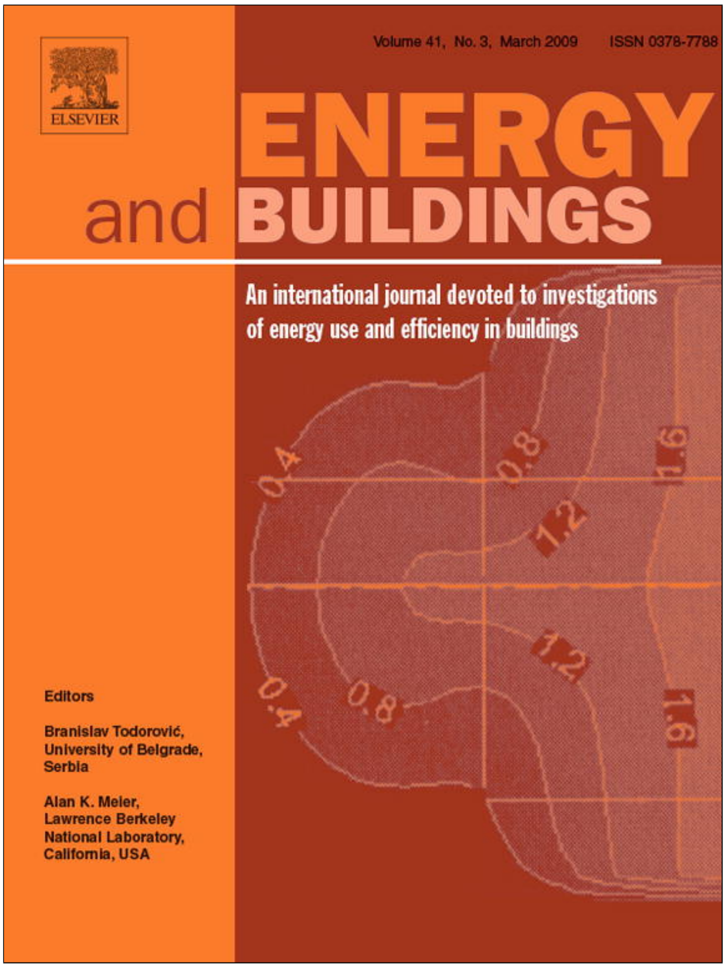
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Review

Exergy analysis of renewable energy-based climatisation systems for buildings: A critical view

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ABSTRACT

Exergy is naturally related to the concept of quality of energy. Therefore, exergy analysis has been widely applied in parallel with energy analysis in order to find the most rational use of energy. Within the built environment a wide margin for exergy saving may be found. Actually, buildings require mostly low quality energy for thermal uses at low temperatures and nowadays their energy demand is mainly satisfied with high quality sources. Exergy analysis of renewable energy-based climatisation systems may be considered an emerging field, where different and often contrasting approaches are followed. Then, in this paper a comprehensive and critical view on the most recent studies on this topic is presented. Special attention is paid to the methodological aspects specifically related to climatisation systems and renewables, and to the comparison of the results. Main renewable energy-based heating and cooling systems are considered in detail. Finally, conclusions regarding the state of the art and possible trends on this field are derived, with the aim to highlight future research issues and promote further developments of this method. Furthermore, conclusions regarding the usability of the exergy method as a tool to promote a more efficient use of available energy sources are also derived.

Contents

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Nomenclature

A	area (m^2)
AHU	air-handling unit
ASHP	air source heat pump
CEC	cumulative exergy consumption
COP	coefficient of performance
c	specific heat (J/kg K)
C	thermal capacity (J/K)
DEC	direct evaporative cooling
DHW	domestic hot water
DIEC	direct–indirect evaporative cooling
EAT	entropy added tax
ECEC	ecological cumulative exergy consumption
EEA	extended exergy analysis
EER	exergy efficiency ratio
ELCA	exergetic life cycle analysis
e	specific total energy (J/kg)
E	total energy (J)
EPC	exergetic performance coefficient
ex	specific exergy (J/kg)
Ex	total exergy (J)
$\dot{E}x$	exergy rate (W)
FPC	flat plate collector
Fq	quality factor
G	solar irradiance (W/m^2)
GHEX	ground heat exchanger
GSASHP	ground source air source heat pump
GSHP	ground source heat pump
h	specific enthalpy (J/kg)
HGHE	horizontal ground heat exchanger
HVAC	heating ventilation air-conditioning
I	global irradiation (J/m^2)
$\dot{I}$	rate of irreversibility, rate of exergy consumption (W)
ICEC	industrial cumulative exergy consumption
IEC	indirect evaporative cooling
LCA	life cycle assessment-based
LNG	liquefied natural gas
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
p	pressure (Pa)
PV	photovoltaic
R	specific gas constant (J/kg K)
REC	regenerative evaporative cooling
s	specific entropy (J/kg K)
S	entropy (J/K)
SAASHP	solar-assisted air source heat pump
SAGSHP	solar-assisted ground source heat pump
SAHP	solar-assisted heat pump
SDHW	solar domestic hot water
SH	space heating
SPEC	specific primary energy consumption
T	temperature (K)
t	time (s)
Q	heat (J)
$\dot{Q}$	heat transfer rate (W)
U	heat transfer coefficient (W/m^2 K)
W	work (or power) (J)
$\dot{W}$	rate of work (or power) (W)
x	solar thermal collector parameter (°C m^2/W)
y	mass fraction

Greek letters

ε	emissivity of the surface; exergy expenditure figure
η	energy (first law) efficiency
φ	relative humidity
μ	chemical potential (J/kg)
ν	specific volume (m^3/kg)
θ	temperature (°C)
σ	Stephan–Boltzmann constant (W/(m^2 K^4)); sensitivity exergy analysis
ω	humidity ratio (g/kg)
ψ	exergetic efficiency (%)
Δ	increment

Indices

0	reference or ambient state
*	restricted reference state
a	air
abs	absorber
AHU	air-handling unit
aux	auxiliary
boil	boiler
cw	condensate water
Carnot	Carnot
cc	cooling coil
ch	chemical
chill	chiller
circ	circulate
coll	collector
comb	combined
concr	concret
cond	condenser
cool	cooling

cool-t	cooling-tower
des	Desired
distr	distribution
dyn	dynamic
el	electrical
emiss	emission
env	envelope
evap	evaporation/evaporator
f	floor
g	ground
gen	generation/generator
heat	heating
hc	heating coil
i	Component, species
in	input
loss	losses
m	mass flow
mech	mechanical
min	minimum
out	output
outdoor	outdoor
ove	overall
p	constant pressure
pl	plate
phys	physical
prim	primary
PV	photovoltaic
rad	radiation
rat	rational
refrig	refrigeration
rej	rejected
ret	return air
room	room
sat	saturation
sc	solar collector
simple	simple
single	single
steady	steady
sol	solar
source	source
sun	sun
sup	supply conditions
tech	technical
th	thermal
thdyn	thermodynamic
tot	total
v	vapour
w	water
W	power
zone	zone

1. Introduction

The exergy of a system in a given environment is the maximum theoretical work that might be extracted from it. Consequently, exergy is a measure of the potential of a given energy flow to be transformed into high quality energy. Exergy analysis has been applied since the early 1970s with the aim of finding the most rational use of energy, which means at the same time reducing fossil fuels consumption, applying energy efficiency and matching the quality levels of the energy supply and demand. After a period during which most scientific efforts were concentrated on energy analysis and CO_2 emission balances, in the last years exergy has been rediscovered and evenly applied to new scenarios for energy supply both at building and community levels (see for instance the following either ended or ongoing international research projects: IEA ECBCS Annex 37 [1] and Annex 49 [2], COST Action C24 COSTeXergy [3]).

While a wide literature exists on exergy analysis of power plants [4–13], the application of the exergy approach to the built environment may be considered at an earlier stage. Nevertheless, most of the energy consumption in the building stock is related to near-environmental temperature thermal uses, namely space heating and cooling and hot water production. These low quality energy demands are mainly satisfied with high quality or high exergy sources (e.g. fossil fuels). Therefore, beside the well-known issue of energy saving, a wide margin for exergy saving exists within the built environment. Nevertheless, since climatisation systems operate closer to the reference environment compared with power plants, a question arises about the suitability of the exergy metrics, mainly developed for plants analysis, for climatisation systems.

On the other side, renewable energy sources may give an essential contribution to the CO_2 emissions reduction. Although some of them may be considered "purely renewable" (e.g. solar energy), some others are not endlessly available (e.g. biomass), depending on how fast they are consumed in relation with their regeneration time. Therefore exergy analysis may be fruitfully applied to renewable energy-based systems in order to identify the optimal and most efficient use of the available renewable sources.

In this paper a critical view on the most recent studies on exergy analysis of renewable energy-based climatisation systems is carried out. Special attention is dedicated to the methodologies, regarded in Section 2, dealing with the choice of the reference state, the levels and the boundaries of the analysis, the steady state or dynamic approach and the performance indicators. The aim is to highlight specificities of the exergy approach applied to both climatisation systems and renewable sources and to stimulate further debate on these topics. Next, the applications of the exergy analysis to renewable energy-based heating and cooling systems are considered more in detail, in Sections 3 and 4, respectively. The considered systems are solar thermal collectors, PV/hybrid systems, ground source and solar-assisted heat pumps, biomass boilers, evaporative cooling systems, absorption systems and dessiccant systems may be solar-driven. The different studies are compared and discussed, regarding both the methodologies adopted and the results achieved. Finally, in Section 5 some conclusions are derived, with the purpose of highlighting present knowledge, state of the art and suggesting future developments for the application of the exergy method to climatisation systems in buildings.

2. Fundamentals: methodologies for exergy analysis

Exergy is defined as the maximum theoretical work obtainable from the interaction of a system with its environment until the equilibrium state between both is reached [14] and can also be seen as the departure state of one system from that of the reference environment [15]. Therefore, exergy is a thermodynamic property dependent on the state of the system under analysis and its surrounding environment, so-called "reference environment". The environment is regarded as a part of the system surroundings, large in extent so that no changes in its intensive properties (pressure (p_0), temperature (T_0), and chemical composition expressed in terms of the chemical potentials $\mu_{i,0}$ of the species present on it) occur as a result of the interaction with the system considered. Furthermore, intensive properties of the environment have to be uniform and the environment is regarded as free of irreversibilities, i.e. only internally reversible processes take place on it [14].

Considering the physical and chemical exergy, the exergy level of a system is determined by the temperature, T, pressure, p, of the system and chemical potentials of the substances comprising the system μ_i, as referred to the pressure, p_0, temperature, T_0, and chemical potentials $\mu_{i,0}$ of the species in the reference environment [16].

Subsequently, for the estimation and calculation of the exergy flows, the choice of extensive properties defining the reference environment is of capital importance.

In Table 1 a list of the equations for estimating the exergy related to the specific energy processes relevant in this paper is shown. A detailed derivation of the formulas may be found in Ref. [17].

2.1. Reference state

As already stated above, the choice and definition of the reference environment is of capital importance for exergy analysis. However, the sensitivity of the results from exergy analysis to different choices of the reference state might vary with the operative conditions of the energy system analysed.

Rosen and Dincer [19] carry out a sensitivity analysis on the results from energy and exergy analyses for different definitions of the dead state, i.e. reference, environment. In their study the reference environment is defined mainly on the base of its pressure and temperature. Superficial regards on the chemical composition and chemical potential of a power plant as case study for the sensitivity analysis are also presented.

The authors define the sensitivity of the exergy, σ, as the ratio of the change in exergy content of the system when varying a magnitude Y in the definition of the reference environment and its initial exergy content. A general mathematical formulation for the sensitivity is given in the following equation:

$$\sigma = \frac{Ex(Y_0 + \Delta Y_0) - Ex(Y_0)}{Ex(Y_0)} \tag{14}$$

The sensitivity for a thermal energy flow, i.e. assuming that the system has the same pressure and chemical composition as the reference environment, $p = p_0$ and $\mu_i = \mu_{i,0}$, and, thus, defining the system and its environment in terms of temperature, is shown in the following equation:

$$\sigma = \frac{\Delta T_0}{T - T_0} \tag{15}$$

Accordingly, when the state of the system is significantly different from that of the chosen dead-state, exergy flows are not very sensitive to the definition of the reference environment. This is the case, for instance, in the energy and exergy analysis of power plants. In turn, when the properties of the system are close to those of the reference environment, results from exergy analysis undergo strong variations depending on the definition of the reference environment chosen. This is the case of exergy analysis of space heating and cooling in buildings.

Subsequently, for this application some authors propose a reference environment defined as the variable outdoor environment surrounding the building [20–23]. This definition of the reference environment requires the use of dynamic energy and exergy analysis, therefore representing a more detailed and complex analysis than mere steady-state assessment.

