balance equations for the organic Rankine cycle power plant

The exergy-balance equations obtained using Eq. (1) for each component in the organic Rankine cycle plant shown in **Figure 2** are as follows.

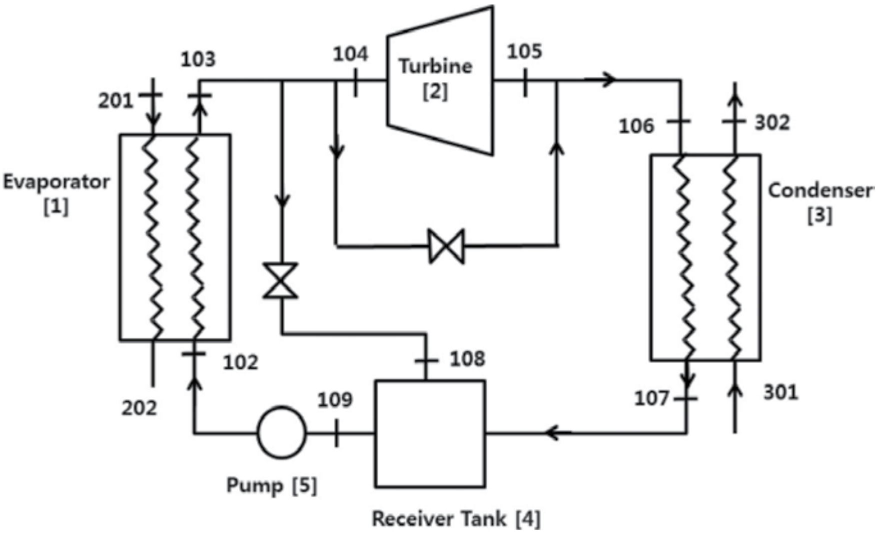

Figure 2. Schematic of an organic Rankine cycle power plant using warm water as a fuel.

Evaporator

$$\left(\dot{E}^{T}_{x,102} - \dot{E}^{T}_{x,103}\right) + \left(\dot{E}^{P}_{x,102} - \dot{E}^{P}_{x,103}\right) + \left(\dot{E}_{x,201} - \dot{E}_{x,202}\right) + T_o\left[\left(\dot{S}_{102} - \dot{S}_{103}\right) + \left(\dot{S}_{201} - \dot{S}_{202}\right) + \dot{Q}_{[1]}/T_o\right] = 0 \tag{27}$$

Turbine

$$\left(\dot{E}^{T}_{x,104} - \dot{E}^{T}_{x,105}\right) + \left(\dot{E}^{P}_{x,104} - \dot{E}^{P}_{x,105}\right) + T_o\left(\dot{S}_{104} - \dot{S}_{105} + \dot{Q}_{[2]}/T_o\right) = \dot{E}^{W}_{x,[2]} \tag{28}$$

Condenser

$$\left(\dot{E}^{T}_{x,106} - \dot{E}^{T}_{x,107}\right) + \left(\dot{E}^{P}_{x,106} - \dot{E}^{P}_{x,107}\right) + \left(\dot{E}_{x,301} - \dot{E}_{x,302}\right) + T_o\left[\left(\dot{S}_{106} - \dot{S}_{107}\right) + \left(\dot{S}_{301} - \dot{S}_{302}\right) + \dot{Q}_{[3]}/T_o\right] = 0 \tag{29}$$

Receiver tank

$$\left(\dot{E}^{T}_{x,107} + \dot{E}^{T}_{x,108} - \dot{E}^{T}_{x,109}\right) + \left(\dot{E}^{P}_{x,107} + \dot{E}^{T}_{x,108} - \dot{E}^{P}_{x,109}\right) + T_o\left(\dot{S}_{107} + \dot{S}_{108} - \dot{S}_{109} + \dot{Q}_{[4]}/T_o\right) = 0 \tag{30}$$

Pump

$$\left(\dot{E}^{T}_{x,109} - \dot{E}^{T}_{x,102}\right) + \left(\dot{E}^{P}_{x,109} - \dot{E}^{P}_{x,102}\right) + T_o\left(\dot{S}_{109} - \dot{S}_{102} + \dot{Q}_{[5]}/T_o\right) = \dot{E}^{W}_{x,[5]} \tag{31}$$

Pipes

$$\begin{aligned}&\left(\alpha\dot{E}^{T}_{x,103}-\dot{E}^{T}_{x,104}\right)+\left(\dot{E}^{T}_{x,105}-\dot{E}^{T}_{x,106}\right)+\left((1-\alpha)\dot{E}^{T}_{x,103}-\dot{E}^{T}_{x,108}\right)\\&+\left(\alpha\dot{E}^{P}_{x,103}-\dot{E}^{P}_{x,104}\right)+\left(\dot{E}^{P}_{x,105}-\dot{E}^{P}_{x,106}\right)+\left((1-\alpha)\dot{E}^{P}_{x,103}-\dot{E}^{P}_{x,108}\right)\\&+T_o\left(\dot{S}_{103}+\dot{S}_{105}-\dot{S}_{104}-\dot{S}_{106}-\dot{S}_{108}+\dot{Q}_{pipes}/T_o\right)=0\end{aligned}\tag{32}$$

Boundary

$$\begin{aligned}&-\left(\dot{E}_{x,201}-\dot{E}_{x,202}\right)-\left(\dot{E}_{x,301}-\dot{E}_{x,302}\right)\\&-T_o\left(\dot{S}_{201}-\dot{S}_{202}+\dot{S}_{301}-\dot{S}_{302}+\dot{Q}_{boun}/T_o\right)=0\end{aligned}\tag{33}$$

The α term given in Eq. (32) is the ratio of the bypass streams from state 103 to 108. The value of the α term can be calculated by applying the mass and energy conservation equations to the receiver tank. The stream bypassed from state 103 to 105 may be neglected. An example of exergy calculation for the organic Rankine cycle plant using a stream of warm water at 28°C as a heat source to the evaporator [20] is shown in **Table 5**. As mentioned in the previous section, a positive value of exergy flow rate represents "product," while a negative value of exergy flow rate indicates "fuel." The last two columns clearly indicate that the electricity comes from expenditure of heat input.

4.2. Cost-balance equations for the organic Rankine cycle power plant

By assigning a unit cost to every thermal exergy of the refrigerant stream (C_{1T}, C_{2T}, C_{3T}, and C_T), mechanical exergy for the refrigerant stream (C_P), cold water (C_3), neg-entropy (C_s), and electricity (C_W), the cost-balance equations corresponding to the exergy-balance equations which are Eqs. (27)–(33) are given as follows. When the cost-balance equation is applied to a specific component, one may assign a unit cost to its main product, which is represented by a Gothic letter.

Component	Refrigerant	Water stream	Irreversibility rate	Heat transfer rate	Work input/output rate
Evaporator	224.59	−233.21	17.52	−8.90	—
Turbine	−24.24	—	3.31	0.83	20.10
Condenser	−178.00	171.26	5.22	1.51	—
Receiver tank	−2.52	—	−11.68	14.20	—
Pump	1.50	—	1.69	−0.15	−3.04
Pipes	−21.33	—	20.31	1.02	—
Boundary	—	61.95	−36.36	−25.58	—
Total	0.00	0.00	0.00	−17.06	17.06

Table 5. Exergy balances for each component in the organic Rankine cycle plant (Unit: kW) [20].

Evaporator

$$\left(\dot{E}^{T}_{x,102}-\dot{E}^{T}_{x,103}\right)\mathbf{C_{1T}}+\left(\dot{E}^{P}_{x,102}-\dot{E}^{P}_{x,103}\right)C_P+\left(\dot{E}_{x,201}-\dot{E}_{x,202}\right)C_2 \\ +T_o\left[\left(\dot{S}_{102}-\dot{S}_{103}\right)+\left(\dot{S}_{201}-\dot{S}_{202}\right)+\dot{Q}_{[1]}/T_o\right]C_S+\dot{Z}_{[1]}=0 \tag{34}$$

Turbine

$$\left(\dot{E}^{T}_{x,104}-\dot{E}^{T}_{x,105}\right)C_T+\left(\dot{E}^{P}_{x,104}-\dot{E}^{P}_{x,105}\right)C_P \\ +T_o\left(\dot{S}_{104}-\dot{S}_{105}+\dot{Q}_{[2]}/T_o\right)C_S+\dot{Z}_{[2]}=\dot{E}^{W}_{x,[2]}\mathbf{C_W} \tag{35}$$

Condenser

$$\left(\dot{E}^{T}_{x,106}-\dot{E}^{T}_{x,107}\right)C_T+\left(\dot{E}^{P}_{x,106}-\dot{E}^{P}_{x,107}\right)C_P+\left(\dot{E}_{x,301}-\dot{E}_{x,302}\right)\mathbf{C_3} \\ +T_o\left[\left(\dot{S}_{106}-\dot{S}_{107}\right)+\left(\dot{S}_{301}-\dot{S}_{302}\right)+\dot{Q}_{[3]}/T_o\right]C_S+\dot{Z}_{[3]}=0 \tag{36}$$