However, the majority of the papers reviewed in this article follow a steady-state approach. The reference environment can then be chosen upon several criteria: seasonal mean values, annual mean values, design conditions, etc. Due to the great sensitivity of exergy analysis for the particular case of space heating and cooling in buildings, each of these choices would significantly influence the outcoming results from exergy analysis and greatly difficults the comparison among results from different analyses. Furthermore, depending on the chosen definition for the reference environment significant mismatching between the steady-state and dynamic assessment of the exergy values can be found. This is addressed in Section 2.3. To the best of the authors' knowledge there is no common agreement for a proper definition of the reference environment for steady state analysis. Therefore, future work in this direction is required.

Climatisation of buildings occurs at atmospheric pressure, so that no mechanical exergy demand is present. However, indoor environment might differ from outdoor reference environment in its temperature level and humidity content. Subsequently, a chemical potential exists in indoor air as compared to outdoor dead-state conditions.

Sakulpipatsin [24] evaluated the influence of including the air humidity in the definition of both the building system and its reference environment on the exergy flows through the building envelope. Two different climatic conditions were investigated: Bangkok (Thailand) as hot and humid climate and De Bilt (The

Table 1
Equations for estimating the exergy related to specific energy processes [17,18].

Total exergy		(1)$ex = ex_{phys} + ex_{ch}$
Physical exergy	General	(2)$ex_{phys} = ex_{th} + ex_{mech}$ (3)$ex_{phys} = (h - h_0) - T_0(s - s_0)$
	Ideal gas	(4)$ex_{phys} = c_p(T - T_0) - T_0\left(c_p \ln\frac{T}{T_0} - R\ln\frac{p}{p_0}\right)$
	Solid/liquid	(5)$ex_{phys} = c\left[(T - T_0) - T_0 \ln\left(\frac{T}{T_0}\right)\right] - \nu(p - p_0)$
	Humid air	(6)$\mathrm{ex_{phys}} = (c_{p,a} + \omega c_{p,v})\left[(T - T_0) - T_0 \ln\frac{T}{T_0}\right] + (1+\omega)R_a T_0 \ln\frac{p}{p_0}$
Mechanical exergy	Ideal gas	(7)$ex_{mech} = RT_0 \ln\left(\frac{p}{p_0}\right)$
Thermal exergy	General	(8)$Ex_{th} = Q\left(1 - \frac{T_0}{T}\right)$
	Contained by a mass of room air	(9)$Ex_{th,room} = c_a m_{room}\left\{(T_{room} - T_0) - T_0 \ln\frac{T_{room}}{T_0}\right\}$
Radiant exergy		(10)$Ex_{rad} = A\varepsilon\sigma\left\{(T^4 - T_0^4) - \frac{4}{3}T_0(T^3 - T_0^3)\right\}$
Chemical exergy	Ideal gas	(11)$ex_{ch} = \sum_i(\mu_i^* - \mu_{oi})y_i$
	Liquid water	(12)$ex_{ch} \cong (p - p_{sat})\nu - RT_0 \ln\phi_0$
	Humid air	(13)$ex_{ch} = R_a T_0\left[(1+\omega)\ln\frac{1+\omega_0}{1+\omega} + \omega\ln\frac{\omega}{\omega_0}\right]$

Netherlands) as cold and dry climate. In both cases regarding dynamic variations in the indoor and outdoor air humidity leads to the most accurate estimation of the exergy flows. In turn, obviating ambient air humidity (i.e. regarded as zero or equal to indoor air humidity), leads to underestimations in the exergy flows arising differences of up to 86% in the total annual exergy flows for the hot and humid climate and around 3% in the cold dry climatic conditions.

In hot and humid climatic conditions buildings are usually equipped with cooling systems managing the temperature and indoor air humidity to be within comfort levels. Therefore, indoor and outdoor air humidity might differ significantly from each other. In this case, it is of great importance to include the humidity in the definition of the system and its environment. In turn, in cold drier climates where the differences between indoor and outdoor air humidity is significantly lower, humidity can be obviated from the definition of both the system and its environment without significant losses in the accuracy of the exergy flows.

In Ref. [25] the reference environment is defined as saturated outdoor air. The choice is based on the will to disregard the exergy content of the condensate water resulting from air dehumidification processes. In turn, Wepfer et al. [26] analyse similar cooling systems using outdoor air with humidity content of 0.104 g/kg$_{air}$ as reference environment. The choice of different reference environments greatly influences the results for chemical exergy analysis of the processes involved, unabling the comparison between the analyses. This highlights the strong need for a common framework for performing exergy analysis.

2.2. Optimisation procedures

In order to improve the efficiency of heat and cold supply in buildings, the whole energy chain for supplying these demands needs to be assessed. In Fig. 1 a simplified schema of such an energy chain for space heating applications is shown. Similar approaches, assessing not only the energy demand of the building or the energy performance of a certain building system component, can be found in new energy regulations [27–30]. All supply steps in the energy chain are directly related to each other and its performance often depends on one another. Following this approach, an overall optimisation of the building and its energy systems can be accomplished, avoiding optimisation of single components which might have a negative influence on other steps.

Similarly, exergy analysis should at first be based on this holistic framework. Optimization of the single components is desirable and required, but the influence of optimising one component on the performance of the following and previous ones should always be regarded. By these means, an optimisation of the integral system is pursued, avoiding optimising single components which might decrease the performance of the system as a whole (see Section 4.2).

For assessing the exergy performance of the complete energy chain, a simplified input/output approach is usually followed, similar as that developed for energy analysis. This whole chain exergy analysis is implemented in an Excel-based pre-design tool developed by Schmidt [31] in the framework of the IEA ECBCS Annex 37 [1] programme.

Following this approach, exergy efficiencies can be derived for characterising the performance of the conversion steps on a component level or on a system level. The first will be called "single exergy efficiencies" and the latest "overall exergy efficiency". For a detailed discussion on the definition of the exergy efficiency see Section 2.3.

A similar methodology, including the whole energy chain for building energy use, has been recently included in a new building regulation on the Geneva canton (Switzerland) [32]. The modular approach presented in Ref. [32], divides the energy chain into the following modules: room convector, building plant, district heating or cooling plant and external power plant. Thus, some of the modules shown in Fig. 1 are combined into only one system, but the underlying principle of analysis is the same.

A more detailed analysis framework is defined in Ref. [33]. The authors perform a thorough analysis of the exergy losses in each energy process, dividing them into avoidable and unavoidable irreversibilities. By this means, the method allows identifying which is the real potential for the optimisation of each component in an energy system. Unavoidable irreversibilities are defined as those happening when all components operate under their ideal thermodynamic efficiencies (e.g. *Carnot* efficiencies for thermal processes). Furthermore, their methodology also distinguishes between endogenous and exogenous irreversibilities for each component. Endogenous irreversibilities are defined as the part of exergy destruction occurring due to irreversibilities in that particular component when all other energy processes in the system are regarded as ideal. Exogenous irreversibilities, in turn, arise in the particular component due to a not ideal performance of the rest of energy processes present in the system. This method, although significantly more complicated than simple exergy analysis, allows identifying processes which are inherently irreversible, i.e. have mainly unavoidable irreversibilities, such as evaporative cooling, from those with avoidable irreversibilities where great optimisation potential still exists. In turn, even if an inherently irreversible process turns out to be a major contributor to the total irreversibility of a system, its improvement is limited

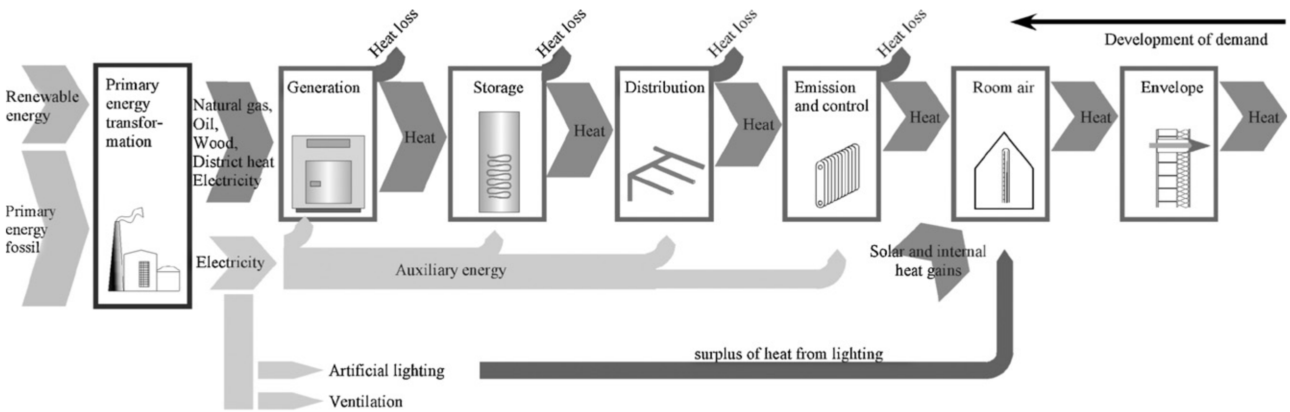

Fig. 1. Energy supply chain for space heating in buildings, including from primary energy transformation into final energy, all intermediate steps until the supply of the building demand [31].

due to the unavoidable nature of its irreversibilities. Morosuk and Tsatsaronis [33] applied this interesting methodology for exergy analysis to the specific case of refrigeration machines. However, their method might be applied to any other energy system, or to the complete energy chain for energy demand in buildings.

2.3. Exergy efficiencies

A good base for comparing heating and cooling systems for buildings is their exergy efficiency. Exergy efficiencies of systems with the same form of output, help identifying those with the less exergy destruction [34], i.e. those which are closer to the ideal maximum output that can be gained from the processes involved.

As any other efficiency, exergy efficiencies are defined as the ratio between the obtained output and the input required to produce it. However, at least two types of exergy efficiencies can be identified and differentiated: "simple" or "universal" and "rational" or "functional" [34,35]. Their mathematical expressions are shown in the following equations:

$$\Psi_{simple} = \frac{Ex_{out}}{Ex_{in}} \tag{16}$$

$$\Psi_{rat} = \frac{Ex_{des,out}}{Ex_{in}} \tag{17}$$

Although the simple exergy efficiency is an unambiguous definition for the exergy performance of a system, it works better when all the components of the incoming exergy flow are transformed into some kind of useful output [34]. In most of the building systems analysed in this paper this is not the case, since some part of the exergy input is fed back again to the energy system and does not constitute a useful output strictly speaking, e.g. in a hydronic heat or cold emission system in a building, outlet water flows back via return pipes into the heat/cold generation system. The rational exergy efficiency, in turn, accounts for this difference between "desired output" and any other kind of outflow from the system. Therefore, it is a much more accurate definition of the performance of a system. It is, in consequence, a term that can be better used without taking to misleading conclusions. Starting from the idea that many existing exergy efficiency definitions were developed for use with larger temperature differences and further from the environmental temperatures, Boelman and Sakulpipatsin [36] presented a critical analysis of exergy efficiency definitions to be potentially used in the field of exergy analysis of building services. Taking into account a simple heat exchanger operating at near-environmental temperatures, they study the sensitivity of both the simple and the rational exergy efficiency to outdoor temperature, fluid inlet temperatures and thermal effectiveness of the heat exchanger, finding that the rational exergy efficiency is more sensitive to the above mentioned parameters.

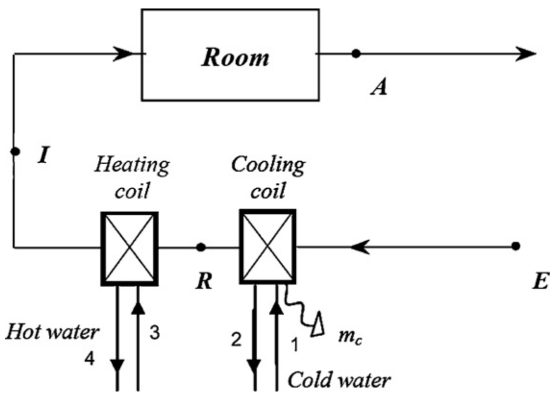

Fig. 2. Schema of an air-handling unit (AHU) for cooling, dehumidification and re-heating of supply air [37], modified.

Table 2
Temperature, water content, enthalpy and exergy regarded for each of the working conditions in the schema of the *AHU* in Fig. 2 [37].