Receiver tank

$$\left(\dot{E}^{T}_{x,107}+\dot{E}^{T}_{x,108}-\dot{E}^{T}_{x,109}\right)\mathbf{C_{2T}}+\left(\dot{E}^{P}_{x,107}+\dot{E}^{P}_{x,108}-\dot{E}^{P}_{x,109}\right)C_P \\ +T_o\left(\dot{S}_{107}+\dot{S}_{108}-\dot{S}_{109}+\dot{Q}_{[4]}/T_o\right)C_S+\dot{Z}_{[4]}=0 \tag{37}$$

Pump

$$\left(\dot{E}^{T}_{x,109}-\dot{E}^{T}_{x,102}\right)C_T+\left(\dot{E}^{P}_{x,109}-\dot{E}^{P}_{x,102}\right)\mathbf{C_P} \\ +T_o\left(\dot{S}_{109}-\dot{S}_{102}+\dot{Q}_{[5]}/T_o\right)C_S+\dot{Z}_{[5]}=\dot{E}^{W}_{x,[5]}C_W \tag{38}$$

Pipes

$$\left[\left(\alpha\dot{E}^{T}_{x,103}-\dot{E}^{T}_{x,104}\right)+\left(\dot{E}^{T}_{x,105}-\dot{E}^{T}_{x,106}\right)+\left((1-\alpha)\dot{E}^{T}_{x,103}-\dot{E}^{T}_{x,108}\right)\right]\mathbf{C_{3T}} \\ +\left[\left(\alpha\dot{E}^{P}_{x,103}-\dot{E}^{P}_{x,104}\right)+\left(\dot{E}^{P}_{x,105}-\dot{E}^{P}_{x,106}\right)+\left((1-\alpha)\dot{E}^{P}_{x,103}-\dot{E}^{P}_{x,108}\right)\right]C_P \\ +T_o\left(\dot{S}_{103}+\dot{S}_{105}-\dot{S}_{104}-\dot{S}_{106}-\dot{S}_{108}+\dot{Q}_{pipes}/T_o\right)C_s+\dot{Z}_{pipes}=0 \tag{39}$$

Boundary

$$-\left(\dot{E}_{x,201}-\dot{E}_{x,202}\right)C_2-\left(\dot{E}_{x,301}-\dot{E}_{x,302}\right)C_3 \\ -T_o\left(\dot{S}_{201}-\dot{S}_{202}+\dot{S}_{301}-\dot{S}_{302}+\dot{Q}_{boun}/T_o\right)\mathbf{C}_S+\dot{Z}_{boun}=0 \tag{40}$$

Seven cost-balance equations for the five components of the plant, pipes, and the boundary were derived with eight unknown unit exergy costs of C_{1T}, C_{2T}, C_{3T}, C_T, C_P, C_3, C_S, and C_W. We can obtain an additional cost-balance equation for the junction of thermal exergy of the refrigerant stream.

Thermal junction

$$\begin{aligned}&\left(\dot{E}^T_{x,102}-\dot{E}^T_{x,103}\right)C_{1T}+\left(\dot{E}^T_{x,107}+\dot{E}^T_{x,108}-\dot{E}^T_{x,109}\right)C_{2T}\\&+\left(\dot{E}^T_{x,103}+\dot{E}^T_{x,105}-\dot{E}^T_{x,104}-\dot{E}^T_{x,106}-\dot{E}^T_{x,108}\right)C_{3T}\\&=\left[\dot{E}^T_{x,102}+\dot{E}^T_{x,105}+\dot{E}^T_{x,107}-\dot{E}^T_{x,104}-\dot{E}^T_{x,106}-\dot{E}^T_{x,109}\right]\mathbf{C_T}\end{aligned}\tag{41}$$

With Eq. (41), we have all the necessary cost-balance equations to calculate the unit cost of all exergies (C_{1T}, C_{2T}, C_{3T}, C_T, and C_3, neg-entropy (C_s) and a product (electricity, C_W) by input (given) of thermal energy (C_2) to the evaporator. The overall cost-balance equation for the Rankine power plant can be obtained by summing Eqs. (34)–(41), which is given by

$$abs\left[\sum\dot{C}_H+\left(\sum\dot{Z}_k+\dot{Z}_{boun}\right)\right]=\dot{E}^W_x C_W\tag{42}$$

where $\sum\dot{C}_H=\sum\dot{Q}_k C_S$ is the net cost flow rate due to the heat transfer to/from the organic Rankine cycle plant. The term $\dot{Z}_{boun}$ in Eq. (42) may represent the cost flow rate related to the construction of the plant [6]. Rewriting Eq. (42), we have the unit cost of electricity from the Rankine cycle power plant.

$$C_W=abs\left[\sum\dot{C}_H+\left(\sum\dot{Z}_k+\dot{Z}_{boun}\right)\right]/\dot{E}^W_x\tag{43}$$

where $\dot{E}^W_x$ is the net electricity obtained from the organic Rankine cycle plant and abs denotes the absolute value of the quantity in parentheses.