	T (°C)	ω (g/kg)	h (kJ/kg)	ex (kJ/kg)
A	26	10.5	55.76	0.61
E	35	21.4	89.90	0.00
I	18.5	9.5	45.28	1.05
R	13.3	9.5	37.29	1.40
1	8	–	–	77.83
2	13	–	–	76.02
3	60	–	–	76.60
4	50	–	–	74.05
C	–	–	–	75.92

To further illustrate the difference between both efficiency definitions, an example based on operational data for an air-handling unit (AHU) from [37] is presented. The example deals with an air-handling unit for cooling, dehumidification and re-heating of supply air (see Fig. 2). Properties from state E (see Table 2) are regarded as reference environment. Water inlet temperatures of 8 and 60 °C and temperature drops of 5 and 10 °C are regarded for the cooling and heating coils, respectively. An air-flow of 1 kg/s is supplied by the unit. Heating and cooling coils are considered adiabatic heat exchangers and resulting mass flows for the heating and cooling coils are 2.51 and 0.126 kg/s, respectively. Eqs. (18) and (19) show the analytical expression for the simple and rational efficiencies applied to this particular example:

$$\Psi_{simple} = \frac{Ex_{out}}{Ex_{in}} = \frac{\dot{m}_a ex_I + \dot{m}_{cw} ex_{cw} + \dot{m}_{cc} ex_2 + \dot{m}_{hc} ex_4}{\dot{m}_a ex_E + \dot{m}_{cc} ex_1 + \dot{m}_{hc} ex_3} = 0.985 \tag{18}$$

$$\Psi_{rat} = \frac{Ex_{des,out}}{Ex_{in}} = \frac{\dot{m}_a (ex_I - ex_E)}{\dot{m}_{cc}(ex_1 - ex_2) + \dot{m}_{hc}(ex_3 - ex_4)} = 0.216 \tag{19}$$

In the example the great difference between so-called simple and rational exergy efficiencies is clearly shown. If all exergy input would be used to provide a given output they would become equivalent. However, as long as some processes are not strictly desired output from the system, mismatching between them arises. The simple efficiency gives a figure on how close are the processes involved to the ideal performance. In turn, rational efficiency shows how much potential is getting lost for providing a specific output. Exergy losses regarded in the rational efficiency are due to both irreversible (not ideal) processes present and to unused output exergy flows, e.g. undesired losses as water condensate, which could be re-used in the process thus lowering exergy losses from dehumidification process, leaves the system.

Since the kind of exergy efficiency definition is a key issue, the choices made by the several authors whose papers are reviewed in the following sections will be highlighted. It will be then evident that there is no general consensus on the definition to be used in the case of climatisation systems.

Furthermore, depending on whether the exergy efficiency is referred to a single component or process of a whole energy system, or whether it refers to all processes and components integrating the system, so-called "single" and "overall" exergy efficiencies can be defined [37–39]. "Overall efficiencies" are those regarding the whole air handling processes in the *AHU* in example above. An example of single and overall efficiencies for the room air subsystem and complete energy chain in Fig. 1 is given in Eqs. (20) and (21). Overall efficiencies are derived from an input/output approach for the analysis of a given energy system and are derived from the product of the single efficiencies of the single processes or components encomprised in the energy system analysed:

$$\Psi_{sngle,room-a} = \frac{Ex_{in,env}}{Ex_{in,room-a}} \quad (20)$$

$$\Psi_{ove} = \frac{Ex_{out,env}}{Ex_{in,\,prim}} \quad (21)$$

2.4. Exergy indicators for the built environment

In 2001 the Swiss canton of Geneva included the exergy efficiency as a new parameter for characterising the energy performance of buildings [32]. Favrat et al. [32] present a discussion and description of some of the exergy efficiencies and exergy-based parameters included in the new regulation. The rational exergy efficiency (see Section 2.3) is the parameter chosen. Heating, cooling and lighting in buildings are the main services related to the new energy code. In this approach, exergy efficiency is only one of several parameters for characterizing the buildings performance, and is not linked to any consideration on the renewability of the energy sources. The overall exergy efficiency, similar as in Eq. (21), is the parameter chosen for characterising the performance of the building and its energy supply chain. In addition, single exergy efficiencies for each of the four subsystems defined in the modular approach followed (room convector, building plant, district heating or cooling plant and power plant) are also given.

Schmidt et al. [40] developed a benchmarking proposal for characterising the performance of energy building systems. The authors define the so-called "exergy expenditure figure" for characterising the exergy supply in buildings. The proposal from Ref. [40] is based on the combined limitation of the primary energy (fossil) consumption of the building and a new parameter called "exergy expenditure figure".

In Eq. (22) the exergy expenditure figure is defined for a general component i of an energy system. This parameter is calculated as the ratio of the exergy input required to supply a given energy demand (effort) and the provided energy demand (use). Therefore, it represents a sort of quality factor (exergy to energy ratio) of the energy processes occurring in the given component.

Energy and exergy losses happening in the component are implicitly taken into account by the ratio of provided output to required input. In consequence, if the energy losses in the component are high, i.e. low energy efficiency, the exergy expenditure figure might reach values higher than 1 (see Eq. (22)).

This parameter needs to be compared to the exergy to energy ratio of the energy demand to be provided, i.e. to the quality factor of the energy demand. Values close to the exergy to energy ratio of the energy demand indicate a good matching between quality levels (i.e. exergy) of the energy supplied and demanded. In turn, values diverging from the exergy to energy ratio of the demand indicate bad matching and, in consequence, lead to conclude that other energy sources shall be used for providing that specific use and/or energy losses need to be reduced.

For the particular application of space heating and cooling of buildings, the quality factors (or exergy to energy ratio) of the energy demanded are very low. In Ref. [40], for space heating applications and reference and indoor air temperatures of 0 and 20 °C, respectively, this quality factor of energy demand is found to be 7%. Therefore, for space heating (and cooling) of buildings, the closer the exergy expenditure figure for a given system to that 7%, the better the system exergy performance is. Subsequently, in space heating and cooling applications, lower exergy expenditure figures indicate more optimised energy supply systems.

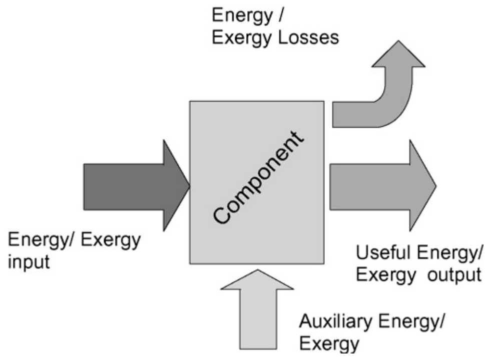

Fig. 3. Graphical representation of the exergy flows included in the exergy expenditure figure for a general component of an energy system [40].

In Fig. 3 the exergy flows regarded for the general definition of the exergy expenditure figure for a component i are shown graphically:

$$\varepsilon_i = \frac{Effort}{Use} = \frac{Ex_{in}}{En_{out,i}} = \frac{F_{q,in,i}}{\eta_i} \quad (22)$$

In order to show the different behaviour and order of magnitude of the two main exergy indicators (exergy efficiency and exergy expenditure figures) defined in Refs. [32] and [40] their values for four different examples of building systems are listed in Table 3.

Values for the exergy expenditure figure have been calculated by the authors of the present study, neglecting the auxiliary energy required for the operation of the systems. Energy efficiencies assumed are shown in Table 3. Eqs. (23) and (24) are used for the calculation of the exergy expenditure figures for the boiler and emission systems, respectively. Values for the single exergy efficiencies are directly taken form [32]. The reference temperature is regarded as 0 °C in all cases:

$$\varepsilon_{boil} = \frac{Ex_{in,boil}}{En_{out,boil}} = \frac{F_{q,\,fuel}}{\eta_{boil}} \quad (23)$$

$$\varepsilon_{emiss} = \frac{Ex_{in,emiss}}{En_{out,emiss}} = \frac{1}{\eta_{emiss}(T_{\sup} - T_{ret})}\left[(T_{\sup} - T_{ret}) - T_0 \ln\frac{T_{\sup}}{T_{ret}}\right] \quad (24)$$

As already stated above, for space heating and cooling applications, the lower the exergy expenditure figure, ε (i.e. the closer it is to the quality factor of energy demanded) the better the system performs. On the contrary, the higher the exergy efficiency of the system Ψ_{single}, the better the system performance. From the results in Table 3 it can be seen that both parameters would lead to similar conclusions regarding the performance of the emission systems. Despite having the same energy efficiency as air convectors, floor-heating systems allow reducing the exergy losses on the energy transfer to the room air, thus having a better exergy performance. This is due to the lower temperature levels of the energy supplied (lower inlet and return temperatures) as compared to the temperature level of the energy demanded (room air, e.g. 20 °C).

However, from the results in Table 3 it can also be observed that exergy efficiency is a more sensitive parameter than the exergy expenditure figure for evaluating the performance of a boiler coupled with different energy supply systems (e.g. emission systems with different inlet and return temperature levels). In the exergy efficiency both the quality levels of the energy supplied and demanded are regarded, i.e. lowering down the supply tempera-

Table 3
Numerical values for the single exergy efficiency as defined in Ref. [32] and exergy expenditure figure as defined in Ref. [40] for four examples of building systems.

Parameter	Reference	Condensing boiler (65/55 °C)	Condensing boiler (45/35 °C)	Convectors (65/55 °C)	Floor heating (45/35 °C)
Ψ_{single}	[32]	0.16	0.12	0.38	0.53
ε	[40]	1.03	1.03	0.19	0.13
η		0.92	0.92	0.95	0.95

Values for the exergy efficiencies are directly taken from [32]; exergy expenditure figures have been calculated by the authors.

ture levels leads to a reduction of the exergy supplied. In turn, since the energy efficiency of the boiler is regarded as constant, and the same energy source is regarded in both cases, e.g. LNG, the required exergy input is the same in both cases. On the contrary, for the exergy expenditure figure only the quality level of the energy supplied (effort) is regarded, whereas the output (use) is regarded in energy terms. Therefore, as long as a certain energy source with its corresponding quality level is used with the same energy efficiency, i.e. energy losses are the same, the exergy expenditure figure would be the same. Thus, the exergy expenditure figure assesses only the quality level used for providing a certain service and the energy losses occurring on one single component and, as long as the operation of the following components in the energy system does not influence the energy performance of the analysed component, the parameter will not vary.

Exergonomics is an assessment framework which combines exergy and cost analysis, based on the idea that exergy is the only rational basis for assigning economic costs to the processes (inputs or outputs) of a certain system [41]. Following this combined exergy and economic analysis, another normative proposal to include exergy in the national energy regulations is described in Ref. [42]. The authors give a suggestion to implement in India two different environmental taxes based on exergy analysis (EA) and life cycle assessment (LCA), respectively. The entropy added tax (EAT) follows the approach from Hirs [43] and it is thought to tax exergy losses or entropy produced when supplying a given demand, so that it might represent an incentive for the manufacturers to introduce proper modifications in the design of components to minimise their exergy loss.

2.5. Steady-state and dynamic exergy analysis

Accurate estimations of the energy demands and flows in buildings are necessarily dynamic or at least quasi-steady state. The last approach is currently used in building regulations in several European countries (e.g. [27]). Exergy evaluations are usually carried out following a steady-state approach. Yet, as discussed in Section 2.1, exergy flows are very sensitive to variations of the chosen reference conditions when the variables of the system and its environment do not differ very much from each other, which is the case of space heating and cooling in the built environment. Sakulpipatsin [24] evaluated inaccuracies for different statistic values of the variables defining the reference environment in different climates. For a cold climate in the Netherlands (De Bilt) inaccuracies of using the mode on an annual basis for indoor and outdoor temperatures yields the smallest error, yet leading to an overestimation of the exergy flows of almost 9%, as compared to dynamic annual simulations. For a hot and humid climate in Thailand (Bangkok), using the average and median to estimate annual indoor and outdoor air temperatures yields the smallest error as compared to dynamic simulations, yet leading to an underestimation of 93% on the exergy flows. For a temperate sea climate in Portugal (Lisbon), average and median indoor and outdoor air temperatures lead to underestimations of 44% on the exergy flows. Therefore, the statistical values leading to minimum differences versus dynamic analysis are different depending on the climatic conditions and a common magnitude leading to minimum mismatching between steady-state and dynamic analysis could not be identified.

Yet in this study, dynamic simulations include the dynamic variation of indoor and outdoor air humidity in the estimation of the exergy flows. In turn, the contribution of air humidity is obviated in annual steady-state analysis, i.e. indoor and outdoor air are considered as dry. Therefore, it is expected that a significant contribution to the mismatching between dynamic and steady-state results is due to disregarding humidity. It would be interesting to investigate further the relevance of each effect (dynamic, humidity) separately.