Figure 3 shows that the unit cost of electricity from the organic Rankine cycle plant and the net cost flow rate due to the heat transfer rate to the plant vary depending on the unit cost of warm water in the evaporator, C_2, appeared in Eq. (34). As the unit cost of warm water increases, the net cost flow rate due to heat transfer to the plant decreases while the unit cost of electricity increases. The cross point between the line for the unit cost of electricity and the line for the total cost flow rate due to heat transfer determines unit cost of electricity. The unit cost of electricity and the net cost flow rate due to heat transfer for a case whose detailed calculation results shown in **Table 6** are \$0.205 and −\$0.941/kWh, respectively. The value of the unit cost C_2 appeared in the cost balance equation, Eq. (34), is approximately \$0.117/kWh for this particular case, which may be considered as a fictional one.

Detailed calculation results reveal that the unit cost of electricity from an organic Rankine cycle plant can be obtained from the following equation:

$$C_W=C_2'+\left(\sum\dot{Z}_k+\dot{Z}_{boun}\right)/\dot{E}^W_x\tag{44}$$

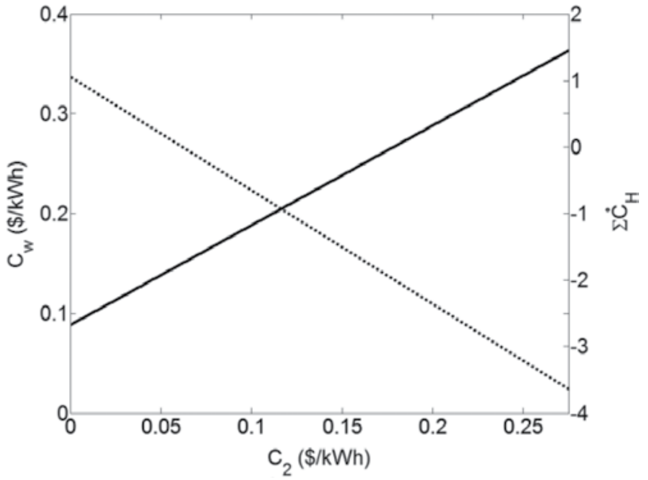

Figure 3. Unit cost of electricity, C_W (solid lines), and net cost flow rate due to heat transfer to the plant, $\sum \dot{C}_H$ (dotted line), depending on the unit cost of supplied hot water to evaporator C_2, for the case shown in **Table 6**.

Component	$\dot{C}_T$	$\dot{C}_P$	$\dot{C}_S$	$\dot{C}_H$	$\dot{C}_W$	$\dot{C}_{wsw}$	$\dot{C}_{dsw}$	$\dot{Z}_k$
Evaporator	27.502	−0.026	0.967	−0.491	—	−27.285	—	−0.666
Turbine	−3.101	−0.613	0.183	0.046	4.126	—	—	−0.640
Condenser	−23.569	−0.004	0.288	0.084	—	—	23.867	−0.666
Receiver tank	0.021	0.012	−0.645	0.784	—	—	—	−0.172
Pump	0.079	0.678	0.093	−0.008	−0.624	—	—	−0.218
Pipes	−0.932	−0.048	1.121	0.056	—	—	—	−0.198
Boundary	—	—	−2.006	−1.412	—	27.285	−23.867	—
Total	−0.000	0.000	−0.000	−0.941	3.502	—	—	−2.561

Using hot water from an incinerator plant as the heat source, C_2 =\$0.117/kWh, C_2' =\$0.055/kWh [20].
Solutions of cost-balance equations [Unit:\$/kWh].
$C_{1T} = 0.122$, $C_{2T} = -0.008$, $C_{3T} = 0.044$, $C_T = 0.132$, $C_P = 0.750$, $C_3 = 0.139$, $C_W = 0.205$, $C_S = 0.055$.

Table 6. Cost flow rates of various exergies, lost work rate due to heat transfer, heat transfer rate, and work input/out rate of each component in the organic Rankine cycle plant (Unit: \$/h).

From Eqs. (43) and (44), one can deduce that

$$C_2' = abs\left(\sum \dot{C}_H\right)/\dot{E}_x^W \tag{45}$$

The calculated value of C_2' using Eq. (45) is approximately \$0.055/kWh, which is quite different from the C_2 = \$0.177/kWh, a value determined from **Figure 3**. The value of C_2' which is the ratio of the absolute value of net cost flow rate of heat to the produced electricity was found to be a real unit cost of hot water stream [20]. Equation (44) tells us that the unit cost of electricity from the organic Rankine cycle is determined by the sum of the unit cost of heat and the ratio of the monetary flow rate of non-fuel items to the produced electric power.

5. 200 kW air-cooled air conditioning unit

Even though the performance evaluation of a household refrigerator using thermoeconomics was performed [21], estimation of the unit cost of heat supplied to the room by air conditioning unit was never tried. In this section, the unit cost of heat for a 120-kW air-cooled air conditioning unit is obtained, which is helpful for the cost comparison between air conditioning unit operated by electricity and absorption refrigeration system running by heat [22].

5.1. Exergy-balance equations for the air conditioning units

The exergy-balance equations obtained using Eq. (1) for each component in an air-cooled air conditioning units shown in **Figure 4** are as follows. The heat transfer interactions with environment for the compressor, TXV, and suction line are neglected.

Compressor

$$\left(\dot{E}_{x,1}^{r,T}-\dot{E}_{x,2}^{r,T}\right)+\left(\dot{E}_{x,1}^{r,P}-\dot{E}_{x,2}^{r,P}\right)+T_o\left(\dot{S}_1^r-\dot{S}_2^r\right)=E_{x,comp}^{W} \tag{46}$$

Condenser

$$\left(\dot{E}_{x,2}^{r,T}-\dot{E}_{x,3}^{r,T}\right)+\left(\dot{E}_{x,2}^{r,P}-\dot{E}_{x,3}^{r,P}\right)+\left(\dot{E}_{x,7}^{a}-\dot{E}_{x,8}^{a}\right)+T_o\left(\dot{S}_2^r-\dot{S}_3^r+\dot{S}_7^a-\dot{S}_8^a+\dot{Q}_{con}/T_o\right)=0 \tag{47}$$

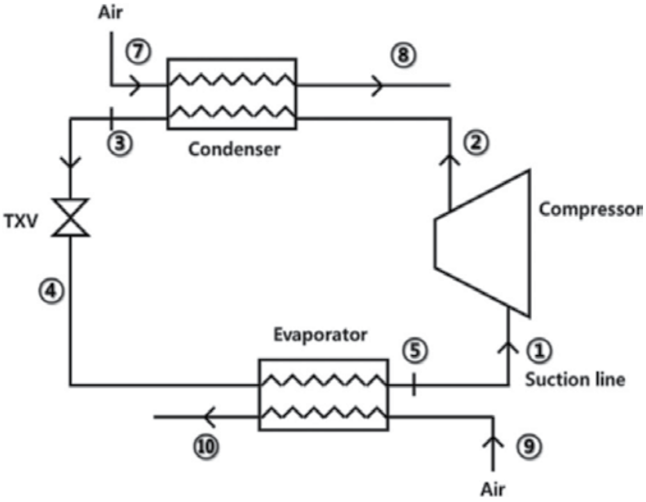

Figure 4. Schematic of a 120-kW air-cooled air conditioning system.

TXV

$$\left(\dot{E}^{r,T}_{x,3} - \dot{E}^{r,T}_{x,4}\right) + \left(\dot{E}^{r,P}_{x,3} - \dot{E}^{r,P}_{x,4}\right) + T_o\left(\dot{S}^{r}_{3} - \dot{S}^{r}_{4}\right) = 0 \tag{48}$$

Evaporator

$$\left(\dot{E}^{r,T}_{x,4} - \dot{E}^{r,T}_{x,5}\right) + \left(\dot{E}^{r,P}_{x,4} - \dot{E}^{r,P}_{x,5}\right) + \left(\dot{E}^{a}_{x,9} - \dot{E}^{a}_{x,10}\right) + T_o\left(\dot{S}^{r}_{4} - \dot{S}^{r}_{5} + \dot{S}^{a}_{9} - \dot{S}^{a}_{10} + \dot{Q}_{evap}/T_o\right) = 0 \tag{49}$$

Suction line

$$\left(\dot{E}^{r,T}_{x,5} - \dot{E}^{r,T}_{x,1}\right) + \left(\dot{E}^{r,P}_{x,5} - \dot{E}^{r,P}_{x,1}\right) + T_o\left(\dot{S}^{r}_{5} - \dot{S}^{r}_{1}\right) = 0 \tag{50}$$

Superscripts r and a given in the above equations represent the fluid stream of the refrigerant and air, respectively, and W denotes work. The amount of heat transferred to the environment in each component was neglected in the exergy-balance equations.