Angelotti and Caputo [22] evaluate the difference between steady state and dynamic analysis for heating and cooling systems in two representative Italian climates, namely Milano and Palermo. Here, only thermal exergy flows are regarded, i.e. the reference and indoor environment are only defined based on their temperature levels and no considerations on air humidity are included. Two different building systems are chosen for comparison: a reversible air-source heat pump (i) and a condensing boiler coupled with direct ground cooling (ii). Steady state exergy analysis is performed using design conditions (i.e. design outdoor temperature) and mean monthly outdoor temperatures for the coldest (January) and warmest (July) months. *COP* of the building systems for steady state analysis are taken accordingly to the outdoor temperature regarded in each case. Dynamic and steady state exergy efficiencies are compared. Eq. (25) shows the expression for the steady state exergy efficiency for the heat pump. Dynamic efficiencies are obtained by averaging instantaneous exergy efficiencies calculated on an hourly basis from dynamic analysis, as shown in Eq. (26) for the same case:

$$\psi_{steady} = COP\left(1 - \frac{T_0}{T_{room}}\right) \tag{25}$$

$$\psi_{dyn} = \left\langle COP\left(1 - \frac{T_0}{T_{room}}\right)\right\rangle = \frac{1}{N}\sum_{i=1}^{N} COP_i\left(1 - \frac{T_{0,i}}{T_{room,i}}\right) \tag{26}$$

Main results for the case of Milano are shown in Table 4. Steady state exergy efficiencies for the heating case using average outdoor temperatures are very close to those resulting from dynamic exergy analysis. However, for the cooling case mean monthly outdoor temperature is below indoor design temperature, and no exergy analysis could be performed. In turn, using design values for the estimation of the exergy efficiency leads to great mismatching as compared to dynamic analysis: differences of up to 42% are found. The authors remark that the difference is larger for cooling rather than heating systems and for Palermo rather than Milano, i.e. the more the *Carnot* factor in Eqs. (25) and (26) is sensitive to outdoor temperature variations.

It is remarkable that despite higher *COP* for the heat pump is achieved in summer (3.40), exergy efficiencies are significantly lower for the cooling case (around 0.05 and 0.15 for the cooling and heating cases, respectively). This rises from the fact that required indoor temperature under cooling conditions is very close to outdoor air temperature and in consequence, exergy demand for

Table 4
Comparison of exergy efficiencies for a reversible air source heat pump (i) and a condensed boiler coupled with direct ground cooling (ii) for January and July (in %) in Milano [22].

System	January	July
Steady-state design conditions		
(i)	18.4	5.2
(ii)	7.9	20.1
Steady-state monthly averages		
(i)	15.7	–
(ii)	6.1	–
Dynamic monthly analysis		
(i)	15.7	3.6
(ii)	6.2	11.7

space cooling purposes is extremely low. Therefore, cooling processes will always have intrinsically low exergy efficiency unless they are supplied with environmental heat. Subsequently, there is a strong necessity of reducing cooling loads to the most and supply them whenever possible by passive means.

Beside the fact that Sakulpipatsin [24] considers also the effects of taking into account air humidity in the reference state and Angelotti and Caputo [22] focus only on the dynamic versus steady state issue, a qualitative agreement between their conclusions may be found, in the sense that dynamic exergy analyses turn out to be recommended whenever climatisation systems operate very near-environmental temperature.

A dynamic exergy analysis is provided by Nishikawa and Shukuya [44]. They describe a method of calculating "cool" and "warm" exergies stored by building envelopes and make a case study to examine the combined effects of shading and natural ventilation on making a better use of heat capacity of the walls for passive cooling during the nighttime in summer in Tokyo. A single room model with a concrete floor is chosen. When the floor is at a temperature level higher than outside it is said to have a heating potential or "warm" exergy, if its temperature is lower it is said to have a cooling potential or "cool" exergy. The amount of cool and warm exergies stored by the floor and the variation of their rate of storage are calculated. The following dynamic exergy balance Eq. (27) derived from the physical exergy (Eq. (3)) is used:

$$\begin{aligned} & A_f U_{f,concr}(T_f - T_{concr})\left(1 - \frac{T_0}{T_f}\right) - A_f s_{gen,concr} T_0 \\ & = C_{concr}\frac{dT_{concr}}{dt}\left(1 - \frac{T_0}{T_{concr}}\right) + \frac{A_f U_{concr,0}(T_{concr} - T_0)^2}{T_{concr}} \end{aligned} \qquad (27)$$

The first term on the right-hand side represents the rate of exergy stored, that can either be positive or negative, meaning that exergy is stored or released.

Three different situations are compared: no shading and no natural ventilation, no shading and natural ventilation and shading by tree and natural ventilation at nighttime. The authors find, on a monthly basis, that only with a proper combination of shading, natural ventilation and heat capacity a cool exergy is released, i.e. the heat capacity makes it possible that the temperature of the concrete floor is slightly lower than the outdoor air temperature and the floor surface temperature.

2.6. Review of boundaries for exergy analysis

Exergy analysis is used to detect and quantify the improving potential of energy systems [45], and makes possible finding suitable energy sources for a certain energy use by matching the quality levels of supply and demand [1]. However, if the whole production chain or the contribution of natural ecosystems is obviated in the exergy analysis, its advantage for environmentally conscious decision-making is greatly reduced. To overcome this barrier, several thermodynamic methods have been developed to analyse systems on scales larger than individual equipment or single processes. Cumulative exergy consumption (CEC) considers exergy consumed in industrial processes from natural resources, all the way through the supply chain [45]. Industrial cumulative exergy consumption (ICEC) is based on a similar approach and includes the exergy of the natural resources consumed directly or indirectly in an industrial process [46], but exergy for the production of those natural resources is excluded from the analysis. In turn, ecological cumulative exergy consumption (ECEC) also accounts for the exergy consumption in ecological systems for the production of those natural resources regarding the exergy of solar radiation, tidal energy and geothermal energy as inputs for the ecological systems [46]. Data on the exergy consumption in ecological and natural processes has been compiled by Szargut [45,47], Wall and Gong [48], Chen [49], Odum [50] and Hermann [51].

Moreover, extended exergy analysis (EEA) and the so-called emergy method, include the contribution of labour to the production chain in the analysis framework [52]. Exergetic life cycle analysis (ELCA) regards, in addition to the supply chain, the exergy consumption in the disposal and use of the products [34].

2.7. Review of boundaries for exergy analysis of direct-solar systems

From an energy point of view, the earth is an open system receiving a net energy flux from the sun in the form of high quality solar radiation. Depending on the use done of this incident solar radiation, energy systems can be divided in direct and indirect solar systems. Direct solar systems are those energy systems where a direct conversion of solar radiation is forced artificially to provide a certain service or output. Typical examples of direct solar systems are solar thermal collectors, photovoltaic (PV) systems or solar thermal power plants. However, it can be stated that all energy sources present on the earth are actually derived to a great extent from the solar radiation incident on earth [45,47,53]. This leads to argue that all other energy systems might be called indirect solar systems, for implicitly solar energy is their primal driver. Potential energy in water masses, energy content of biomass and crops or fossil fuels is to a great extent derived from the incident solar radiation.

Direct solar systems show a great potential for being used in the field of building climatisation. However, two very different approaches for their exergy assessment have been found in the literature. The main difference between them consists on whether the conversion of solar radiation into the energy service provided by the direct solar system is regarded or not. Thus, the boundary used for the analysis has a strong influence on the results from exergy analysis.

Most of the papers devoted to exergy analysis of direct solar systems include the conversion process of solar radiation into other energy forms. Eq. (28) shows the expression of the single exergy efficiency for a solar collector (as example for a direct solar system) following this approach, i.e. when the boundary for the system analysis is withdrawn including the conversion of solar radiation into heat. This analysis methodology will be referred to in the following as "technical boundary" [54]:

$$\Psi_{coll} = \frac{Ex_{coll}}{Ex_{sol}} \qquad (28)$$

In Eq. (28) the exergy of incident solar radiation, Ex_{sol}, is included as input into the system regarded. It is calculated as a function of the incident solar radiation, G, onto collector surface, A_{coll}. In Eqs. (29) and (30) two different approaches for assessing the exergy of solar radiation are given. Eq. (29) shows the most simplified approach for evaluating the exergy of solar radiation as derived by Jeter [55], where solar radiation is regarded as a heat flow at the sun temperature. In Eq. (30) another approach developed by Petela [56] where the exergy of solar radiation is regarded as thermal radiation at the sun temperature is shown. Both approaches are commonly used in the literature. In Eqs. (29) and (30) T_{sun} represents the sun temperature, which is typically regarded as 6000 K [21,57,58]:

$$Ex_{sol} = GA_{coll}\left[1 - \frac{T_0}{T_{sun}}\right] \tag{29}$$

$$Ex_{sol} = A_{coll}G\left[1 + \frac{1}{3}\left(\frac{T_0}{T_{sun}}\right)^4 - \frac{4}{3}\left(\frac{T_0}{T_{sun}}\right)\right] \tag{30}$$

For estimating the exergy output from the collector, Eq. (31) is used.

$$Ex_{coll} = \dot{m}_{coll}c\left[(T_{out} - T_{in}) - T_0\ln\left(\frac{T_{out}}{T_{in}}\right)\right] \tag{31}$$

This approach, i.e. the "technical boundary", can be found in a great number of papers in the literature [58–64].

Since the conversion of solar radiation into heat has been taken into account, one of the main conclusions derived by the authors above is that the greatest exergy losses in the system (on the range of 80 to 90% of the total losses) occur in the collector field, i.e. in the direct solar system. These exergy losses, occurring because of the conversion of high quality solar radiation into low temperature heat, are also present in the production of other so-called "primary energy sources", e.g. fossil fuels or ground source heat, but they are usually disregarded in the analysis. In order to reduce the great exergy losses in the collector system which arise following this approach, collector outlet temperature should be maximised.

A different approach found in the literature consists on disregarding the conversion of solar radiation into heat in the collector field. A discussion on the physical consistency and theoretical derivation of this boundary for the exergy analysis of direct solar systems can be found in Ref. [54]. As already stated above, it can be argued that all energy processes on earth are to some extent due to the incident solar radiation. Great exergy losses occur in the production of most so-called "primary energy sources", e.g. fossil fuels or superficial ground heat. However, these exergy losses are typically disregarded in energy analysis of systems making an indirect use of solar energy. Similarly, for physical consistency with the previous approach, exergy losses arising from a direct use of solar radiation, e.g. solar thermal or photovoltaic systems, should also be disregarded.

Thus, this approach will be referred to in the following as "physical boundary". Meir [65] states that even if solar radiation is a high quality source, its use for producing low temperature heat only introduces an intermediate temperature level before this high exergy flow irreversibly dissipates, thus making effective use of that energy flow. Sandnes [57] also takes this approach for analysing control strategies and operational modes for solar combi-systems. Sandnes argues that, as solar radiation is an unlimited resource which is difficult to store, the efficiency of the conversion from solar radiation into low-temperature heat is of limited interest.

Since in the "physical boundary" exergy losses in the collector field are obviated, the optimal operation of the solar thermal system depends mainly on the demand to be provided. Sandnes [57] concludes, therefore, that strategies should be used which lead to minimising the collector outlet temperature while keeping it high enough to supply the load.

However, different approaches to evaluate the exergy performance of solar collectors can be derived depending on where the boundary for exergy analysis is drawn. Torres-Reyes et al. [66] evaluate the exergy input into the solar collector as that of the incident solar radiation at the temperature of the collector absorber plate (Eq. (32)), T_{pl}. Yet, in their approach total solar radiation incident onto the collector is regarded as a heat flow at the plate temperature. Optical losses from the solar collector are not regarded. Following this method, single exergy efficiency for the collector field can be derived as the ratio of the exergy output (Eq. (31)) to exergy input (Eq. (32)). Exergy efficiency would then characterize the thermal conductivity of the absorber plate to the collector fluid. This assessment approach is also found in Refs. [67] and [68]:

$$Ex_{in,coll} = GA_{coll}\left[1 - \frac{T_0}{T_{pl}}\right] \tag{32}$$

Another approach would be to obviate the exergy input into the direct solar system (e.g. solar collector) and regard only the exergy output following Eq. (31). This method might be of interest when the analysis and optimisation of the solar collector as a single component is not of interest, but in turn its efficient integration into an energy system is the main pursued aim. Following this approach, it is not possible to derive the single exergy efficiency for the solar collector.

In Ref. [23] a comparison between both analysis boundaries, technical and physical, can be found. Eq. (29) is used for assessing the solar exergy input in the collector field following the technical boundary. In the physical boundary, only the output from the collector field (Eq. (31)) is regarded. The exergy performance of different solar thermal systems for heating and cooling of a hotel building is analysed dynamically. Results show that using the "technical boundary" the substitution of direct solar systems by fossil fuels seems of advantage. This is due to physical inconsistencies in the definition of the assessment boundary. Great exergy losses occur due to the degradation of solar energy required for the formation process of fossil fuels. However, following the technical boundary, these losses are only regarded in the direct conversion of solar radiation (e.g. in low temperature heat, or electricity) and disregarded in any other indirect conversion, e.g. solar energy stored in biomass and fossil fuels [54].