In Eqs. (47) and (49), the difference in the exergy and entropy for air stream is just the difference in the enthalpy so that these terms can be written with help of Eq. (2) as

$$\left(\dot{E}^{a}_{x,7} - \dot{E}^{a}_{x,8}\right) + T_o\left(\dot{S}^{a}_{7} - \dot{S}^{a}_{8}\right) = \left(\dot{H}^{a}_{7} - \dot{H}^{a}_{8}\right) = -\dot{Q}^{H}_{env} \tag{51}$$

$$\dot{E}^{a}_{x,9} - \dot{E}^{a}_{x,10} + T_o\left(\dot{S}^{a}_{9} - \dot{S}^{a}_{10}\right) = \left(\dot{H}^{a}_{9} - \dot{H}^{a}_{10}\right) = \dot{Q}^{H}_{room} \tag{52}$$

The deposition of heat into the environment and the heat transferred to room are hardly considered to be dissipated to the environment. For such heat delivery system, it may be reasonable that the delivered heat rather than its exergy is contained in the exergy-balance equation. With help of Eqs. (51) and (52), the exergy-balance equation for the condenser and evaporator become

$$\left[\left(\dot{E}^{r,T}_{x,2} - \dot{E}^{r,T}_{x,3}\right) + \left(\dot{E}^{r,P}_{x,2} - \dot{E}^{r,P}_{x,3}\right) + T_o\left(\dot{S}^{r}_{2} - \dot{S}^{r}_{3}\right)\right] + \left[-\dot{Q}^{H}_{env} + \dot{Q}_{con}\right] = 0 \tag{47′}$$

$$\left[\left(\dot{E}^{r,T}_{x,4} - \dot{E}^{r,T}_{x,5}\right) + \left(\dot{E}^{r,P}_{x,4} - \dot{E}^{r,P}_{x,5}\right) + T_o\left(\dot{S}^{r}_{4} - \dot{S}^{r}_{5}\right)\right] + \left[\dot{Q}^{H}_{room} + \dot{Q}_{evap}\right] = 0 \tag{49′}$$

The terms, $\dot{Q}_{con}$ in Eq. (47′) and $\dot{Q}_{evap}$ in Eq. (49′) represent the irreversibility corresponding to the terms $-\dot{Q}^{H}_{env}$ and $\dot{Q}^{H}_{room}$, respectively. The pair terms in the second bracket in Eqs. (47′) and (49′) are equal in magnitude but opposite in sign to vanish completely because the terms in the first bracket in those equations vanish. This assumption is legitimate since the entropy generation due to the heat transfer between flow streams [23] in the condenser and evaporator was calculated to be negligibly small.

The simulated data for the difference in the thermal and mechanical exergy flow rates at each component under normal operation for a 120-kW air-cooled air conditioning system [24] is displayed in **Table 7**. The cooling capacity of the system ($\dot{Q}^{H}_{room}$) is considered as the heat

Component	$\Delta\dot{E}_x^{T,r}$	$\Delta\dot{E}_x^{P,r}$	$\dot{Q}_H$	$\dot{E}_x^W$	$\dot{I}$
Compressor	0.18	22.85		−32.10	9.07
Condenser	−8.95	−0.09	(−88.92)		(88.92) 9.04
TXV	18.29	−21.97			3.68
Evaporator	−9.52	−0.66	(121.02)		(−121.02) 10.18
Suction line		−0.13			0.13
Total	0.0	0.0	32.10	−32.10	0.0

The numerical values in parentheses are the heat flow rate of air (third column) and the corresponding lost work rate (fifth column).

Table 7. Exergy balances for each component in the 120-kW air conditioning system at normal operation (Unit: kW).

gained by the refrigerant in the evaporator. However, the heat output to the environment through the condenser was taken to satisfy the exergy-balance equation for the condenser as well as the overall system. The irreversibility rate due to the entropy generation at each component was calculated using the exergy-balance equations for each component given from Eq. (46) to (50). The values in the parentheses in the third and fifth columns represent the first and second quantity inside the second bracket in Eqs. (47′) and (49′), respectively. Note that minus and plus sign indicate the resource or fuel and product exergies, respectively, as usual. The sign of the irreversibility rate is minus at the evaporator, while it is plus at other units which play as boundary.

5.2. Cost-balance equations for the air-cooled air conditioning units

By assigning a unit cost to every thermal and mechanical exergy stream of the refrigerant (C_T, C_P), lost work (C_S), heat (C_H), and work (C_W), the cost-balance equations corresponding to the exergy-balance equations, i.e., Eqs. (46), (47′), (48), (49′), and (50), are as follows. In this particular thermal system, a unit to a principal product for each component is not applied because the working fluid that flows through all the components makes a thermodynamic cycle.