In addition, results show that using the technical boundary the influence of different control strategies based on lower outlet temperature for the collector field cannot be recognized. Using this approach, even if higher fossil energy savings were achieved by lowering the required outlet temperature, main exergy losses in the system still arise due to the conversion of solar radiation into low temperature heat. Exergy losses due to this conversion process depend mainly on the collector area, which is the same in both cases, and subsequently similar overall exergy efficiencies are obtained.

3. Heating systems

In this section exergy analysis applied to renewable-energy-based heating systems is considered, including solar systems, ground source and solar source heat pumps and biomass boilers.

3.1. Solar systems

3.1.1. Solar thermal systems

Several authors have applied the exergy concept and method for the optimisation of solar thermal collectors as a single component of an energy system. For this aim, the technical boundary (see Section 2.7) is used. Thus, the exergy extracted from the collector (Eqs. (31) and (32)) and the exergy of solar radiation incident onto its surface (Eqs. (29) and (30)) are relevant.

Bejan [60] determines the optimum collector outlet temperature as a function of the incident solar radiation. The author concludes that maximum exergy extraction, i.e. maximum exergy efficiency, is reached if the outlet collector temperature is controlled as a function of the incident solar radiation. In turn, operating the collector to give a constant outlet temperature would decrease the obtained exergy output.

Gunerhan and Hepbasli [58] evaluate the exergy performance of a solar collector. Results are based on measurements from an experimental set up system at the Ege University (Turkey). The system consists on a 2 m^2 collector surface oriented south and tilted 45° over the horizontal plane. The exergy efficiency of the collector is defined according to Eq. (28). However, instead of following the simplified approach proposed by Jeter [55] to evaluate the exergy of solar radiation (Eq. (29)), the approach proposed by Petela [56] is used (see Eq. (30)).

For the exergy evaluation, experimental data for 2.5 h of a summer day are treated as if all processes occurred under steady-state conditions. The temperature drop in the collector is 10–5 K depending on the operating conditions. Inlet temperatures are on the range from 25 to 70 °C. Under this assumption, exergy efficiency of the solar collector is found to be between 2 and 3.5%, depending on the operating conditions: for a given incident radiation, and ambient temperature, increasing outlet temperature increases also the exergy efficiency of the solar collector, despite the energy efficiency is reduced. Trends for the energy and exergy efficiencies obtained are shown in Figs. 4 and 5, respectively, as a function of a parameter, *x*, which is the ratio between the temperature difference between the mean collector and ambient temperatures and the incident solar radiation per collector area. Since opposite trends are found it may be suggested that further comments on the way to combine results from the energy and the exergy point of view to this particular system would be useful. Furthermore, an analysis on the applicability of exergy as decision-making criteria for optimising the design of this particular solar water heating system would also be of great interest.

Xiaowu and Ben [63] analyse the exergy performance of a solar thermal system for domestic hot water (DHW) production following a steady-state approach and the same boundary for the analysis as in Ref. [58]. A 2.5 m^2 collector surface, tilted 50° over the horizontal plane, and 196 litres water storage tank are regarded. Average outdoors temperature of 25 °C and average water temperature from local net of 24 °C is considered. The exergy of solar radiation is included in the analysis but the approach followed by the authors in this regard is not explicitly stated.

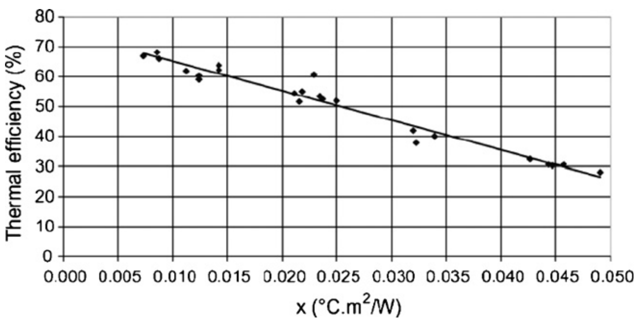

Fig. 4. Energy efficiency for the tested solar collector as a function of the ratio x between the temperature difference between the mean collector and ambient temperatures and the incident solar radiation per collector area [58].

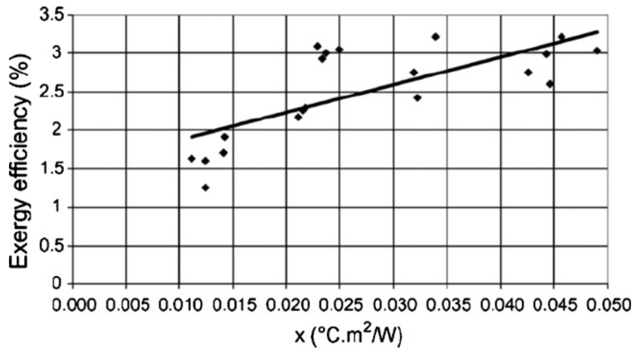

Fig. 5. Exergy efficiency for the tested solar collector the ratio × between the temperature difference between the mean collector and ambient temperatures and the incident solar radiation per collector area [58].

Collector exergy efficiency is defined following Eq. (28), and has values between 1 and 3%, being coherent with the results shown in Ref. [58]. In addition, the authors perform a broader exergy analysis of the system including the storage tank. Overall exergy efficiency for the collector and storage systems is 0.77%.

Luminosu and Fara [61] carry out an exergetic optimisation for the operation and design of a solar thermal collector. The energy and exergy flows of incident solar radiation are assumed to be equal. An optimisation algorithm is developed for an open operation of the collector field, assuming an inlet collector temperature similar to the ambient temperature. This latest condition would be similar to those assumed in Ref. [63]. The exergy efficiency of the solar collector is also in the range of 3%, depending on the regarded operating conditions.

Results from the authors above show that even under optimum operation and design conditions, exergy efficiency of the solar collectors is very low (approx. 3%). This is mainly due to the conversion of solar radiation (high exergy energy source) into low temperature heat. However, as already stated in Section 2.7, regarding this process is physically inconsistent with the evaluation framework used for other energy sources [54]. In other words, when following the "technical boundary" (see Section 2.7), maximizing the output from solar radiation in exergy terms means to maximize collector outlet temperature.

Moreover, if a whole system analysis is carried out, i.e. including the final exergy demand for DHW or space heating as final output of the system, increasing collector outlet temperatures beyond the temperature level of the energy demand might reduce exergy losses in the collector field, but would increase energy and exergy losses in the storage tank, distribution and emission systems for a given demand. Following, greater mismatching between the solar energy supplied and the actual exergy demanded would arise, and the overall exergy efficiency of the whole system would be expected to decrease.

Torio and Schmidt [23] perform a dynamic energy and exergy analysis of two different solar thermal systems for space heating and cooling of a hotel building located in Freiburg (Germany). A solar collector field integrated with a boiler provide heating in winter, and in summer they feed an absorption-cooling machine to provide cooling. The main differences between the systems analysed are shown in Table 5. Different control strategies based on the setpoint for the outlet collector temperature required (80 °C in case I and 55 °C in case III) are used in each of the systems and subsequently different space heating systems demanding different supply temperatures (80 °C in case I and 55 °C in case III) are considered. Energy and exergy flows in the systems are dynamically simulated on an annual basis with hourly time steps. This study does not focus on an exergy analysis of the single

Table 5
Brief description and overall exergy efficiencies for two of the cases analysed in Ref. [23]. Overall exergy efficiencies obtained using the physical boundary.

System	Collector area (m^2)	Storage size (m^3)	Control strategy, $T_{min,outlet}$ $T_{heating}/T_{cooling}$ (°C)	$\Psi_{ove,\ phys}$ (%)
I	100	12	80/80	5.29
III	100	12	55/75	5.83

components of the building systems (e.g. on the solar thermal collector), but on the overall performance of the building systems. Thus, the overall exergy efficiency of the systems as shown in Eq. (33), i.e. related to the exergy of the required final demand, is chosen as a main parameter for comparing the performance of the systems. In Table 5 overall exergy efficiencies obtained using the physical boundary for two of the systems analysed are shown.

In this study the efficiency for the electricity generation is not included, and electricity is then regarded as a direct input into the overall system. Thus, the energy chain from the generation system onwards according to Fig. 1 is considered. Due to the extremely low electricity demand required as auxiliary energy (0.004% of the energy supplied), very small variations of the results can be expected if the electricity generation efficiency would be included.

$$\Psi_{ove} = \frac{Q_{heat,cool}\left(1 - \frac{T_0}{T_{room}}\right)}{Ex_{boil} + Ex_{sol} + Ex_{cool-t} + Ex_{aux}} \tag{33}$$

Results show that reducing the required outlet temperature from the collector field allows higher solar fractions to be achieved, substituting a greater amount of fossil fuels by low-temperature solar heat. Energy from solar thermal field has a quality factor of 0.174 and 0.161 in cases I and III, respectively. In consequence, higher overall exergy efficiencies for case III are partly due to the greater substitution of fossil fuels by solar heat, and also due to the lower exergy content of the solar heat due to its lower supply temperatures.

However, simplified control strategies for the collector field based on a constant minimum collector outlet temperature were used in this study [23]. In consequence, the energy output of case I is significantly limited by the higher outlet collector temperature. Further investigations on the performance of the whole system (energy chain in Fig. 1) with variable collector outlet temperatures are required.

3.1.2. Hybrid photovoltaic–thermal solar systems

The use of exergy as parameter for evaluating the performance of hybrid photovoltaic/thermal (PV/T) solar systems enables a direct quantitative and qualitative comparison of the electrical and thermal energy yielded from the solar collector [69]. A methodology for thermodynamic exergy-based analysis for hybrid PV/T solar collectors is presented by Coventry and Lovegrove [70] in addition to other possible analysis methodologies such as market analysis and environmental analysis based on avoided greenhouse gas emissions.

Saitoh et al. [71] evaluate the energy and exergy performance of a hybrid solar PV/T collector as compared to the efficiencies of solar thermal and photovoltaic applications when installed individually. Combined energy and exergy efficiencies are calculated as shown in the following equations:

$$\eta_{comb} = \frac{E_{PV} + Q_{coll}}{G} = \eta_{PV} + \eta_{coll} \tag{34}$$

$$\Psi_{comb} = \frac{E_{PV} + Q_{coll}\left(1 - \frac{T_0}{T_{in}}\right)}{G\left(1 - \frac{T_0}{T_{sol}}\right)} \tag{35}$$

It is remarkable that the authors calculate the exergy efficiency of the thermal collector as a function of the supply temperature instead of using Eq. (31). This might lead to an unaccurate estimation of the exergy output from the solar thermal collector. Ambient temperature of 25 °C, constant radiation of 700 W/m^2 and a water supply temperature of 40–50 °C are regarded for the comparison. Energy and exergy efficiencies for the single and hybrid systems are shown in Table 6. Although the solar thermal has the highest energy performance, hybrid PV/T collector shows the best exergy performance.

Similar results are obtained by Fujisawa and Tani [72]. The authors evaluate the energy and exergy performance of two hybrid solar collectors (with and without glass cover) and compare them to single thermal and photovoltaic systems. Evaluation is based on experimental measured data over one year. Main results from their analysis are shown in Table 7. The exergy output density for the combined photovoltaic (PV) and flat plate collector (FPC) system refers to a system where solar thermal collectors and photovoltaic modules are used together side by side as independent components. The percentage difference on the exergy output density is calculated taking exergy output density of such a combined system as reference. However, the approach followed for calculating the exergy of the thermal output from collector is not specified. According to their study, the hybrid system without glass cover has the best exergy performance, since a higher cooling effect of the PV cells can be achieved and, thus, exergy output related to electricity production can be maximized. In turn, exergy associated to thermal energy output is strongly decreased as compared to the covered collector. Results in this study are presented in terms of annual energy yield relative to the collector area. Respective energy and exergy efficiencies cannot be determined for the values of annual incident irradiance are not given.

Table 6
Comparison of energy and exergy efficiency of solar thermal, photovoltaic and hybrid collectors [71].

	Solar collector	Photovoltaic	Hybrid collector
Energy efficiency (%)			
Heat	46.2	–	32
Power	–	10.7	10.6
Total	46.2	10.7	42.6
(Single) Exergy efficiency (%)			
Heat	4.4	–	2.1
Power	–	11.2	11.2
Total	4.4	11.2	13.3

Table 7
Comparison of exergy output from flat plate solar thermal (FPC), photovoltaic (PV) and hybrid collectors without glass cover (PV/T^{I}) and with one glass cover (PV/T^{II}).