Compressor

$$\left(\dot{E}_{x,1}^{r,T}-\dot{E}_{x,2}^{r,T}\right)C_T+\left(\dot{E}_{x,1}^{r,P}-\dot{E}_{x,2}^{r,P}\right)C_P+T_o\left(\dot{S}_1^r-\dot{S}_2^r\right)C_S+\dot{Z}_{comp}=\dot{E}_{x,comp}^{W}C_W \tag{53}$$

Condenser

$$\left(\dot{E}_{x,2}^{r,T}-\dot{E}_{x,3}^{r,T}\right)\mathrm{C_{1T}}+\left(\dot{E}_{x,2}^{r,P}-\dot{E}_{x,3}^{r,P}\right)C_P-\dot{Q}_{env}^{H}\cdot 0+\left[T_o\left(\dot{S}_2^r-\dot{S}_3^r\right)+\dot{Q}_{con}\right)]C_S+\dot{Z}_{con}=0 \tag{54}$$

TXV

$$\left(\dot{E}_{x,3}^{r,T}-\dot{E}_{x,4}^{r,T}\right)C_T+\left(\dot{E}_{x,3}^{r,P}-\dot{E}_{x,4}^{r,P}\right)C_P+T_o\left(\dot{S}_3^r-\dot{S}_4^r\right)C_S=0 \tag{55}$$

Evaporator

$$\left(\dot{E}_{x,4}^{r,T} - \dot{E}_{x,5}^{r,T}\right)C_T + \left(\dot{E}_{x,4}^{r,P} - \dot{E}_{x,5}^{r,P}\right)C_P + \dot{Q}_{room}^{H}\mathrm{C}_{\mathrm{H}} + \left[T_o\left(\dot{S}_4^{r} - \dot{S}_5^{r}\right) + \dot{Q}_{evap}\right]C_S + \dot{Z}_{eva} = 0 \qquad (56)$$

Suction line

$$\left(\dot{E}_{x,5}^{P,r} - \dot{E}_{x,1}^{P,r}\right)C_P + T_o\left[\dot{S}_5^{r} - \dot{S}_1^{r}\right]C_s + \dot{Z}_{sl} = 0 \qquad (57)$$

We now have five cost-balance equations to calculate two unit costs of exergies (C_{T} and C_{P}), neg-entropy (C_{S}), and a product, heat (C_{H}) by input of electricity (C_{W}). So, it is better to combine the cost-balance equation for the evaporator and suction line, which can be written as

$$\left(\dot{E}_{x,4}^{r,T} - \dot{E}_{x,1}^{r,T}\right)C_T + \left(\dot{E}_{x,4}^{r,P} - \dot{E}_{x,1}^{r,P}\right)C_P + \dot{Q}_{room}^{H}C_H + \left[T_o\left(\dot{S}_4^{r} - \dot{S}_1^{r}\right) + \dot{Q}_{evap}\right]C_S$$

$$+\left(\dot{Z}_{evap} + \dot{Z}_{sl}\right) = 0 \qquad (58)$$

The overall cost-balance equation for the air conditioning units can be obtained by summing Eqs. (53)–(55) and (58);

$$\dot{Q}_{room}^{H}C_H = \sum \dot{Z}_k + \dot{E}_x^{W}C_W \qquad (59)$$

Table 8 lists the initial investment, the annuities including the maintenance cost, and the corresponding monetary flow rates for each component of the air-cooled air conditioning system. Currently, the installation cost of an air-cooled air conditioning system with a 120-kW cooling capacity is approximately $17,000 in Korea. The levelized cost of the air conditioning units was calculated to be 0.3122$/h with an expected life of 20 years, an interest rate of 5% and salvage value of $850. The operating hours of the air conditioning system, which is crucial in determining the levelized cost, were taken as 4500 h. The maintenance cost was taken as 5% of the annual levelized cost of the system.

The cost flow rates of various exergies and irreversibility rate at each component in the air conditioning system at the normal operation are shown in **Table 9**. The sign convention for the cost flow rates is that minus and plus signs indicate the resource and product cost flow rates, respectively. Erroneously, reverse sign convention was used in their study on the thermoeconomic

Component	Initial investment ($)	Annualized cost ($/year)	Monetary flow rate ($/h)
Compressor	5000	393.4	0.0918
Condenser	4000	314.8	0.0735
TXV	2000	157.4	0.0367
Evaporator + Suction line	6000	472.1	0.1102
Total	17,000	1337.7	0.3122

Table 8. Initial investments, annualized costs, and corresponding monetary flow rates of each component in air conditioning system with a 120-kW capacity.

Component	$\dot{C}_T$	$\dot{C}_P$	$\dot{C}_H$	$\dot{C}_W$	$\dot{C}_S$	$\dot{Z}$
Compressor	0.03506	3.73914		−3.8520	0.16960	−0.09180
Condenser	−1.74352	−0.01473			1.83175	−0.07350
TXV	3.56302	−3.59513			0.06881	−0.03670
Evaporator+ Suction line	−1.85456	−0.12928	4.16420		−2.07016	−0.11020
Total	0.0	0.0	4.16420	−3.8520	0.0	−0.3122

The unit cost of irreversibility, C_S is 0.00187 \$/kWh and the unit cost of cooling capacity, C_H is 0.0344\$/kWh