Parameter	FPC	PV	PV/T^{I}	PV/T^{II}
Electrical exergy gain (kWh)	–	72.6	78.4	66.0
Thermal exergy gain (kWh)	6.0	–	2.4	5.6
Total exergy gain (kWh)	6.0	72.6	80.8	71.5
Installation area (m^2)	0.51	0.7	0.7	0.7
Exergy output density (kWh/m^2)	65.0[a]	115.4	102.1	
Percentage difference (%)	–		76	57

Results from Ref. [72], taken from the review in Ref. [69].
[a] Exergy output density here refers to a system combining the FPC and PV systems as independent components.

Chow et al. [21] also compare the performance of glazed and unglazed PV/T collectors for Hong Kong Chinese climatic conditions. The authors accomplish an energy and exergy performance analysis based on dynamic simulations for 1 day. The hybrid PV/T systems consisted on a thermosyphon 1.44 m^2 hybrid solar collector connected to a 155 l storage tank. The transmissivity of the glass cover is assumed to be 0.88. Average solar radiation and ambient temperature for the simulated day are 600 W/m^2 and 25 °C. Inlet water temperatures of 19 °C are assumed at the early morning for daily simulations. Energy performance of the glazed collector is better, for heat output is maximized. However, glazing reduces electrical output due to the optical efficiency of the cover and overheating of the PV cells. In exergy terms the thermal contribution becomes irrelevant as compared to the electrical exergy output. In consequence, exergy efficiencies are higher for the unglazed hybrid collectors. Exergy efficiencies for the hybrid collectors investigated are on the same range as those presented by other authors (≈13%).

Bosanac et al. [73] evaluate the performance of three different types of hybrid PV/T solar collectors for Danish climate: hybrid collector without glass cover, with 15 mm acrylic glass cover with good optical transmissivity, and collector with 15 mm acrylic glass cover and air gap between the absorber and PV cells. The authors perform annual dynamic energy analysis using climatic data from Danish test reference year (TRY). However, exergy analysis is performed on a steady-state basis, using the annual energy flows from dynamic energy analysis and choosing a constant reference temperature of 20 °C. The authors find that the last two configurations increase thermal performance of the hybrid collector, without substantially lowering its electrical efficiency. In consequence, using glass cover allows similar electrical exergy efficiencies (12%) and higher thermal exergy efficiencies (3%) as compared to those obtained without cover (1.8%). Since the outlet temperature from the thermal component of the hybrid collector is relatively close to outdoor reference temperature, the choice of a dynamically varying reference temperature throughout the year would influence the results for the thermal exergy performance. Outdoor air temperature for Danish climatic conditions can be significantly lower than 20 °C. Therefore, it might be expected that using a variable dynamic outdoor temperature as reference would lead to a higher exergy performance of the thermal unit, thus increasing the difference in the performance between the glazed and unglazed systems.

In Ref. [73], higher overall exergy efficiencies can be obtained using glass cover (14%), i.e. increasing the insulation of the collector envelope, as compared to systems without cover (12%). These conclusions are different to those obtained by Fujisawa and Tani [72] and Chow et al. [21] for Japanese and Chinese climatic conditions, respectively. The different behaviour of the efficiency with and without glass cover may be due to the different climatic conditions: whereas in hot climates cooling of PV cells has a strong influence on the electrical performance of the system, in colder climates insulating the system to increase thermal performance does not imply any significant reduction on the electrical performance of the hybrid collector.

Results in Tables 6 and 7 show that the solar thermal system strongly determines the energy performance figure, whereas the exergy efficiency is greatly determined by the efficiency of the electricity production. Thus, on a component level, optimum operating conditions should always try to maximize the power output of the system, instead of the thermal energy yield.

However, all analyses reviewed here refer only to the hybrid solar collector as component of an energy system. In turn, optimisation of the whole energy system should also be regarded, i.e. including the demand that the system is intended to provide. In the case of buildings, low temperature heat usually represents a greater proportion of the energy demand than electricity required for appliances and auxiliary energy. Since for climatisation purposes usually high exergy energy sources are used, increasing solar thermal yield would also increase significantly the efficiency of space heating and DHW production [57]. Therefore, if an analysis of the whole energy chain is done for the particular case of building climatisation, electrical output might not be the most relevant output of the hybrid system.

Unfortunately, to the best of the authors' knowledge, such kind of comprehensive analysis of the systems including the ultimate energy demand or service to be delivered as shown in Fig. 1, are not available.

3.2. Heat pumps

3.2.1. Ground source heat pumps

Modeling the ground and the building indoor environment as heat reservoirs at the temperatures, respectively T_g and T_{room} (with $T_0 < T_g < T_{room}$ in the heating mode), the exergy balance of a ground source heat pump system (GSHP) may be written in the form:

$$\dot{E}x_g + \dot{W} = \dot{E}x_{room} + \dot{I} \tag{36}$$

where $\dot{E}x_g$ and $\dot{E}x_{room}$ are the exergy transfer rate, respectively at the ground and at the indoor environment, $\dot{W}$ is the power input and $\dot{I}$ is the irreversibility production rate.

The overall functional exergy efficiency ψ of a GSHP may then be expressed as in Eq. (37), where $\dot{Q}_{room}$ is the heat transfer rate to the indoor environment and COP the heat pump coefficient of performance. Coherently with the hypothesis of the ground as a heat reservoir, the ground temperature to be used in Eq. (37) is the undisturbed value:

$$\Psi_{ove} = \frac{\dot{E}x_{room}}{\dot{E}x_g + \dot{W}} = \frac{\dot{Q}_{room}\left(1 - \frac{T_0}{T_{room}}\right)}{(\dot{Q}_{room} - \dot{W})\left(1 - \frac{T_0}{T_g}\right) + \dot{W}} = \frac{COP\left(1 - \frac{T_0}{T_{room}}\right)}{1 + (COP - 1)\left(1 - \frac{T_0}{T_g}\right)} \tag{37}$$

In Eq. (37) the exergy inputs to the heat pump are the power input and the environmental exergy from the natural reservoir. Another exergy efficiency definition may be found in literature [74,75], where only the electrical power is taken into account as the exergy used (Eq. (38)):

$$\Psi = \frac{\dot{E}x_{room}}{\dot{W}} = \frac{\dot{Q}_{room}\left(1 - \frac{T_0}{T_{room}}\right)}{\dot{W}} = COP\left(1 - \frac{T_0}{T_{room}}\right) \tag{38}$$

According to this approach, ground exergy is considered "free" exergy. Both definitions assume that the boundary of the analysis is at the indoor environment, while some authors [67,38,39] prefer to limit the boundary to the output of the emission system in the building. Considering as an example the case of a fan coil unit, the exergy efficiency of the overall system is shown in Eq. (39), where $\dot{E}x_{a,out}$ and $\dot{E}x_{a,in}$ are the exergy flow of the air stream, respectively at the outlet and inlet of the fan coil unit.

$$\Psi = \frac{\dot{E}x_{a,out} - \dot{E}x_{a,in}}{\dot{W}} \tag{39}$$

It is clear that the larger the boundary for the exergy analysis, the lower the exergy efficiency value, as shown in many examples in Refs. [31,76]. Indeed, recalling the single and overall exergy efficiencies defined in Section 2.3, the larger the boundary is, the greater is the number of single exergy efficiencies that are multiplied to give the overall efficiency.

All the exergy definitions in Eqs. (37)–(39) disregard the electricity production step, that is, they set the boundary of the analysis at the heat pump level only.

Exergy analysis has been applied to both vertical ground source heat pumps [74] and horizontal ones [67,38]. A common approach may be found in the mentioned papers: based on measured data from steady state operation in the heating mode of the installed systems, the authors apply an exergy balance equation to every component, calculate the irreversibility rate in each part and finally the overall exergy efficiency. The main purposes are highlighting major entropy productions and suggesting potential improvements.

In Ref. [74] the authors report about a GSHP connected to a classroom in the Solar Energy Institute of the Ege University in Izmir, Turkey. The heat pump is coupled with a 50 m long U bend ground heat exchanger and, on the distribution side, with two fan coil units. The authors point out that the main irreversibility lies in the motor-compressor assembly. In order to reduce it, besides improving the motor efficiency, improving the heat exchangers to bring the condensing and evaporating temperatures closer together would be an important issue.

In Ref. [67] a GSHP connected with a horizontal GHEx buried at 1 m depth installed at the Ege University in Izmir, Turkey, is considered. The most important irreversibility is in the fan coil units. Looking at the heat pump unit alone, the major irreversibility is in the condenser.

In Ref. [38] a GSHP alternatively coupled with two horizontal GHEx, buried at the depth of 1 and 2 m, and installed at the University of Firat, Elazig, Turkey, are compared. For both systems the main irreversibility lies in the condenser fan on a whole system basis and in the evaporator on a heat pump basis. The system coupled with the deepest GHEx comes out to be the more efficient, from both an energy and exergy perspective. However, looking at the ratio between the two *COPs* on one side and the ratio between the exergy efficiencies on the other, it is interesting to notice that the advantage of the deepest system on an energy basis is reduced on an exergy basis.

A synthesis of the above described studies is shown in Table 8, where some temperature levels relevant for the exergy analysis are shown (ground temperature, outdoor and indoor temperature, reference temperature), as well as the overall *COP*, the heat pump unit exergy efficiency, the overall exergy efficiency with the relative boundary adopted and finally a list of the location of the irreversibilities in order of relevance.

By comparing the results, several observations may be derived. First, it is important to notice that the range of the overall exergy efficiency (from 2.9% to 80.7%) is much larger than that of the energy efficiency or *COP* (from 1.65 to 2.8). This result may be influenced by the different boundaries adopted and by the different choices of the reference temperature. Actually, T_0 is set equal to the average atmosphere temperature 25 °C in Ref. [74], to the design heating temperature for the site in Ref. [67], and to the average outdoor temperature measured during the monitoring in Ref. [38]. As already discussed in Section 2.1 the dead state chosen plays a crucial role in the exergy analysis of climatisation systems. Esen et al. [38] actually find that the overall exergy efficiency of the GSHP coupled with the two horizontal ground heat exchangers (named HGHE1 and HGHE2 in Fig. 6) is quite sensitive to the reference temperature, i.e. it strongly decreasing as T_0 increases.

3.2.2. Solar-assisted heat pumps

A solar-assisted heat pump (SAHP) is studied both experimentally and theoretically in Ref. [66]. The prototype heat pump is a water-to-air system where the refrigerant evaporation takes place directly within the solar collector. The authors derive the exergy balance in Eq. (40) for the solar heat pump, where $\dot{E}x_{sol}$ is

Table 8
Comparison among different GSHP exergy analyses [74,38,67].

System description	Location	$T_{outdoor,a}$ (°C)	T_{room} (°C)	T_g (°C)	T_0 (°C)	Overall *COP*	$\Psi_{single,HP}$	Ψ_{ove}	Boundary	Irreversibility list	Ref.
GSHP vertical GHEx (50 m long U pipe) fan coils	Izmir, Turkey	8.14	19.8	12.99	25	1.65[a]	–	2.9%	Indoor air	1. Motor-compressor 2. Condenser 3. Expansion valve 4. Evaporator 5. Fan coil 6. GHEx	[74]
GSHP horizontal GHEx (1 m depth) fan coils	Izmir, Turkey	0	–	15	1[b]	2.72	83.2%	80.7%	Fan coil	1. Fan coil 2. Evaporator 3. Condenser 4. Compressor 5. Expansion valve 6. Ghex	[67]
GSHP horizontal GHEx (1 m depth)	Elazig, Turkey	1	–	–	1	2.5	62.9%	53.1%	Condenser fan	1. Condenser fan 2. Evaporator 3. Condenser 4. Compressor 5. Capillary tube 6. GHEx	[38]
GSHP horizontal GHEX (2 m depth)	Elazig, Turkey	1	–	–	1	2.8	65.4%	56.3%	Condenser fan	1. Condenser fan 2. Evaporator 3. Condenser 4. Compressor 5. Capillary tube 6. GHEx	[38]

[a] Calculated by the authors on the basis of the data in Ref. [74].
[b] Design heating temperature for Izmir [67].