Table 9. Cost flow rates of various exergies and irreversibility of each component in the air conditioning unit at normal operation (Unit: \$/h).

analysis of ground-source heat pump systems [25]. The lost cost flow rate due to the entropy generation appears as consumed cost in the evaporator; on the other hand, it appears as production cost in other components. The unit cost of heat delivered to the room or the unit cost of the cooling capacity is estimated to be 0.0344\$/kWh by solving the four cost-balance equations given from Eqs. (53) to (59) with unit cost of electricity of 0.120 \$/kWh. The unit cost of thermal and mechanical exergies and the irreversibility are $C_T = 0.1948$, $C_P = 0.1636$, and $C_S = 0.0187$ \$/kWh at the normal operation. It is noted that the unit cost of heat C_H can be obtained from Eq. (60) directly with known values of C_W, COP (β) and the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of input (electricity). **Table 9** confirms that cost-balance balance is satisfied for all components and the overall system.

Rewriting Eq. (59), we have [25]

$$C_H = \frac{C_W}{\beta}\left[1 + \frac{\sum \dot{Z}_k}{C_W \dot{E}^W_{x,comp}}\right] \tag{60}$$

where β is the COP of the air conditioning units. Equation (60) provides the unit cost of cooling capacity as 0.0344 \$/kWh with a unit cost of electricity of 0.120 \$/kWh, β of 3.77, and a value of 0.081 for the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of consumed electricity.

6. Conclusions

Explicit equations to obtain the unit cost of products from gas-turbine power plant and organic Rankin cycle plant operating by heat source as fuel and the unit cost heat for refrigeration system using the modified-productive structure analysis (MOPSA) method were obtained. MOPSA method provides two basic equations for exergy-costing method: one is a general exergy-balance equation and the other is cost-balance equation, which can be applicable to any components in power plant or refrigeration system. Exergy-balance equations can be obtained for each component and junction. The cost-balance equation corresponding to the exergy-balance equation can be obtained by assigning a unit cost to the principal product of each component. The overall exergy-costing equation to estimate the unit cost of product from the power plant and refrigeration system is obtained by summing up all the cost-balance equations for each component, junctions, and boundary of the system. However, one should solve the cost-balance equations for the components, junctions, and system boundary simultaneously to obtain the lost cost flow rate due to the entropy generation in each component. It should be noted that the lost work rate due to the entropy generation plays as "product" in the exergy-balance of the component, while the lost cost flow rate plays as "consumed resources" in the cost-balance equation. This concept is very important in the research area of thermoeconomic diagnosis [18, 26–28].

Nomenclature

C	unit cost of exergy ($/kJ)
C_i	initial investment cost ($)
C_H	unit cost of heat ($/kWh)
C_o	unit cost of fuel ($/kWh)
C_S	unit cost of lost work due to the entropy generation ($/kWh)
C_W	unit cost of electricity ($/kWh)
$\dot{C}$	monetary flow rate ($/h)
COP	coefficient of performance
CRF	capital recovery factor
e_x	exergy per mass
$\dot{E}_x$	exergy flow rate (kW)
h	enthalpy per mass
$\dot{H}$	enthalpy flow rate (kW)
i	interest rate
$\dot{I}$	irreversibility rate (kW)
$\dot{m}$	mass flow rate
PW	amortization cost
PWF(i,n)	present worth factor
$\dot{Q}_{cv}$	heat transfer rate (kW)
$\dot{S}$	entropy flow rate (kW/K)

Sn	salvage value (KRW)
T_o	ambient temperature (°C)
$\dot{W}_{cv}$	work production rate (kW)
$\dot{Z}_k$	capital cost flow rate of unit k ($/h)

Greek symbols

β	coefficient of performance
δ	operating hours
η_e	exergy efficiency
ϕ_k	maintenance factor of unit k

Subscripts

a	air stream
comp	compressor
con	condenser
env	environment
evap	evaporator
H	heat
k	kth component
r	refrigerant stream
ref.	reference condition
room	room
s	entropy
sl	suction line
W	work or electricity

Superscripts

a	air stream

CHE	chemical exergy
H	heat
P	mechanical exergy
r	refrigerant stream
T	thermal exergy
W	work or electricity

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The main scope of this study is to emphasize exergy efficiency in all fields of industry. The chapters collected in the book are contributed by invited researchers with a long-standing experience in different research areas. I hope that the material presented here is understandable to a wide audience, not only energy engineers but also scientists from various disciplines. The book contains seven chapters in three sections: (1) "General Information about Exergy," (2) "Exergy Applications," and (3) "Thermoeconomic Analysis." This book provides detailed and up-to-date evaluations in different areas written by academics with experience in their fields. It is anticipated that this book will make a scientific contribution to exergy workers, researchers, academics, PhD students, and other scientists in both the present and the future.

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