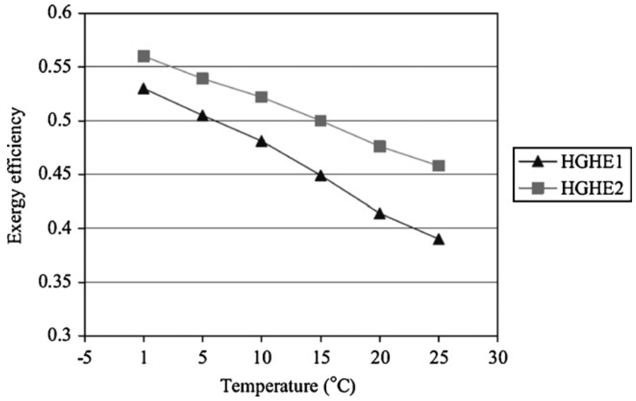

Fig. 6. Exergy efficiency of ground source heat pump HGHE1 and HGHE2 versus reference temperature [38].

the solar exergy incoming, $\dot{E}x_{in-a}$ and $\dot{E}x_{out-a}$ are the exergy flows, respectively of the air inlet and outlet at the condenser, $\dot{W}$ is the power input and $\dot{I}$ is the irreversibility production rate:

$$\dot{E}x_{sol} + \dot{E}x_{in-a} + \dot{W} = \dot{E}x_{out-a} + \dot{I} \tag{40}$$

In case another condensation system on the delivery side is adopted, $\dot{E}x_{in-a}$ and $\dot{E}x_{out-a}$ should be replaced by the exergy flows at the inlet and outlet of the secondary side of the condenser. Adopting a physical boundary for the conversion of solar energy into heat (see Section 2.7), the exergy of the solar energy absorbed is calculated as in Eq. (32):

$$\Psi = \frac{\dot{E}x_{out-a} - \dot{E}x_{in-a}}{\dot{E}x_{sol} + \dot{W}} \tag{41}$$

Although the efficiency of the electricity production is not included in Eq. (41), experimental values of the exergy efficiency are found to be quite low, ranging approximately from 1.5% to 3.5%. The highest irreversibility is located in the collector–evaporator, followed by the condenser. A comparison between the measured refrigerant outlet temperatures from the condenser when the heat pump operates in the conventional air source mode and in the solar-assisted mode shows that the SAHP performance is limited by the condenser heat exchange area. At the same time an optimal evaporation temperature, maximizing the exergy efficiency of the collector-evaporator, is identified. This optimal parameter is a function of the air temperature and the sol-air temperature of the collector, thus involving ambient parameters and collector properties. In a following study by the same authors [77] actual measured evaporation temperature of the SAHP is compared with the optimal value. The difference between the optimal and the actual values is found to be an exponentially decreasing function of the wind velocity, resulting in an average temperature difference of 46 °C. The authors then suggest that an automatic control variation of the fluid flow in the condenser, using the sol-air temperature as the monitoring parameter, would reduce the observed deviation from optimal operation and improve the system efficiency.

Dikici and Akbulut [68] perform an energy and exergy analysis based on experimental data of different sources heat pumps installed in Turkey. They consider a solar-assisted heat pump (SAHP), a ground source heat pump (GSHP), a conventional air source heat pump (ASHP) and test some multiple sources configurations, namely the solar-assisted ground source (SAGSHP), the ground source air source (GSASHP) and the solar-assisted air source (SAASHP). The energy performance is measured by the *COP* while an exergetic performance coefficient (*EPC*) is used to evaluate the heat gain by the heat pump compared with the

Table 9
Energy and exergy comparison among different sources heat pumps [68].

Heat pump system	Sources	Energy *COP*	Exergy *EPC*
SAHP	Solar	2.95	1.23
GSHP	Ground	2.44	2.39
ASHP	Air	2.22	2.31
SAGSHP	Solar-ground	3.36	1.04
SAASHP	Solar-air	2.90	0.74
GSASHP	Ground-air	2.14	1.08

dissipation to the environment. The *EPC* is defined as shown in the following equation:

$$EPC = \frac{\dot{Q}_{cond}}{\dot{E}x_{loss,total}} \tag{42}$$

In Eq. (42) $\dot{Q}_{cond}$ represents the heat transfer rate at the condenser and $\dot{E}x_{loss,total}$ is the total exergy loss rate. As shown in Table 9, the highest *COP* among the single sources heat pumps is achieved by the SAHP, while considering also the multiple sources configurations the SAGSHP had the best energy performance. On the contrary, the highest *EPC* is achieved by the GSHP.

This case is representative of parallel energy and exergy analyses leading to different conclusions. A question may then arise on how to combine the results or, referring to the different sources heat pumps in the above-mentioned study, which system should be considered the best one. A possible conclusion could be that the SAGSHP provides the highest energy savings, but as indicated by the exergy efficiency parameter, improvements on the system would be necessary. Looking at the three heat pumps using a single source, it is clear that the energy analysis alone would lead to choose the SAHP as the most performing, while the added exergy analysis probably leads to choose the GSHP, which combines a good *COP* with the best *EPC*.

The irreversibility analysis at the components level carried out by Dikici and Akbulut [68] points out that the major irreversibility in the SAHP, SAASHP and SAGSHP lies in the solar collector. An exergy analysis of a similar SAGSHP is also reported by [67], who on the contrary find that the solar collector has high exergy efficiency (97.9%) and represents the lowest source of irreversibility within the whole system. Since in both cases a physical boundary for solar exergy is adopted, again we may argue that the different choice of the reference temperature (T_0 = 1 °C in Ref. [67] and T_0 = 25 °C in Ref. [68]) plays an important role in the achieved results. Furthermore, a different energy efficiency regarded for the solar collectors would also be reflected on the exergy performance. Thus, it would be useful to provide both energy and exergy data of any studied system or component.

3.3. Biomass-based heating systems

Biomass is regarded as a renewable energy source. According to the German regulation [78], wood pellets and bricks for warm water and space heating applications are regarded as mainly renewable energy source, with a closed CO_2 emissions cycle, independently of the amount of energy demand they are covering. Their environmental impact is restricted to the depletion and emission potential of fossil fuels used during the extraction and processing phase of their production. Subsequently, with this evaluation framework, an environmentally friendly manner of refurbishing an old building could just include exchanging the old boiler by a wooden-based one, without applying any measures to lower down the energy consumption of the building.

However, the renewability and sustainability of wood as fuel is restricted to the conditions and constrains of its use. In other words, wood is not endlessly available as renewable source to cover any energy demand, for cutting down the total forested area of a country would be neither renewable nor sustainable, and the CO_2-emissions cycle could not be regarded as closed any more. Evaluation frameworks including this regard, such as the so-called "bio-budget" method, which include the maximum energy demand that could be covered sustainably and renewably by wood-based fuels, have been developed [79].

In Refs. [80,40], a wood-pellet boiler is compared from an exergy perspective to several other building systems, including a condensing boiler and a ground source heat pump. The building chosen for analysis is a single-family house built among 1995–2000, according to the German residential building typology [81]. Further details about the building can be found in [40] and [81]. Steady state exergy analyses assuming a reference outdoor temperature of 0 °C and desired indoor air temperature of 21 °C are performed.

In Fig. 7 the comparison between the primary energy and exergy inputs, divided into its fossil and renewable parts, is shown graphically. Despite the regarded renewability of the wooden fuel, its exergy content is still comparable to those of the liquefied natural gas (LNG). The whole energy chain for building energy supply as shown in Fig. 1 is included in the analysis, i.e. conversion efficiency for electricity production is regarded.

Total energy and exergy demands of the building analysed, including space heating and DHW production, are 3446 and 317.4 W, respectively. As a ratio of the exergy demanded and total exergy supplied in each case, overall exergy efficiencies of the systems under analysis can be obtained, even if they were not explicitly shown in Refs. [80] or [40]. Overall exergy efficiencies for the condensing boiler, wood-pellet boiler and GSHP are 5.9%, 5.53% and 7.4%, respectively.

Lower energy efficiency has been assumed for the pellet boiler than for the condensing boiler based on LNG (for higher energy input is required when a pellet boiler is assumed to cover the same energy demand). Due to the high quality of wood as an energy source, i.e. its high exergy content, overall exergy efficiency of this system could be at best similar to that obtained for the condensing boiler (assuming the same energy efficiency for both boilers).

In turn, the ground source heat pump allows harvesting a significant amount of renewable environmental heat from the ground with very low exergy content. This system shows the lowest exergy input required for providing the given demands, i.e. it exhibits the highest overall exergy efficiency and lowest exergy input of the three building systems analysed.

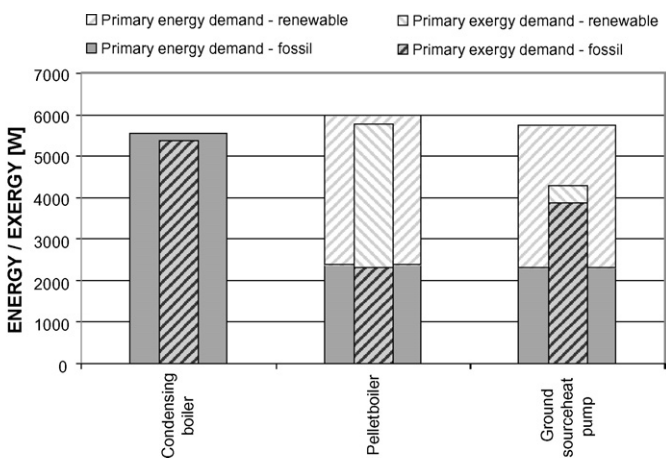

Fig. 7. Primary energy and exergy demand (input) into three different building systems analysed. Primary demands are divided into their renewable and fossil parts [40], modified.

From the comparison of the three building systems it can be concluded that the ground source heat pump is the most advisable, from an exergy perspective. Pellets are regarded here as a renewable energy source, and show the minimum primary energy input, i.e. minimum environmental impact. However, their low exergy efficiency rises partly due to the use of a high quality energy source (wooden fuels) for low exergy purposes (space heating and DHW production). To avoid this mismatching between the quality levels of supply and demand, and better exploit the whole exergy content of wooden fuels, they should firstly be used for providing high exergy demands, e.g. electricity production. Similar conclusions could be derived for the LNG-fired condensing boiler.

Actually, the exergy efficiency of the building systems is increased if LNG or pellets are used to produce electricity and power a ground source heat pump, instead of directly burning these high quality energy carriers to produce low temperature heat.

Thus, in this study a kind of competition between the aim to reduce total primary exergy demand (accomplished through the GSHP) and the aim to reduce fossil primary energy demand (accomplished through the biomass boiler) is addressed. In Refs. [80] and [40], it is claimed that both primary energy and primary exergy demands should try to be minimized. Thus, combined energy and exergy analysis is required, and the choice on the optimum system should be done on the basis of minimizing the two parameters. Following this approach, a system with very low primary energy input but with significantly higher primary exergy input as other conventional systems (i.e. biomass boiler) would not be advisable. However, as already pointed out in Section 3.2.2, further debate and analysis would be necessary on how to combine energy and exergy-based indicators for decision-making on the choice and optimisation of energy systems for buildings.

4. Cooling systems

In this section renewable energy-based cooling systems are reviewed, starting from evaporative cooling systems. A detailed analysis of results from exergy analysis of thermally driven compression (TDC) systems and, particularly, of solar-driven TDC units, follows. Similarly, results from analysis of dessiccant cooling systems (DEC) in general and solar-driven DEC systems are also presented.

4.1. Evaporative cooling

Chengquin et al. [25] present an exergy-based comparison among different evaporative cooling schemes. Before approaching the analysis, the authors discuss the opportunity of defining a novel dead state for HVAC applications. Since unsaturated atmospheric air still possesses available energy, they suggest adopting the atmospheric state (T_0, p_0, ω_{0s}) as the reference, where ω_{0s} denotes the humidity ratio of the saturated air. This choice would reduce the exergy loss in HVAC systems due to the condensed water flow, which is actually not important in the effective use of energy.

In order to analyse the evaporative cooling process Chengquin et al. [25] break down the exergy of air into three components, namely thermal, mechanical and chemical. Evaporative cooling is accomplished by sacrificing the chemical exergy of the primary and/or secondary air in order to increase the thermal exergy of primary air. Subsequently they define a thermal, a chemical and a total exergy efficiency ratio (EER), as the ratio between the relative exergy increase of the primary air and the fan power consumption, taken as the exergy decrease in mechanical exergy. They consider the following evaporative cooling schemes: direct evaporative (DEC), indirect evaporative (IEC), direct–indirect (DIEC) and finally regenerative (REC). They find that the REC scheme has the best performance or the largest total EER.

In a more fundamental paper by Wepfer et al. [26] several psychrometric processes are analysed from an exergy perspective. Among them, evaporative cooling is also considered. The reference state here is assumed to be T_0 = 35 °C, p_0 = 1 atm and ω_0 = 0.01406. Differing from [25], the reference state is not saturated air. The analysis is focused on the air and water streams, so that no power consumption from the auxiliary energy is taken into account. The exergy efficiency of the process is then calculated as shown in Eq. (43). Following this definition, exergy efficiency for the evaporative cooling under the chosen reference environment is found to be 38%:

$$\psi = \frac{\dot{E}x_{a,out} - \dot{E}x_{a,in}}{\dot{E}x_{w,in}} \tag{43}$$

4.2. Thermally driven compression systems

In Fig. 8 a simplified schema of the components of an absorption-cooling machine is shown. Eqs. (44)–(46) show the analytical expression for the evaluation of the single exergy efficiency in the absorption machine as a component of an energy system. In Eq. (47) the expression for the overall exergy efficiency, regarding the exergy of the cooling demand as desired output is shown:

$$Ex_{in} = \frac{W_{circ}}{\eta_{el}} + Ex_{heat,gen} + m_{rej}(ex_4 - ex_3) + m_{rej}(ex_6 - ex_5) \tag{44}$$

$$Ex_{des,out} = m_{cool,w}(ex_2 - ex_1) \tag{45}$$

$$\Psi_{single} = \frac{m_{cool,w}(ex_2 - ex_1)}{\frac{W_{circ}}{\eta_{el}} + Ex_{heat,gen} + m_{rej}(ex_4 - ex_3) + m_{rej}(ex_6 - ex_5)} \tag{46}$$

$$\Psi_{ove} = \frac{Q_{cool}\left(1 - \frac{T_0}{T_{room}}\right)}{\frac{W_{circ}}{\eta_{el}} + Ex_{heat,gen} + m_{rej}(ex_4 - ex_3) + m_{rej}(ex_6 - ex_5)} \tag{47}$$

Pons et al. [82] perform a comparison of thermally driven compression systems for different uses. Criteria for the comparison are based both on the first and second law of thermodynamics. For this aim, on one hand the *COP* of the machines and the so-called "thermodynamic efficiency", defined as a ratio between the real *COP* of the cooling machine and the ideal *Carnot COP* that could be achieved between the temperature levels in which the cooling cycle is operated, are used. Therefore, the thermodynamic efficiency is not equivalent to the exergy efficiency but rather shows how well a cycle performs as compared to ideal conditions, i.e. pinpoints the magnitude for possible improvements and optimisation in the cycle. Eqs. (48) and (49) show the definition of the thermodynamic and *Carnot* efficiencies:

$$\eta_{thdyn} = \frac{COP}{COP_{Carnot}} \tag{48}$$

$$COP_{Carnot} = \frac{(1 - (T_{abs}/T_{gen}))}{((T_{cond}/T_{evap}) - 1)} \tag{49}$$

In this study [82], several single, double and triple effect absorption machines are analysed based on typical operation conditions for each of them. Despite double and triple effect operation always lead to higher *COP* of the cooling machines,

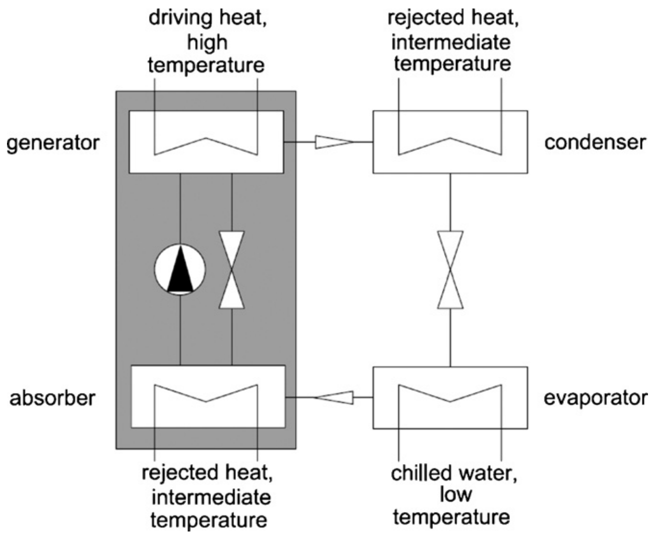

Fig. 8. Simplified schema of the components and operating cycle in an absorption cooling machine; the solution heat exchanger between the rich and poor refrigerant solutions, usually found between the absorber and generator is not shown in the schema [91], modified.

thermodynamic efficiency of single-effect cycles is higher. This indicates that the thermodynamic potential of the cooling cycles is maximized and irreversibilities are minimized when lower driving temperatures are used for the operation of the cooling machines.

Sencan et al. [83] perform steady-state exergy analysis of an absorption machine for different operative conditions. The electricity demand for the solution pump and frictional losses in the whole cycle are neglected. The main operative conditions regarded by the authors are shown in Table 10. In addition, results from exergy analysis are presented also in Table 10, in terms of irreversibilities present in each component of the absorption-cooling machine. According to the authors, irreversibilities in the generator are due to the necessity of super-heat the refrigerant vapour above its evaporation point in pure state, i.e. without being immersed in a solution. Similarly, absorption heat would also be the main cause of irreversibilities in the absorber. Exergy losses in the evaporator and condenser are due to the temperature difference between both sides of the heat exchanger. The authors investigate also the effect of higher generator temperatures in the energy and exergy efficiencies of the thermodynamic cycle. As the driving (i.e. generator) temperature increases from 60 to 120 °C, *COP* increases also from 0.77 to 0.8. In turn, overall exergy efficiency decreases from 0.32 to 0.12. As driving temperature increases more cooling energy can be produced, i.e. higher exergy output. However, higher irreversibilities would also arise in the heat transfer processes, i.e. higher temperature differences in the heat exchange processes, which are not able to compensate for the surplus exergy output obtained.

In addition, the influence of different chilled water temperatures (i.e. cooling water supplied to the room cooling units) in the energy and exergy performance of the absorption machine is also investigated. The authors show that lower chilled water temperatures (from 20 to 12 °C) decrease the *COP* of the cooling machine (from 0.8 to 0.77), but in turn, increase the single exergy efficiency of the unit (from 0.14 to 0.16). This last effect is due to the higher exergy content of the energy output if it happens at lower temperatures. This conclusion is valid for the analysis of the absorption machine as a component of a whole energy system (i.e. single exergy efficiency). However, lower chilled water temperatures would decrease the exergy efficiency of the cooling transfer to the room air in a building, for desired indoor air temperature

Table 10
Comparison of the main assumptions and results from exergy analysis of different absorption machines.

System description	T_0 (°C)	T_{gen} (°C)	T_{evap}	T_{cond}	COP	Ψ_{single}	Irreversibility list and magnitude	Ref.
LiBr (absorption)	25	100	10	27	0.8	0.16	Absorber (62%) Generator (14%) Solution heat exchanger (11%) Condenser (7%) Evaporator (6%)	[83][a]
LiBr (absorption)	27	40–100	2–10	34–40	–	0.55–0.65	Generator (32–43%) Absorber (25–33%) Condenser (16–24%) Evaporator (10–15%) Solution heat exchanger (2.5–6.5%) Throttling valve (0.5–1.4%)	[85]
NH_3 (absorption)	–	–	–	–	–	–	Absorber (34%) Generator (22%) Solution heat exchanger (21%) Condenser (16%) Evaporator (7%)	[84][b]
NH_3 (absorption)	20	140	−10	20	–	–	Absorber (40%) Generator (39%) Condenser (16%) Throttling valve (3%) Evaporator (1.1%)	[33][c]

[a] Exergy input from cooling tower of the absorption cycle is not included in the single exergy efficiency of the absorption cycle.
[b] Irreversibilities in the throttling valve were disregarded.
[c] Solution heat exchanger is not included in the thermodynamic cycle.

would be the same in any case (e.g. 26 °C). Thus, different conclusions might be obtained if an analysis of the whole energy supply chain (Fig. 1), i.e. including the final cooling demand as desired output, would be performed.

It is important to remark that in Ref. [83], the exergy input from the absorber and condenser into the absorption cycle, $m_{rej}[(ex_4 - ex_3) + (ex_6 - ex_5)]$ in Eqs. (46) and (47), is not included as an input in the overall exergy efficiency. However, variations of the overall exergy efficiency including the energy input from the cooling tower are expected to be low.

Boer et al. [84] perform also steady-state analysis of water–ammonia absorption machine for cooling applications. A parametric study is also carried out to determine the influence on the whole cycle of varying the operative conditions on one of the components. Results are presented in terms of irreversibilities existing in each component. Results for the irreversibilities show the same trend and order than in Refs. [83] and [33]. However, the magnitudes of the irreversibilities differ significantly. This might be due to the different operative conditions and reference temperature considered for the analyses. Unfortunately detailed data regarded in Ref. [84] are not available.

Khaliq and Kumar [85] carry out exergy analysis for water–ammonia and lithium–bromide absorption cycles under steady state conditions. Results for the relative irreversibilities in each component are shown in Table 10. The ranges correspond to the parametric study carried by the authors in order to check the influence on the whole cycle of reducing irreversibilities on one single component. The authors conclude that exergy efficiency increases for increasing absorber temperature. However, it is important to remark that significantly higher exergy efficiencies were obtained by the authors as compared to values in Ref. [83]. Khaliq and Kumar [85] do not show explicitly the definition used for the exergy efficiency in their study. The use of the simple exergy efficiency definition would probably help explaining the great mismatching.

Due to the link between the different components in the system, the optimisation of one of them might lead to reduce the performance on the rest. Results from the parametric study in Ref. [84] show that an increase in the temperature drop in the generator would lead to an increase in the irreversibilities on this component, but in turn, reduces significantly the irreversibilities in the absorber. By these means, reductions in the total irreversibilities in the system can be achieved [84].

The impact of optimising each of the single components on the overall performance can also be assessed with the method presented in Ref. [33] (see Section 2.2). Here, the link between the irreversibilities in the different components is done in the form of endogenous and exogenous irreversibilities. The authors conclude that even though irreversibilities are higher in the absorber, efforts should be directed to optimising the generator for it has high avoidable endogenous irreversibilities. Furthermore, they conclude that this would also lead to reductions in the irreversibilities in other components with high exogenous avoidable irreversibility rates.

Subsequently, unless a detailed analysis such as described in Ref. [33] is done, optimisation should always be done from a holistic perspective, aiming at reducing the irreversibilities on the whole system and not only on the single components. Parametric analysis like those done by [84] and [85] are also useful for this aim.

All papers reviewed here focus on the energy and exergy analysis of absorption cooling machines as a single component of an energy system. As already stated in previous sections referred to the analysis of other energy systems, holistic analysis including the whole energy supply chain and the final demand to be provided would be of great interest.

4.2.1. Solar-driven TDC systems

Ravikumar et al. [86] perform an exergy analysis of a solar-assisted double effect absorption cycle. In Ref. [84] the water–ammonia absorption cooling system analysed is also coupled with solar collectors. Khaliq and Kumar [85] perform exergy analysis in LiBr–H_2O and NH_3–H_2O absorption-cooling cycles coupled with solar collectors. Izquierdo et al. [87] perform a comparison between the exergy performance of solar-driven single and double effect absorption chillers. Ezzine et al. [88] perform a thermodynamic analysis of a solar-driven ammonia–water double-effect absorption chiller.

However, the authors above focus on the performance of the single components within the absorption machine and cycle and energy and exergy performance of the solar collectors is not included in the scope of the analysis. Following, the thermal energy input into the generator, required for driving the thermally driven compression cycles, is treated as an energy flow at a given temperature (e.g. collector outlet). But an exergy assessment of the production of that heat flow is left out of the analysis. Thus, if the analysis solely focuses on the absorption machine the advantage of coupling thermally driven compression cycles with solar energy cannot be recognized. In turn, an overall exergy analysis of the system, i.e. including the boiler or solar thermal system providing the generation heat, would be able to pinpoint improvements in the use of this technologies when solar thermal units provide a part of the required driving heat. The benefits of using solar energy to drive absorption cycles would significantly depend on the approach chosen for the solar energy assessment (physical or technical boundaries as discussed in Section 2.7).

4.3. Desiccant cooling systems

In this section exergy analysis of conventional desiccant cooling systems in general, where the necessary heat is provided by a boiler, is considered. The special case of solar-driven desiccant cooling, more relevant for the purpose of this paper, is considered in the following Section 4.3.1.

Fig. 9 shows a typical layout for desiccant cooling systems (DEC), where the energy and exergy inputs into the system can be clearly recognized. Energy and exergy inputs into the desiccant cooling air handling unit are the energy and exergy demand required for the operation of the chiller, E_{chill} and Ex_{chill}, the energy and exergy input into the heat generation system for the regenerator (e.g. boiler or heater), $E_{heat,gen}$ and $Ex_{heat,gen}$, energy and exergy required for the operation of the fans, W_{circ}, energy and exergy input into the evaporative cooling coils, $m_{eva\,p}e_w$ and $m_{eva\,p}ex_w$, and energy and exergy associated to the return air (bottom channel on the schema in Fig. 9), $m_{ret}\,e_{ret}$ and $m_{ret}\,ex_{ret}$. In $E_{heat,gen}$, and E_{chill}, the efficiency and *COP* of the chiller and heat generation system is already included. Energy and exergy desired outputs for the *AHU* are those associated to the supply air at the supply conditions as it reaches indoor environment, $m_{sup}\,e_{sup}$ and $m_{sup}\,ex_{sup}$ (Eq. (52)). In turn, energy and exergy output associated to the overall system are the cooling energy and exergy demands of the building (numerator in Eq. (54)). In Eqs. (53) and (54)

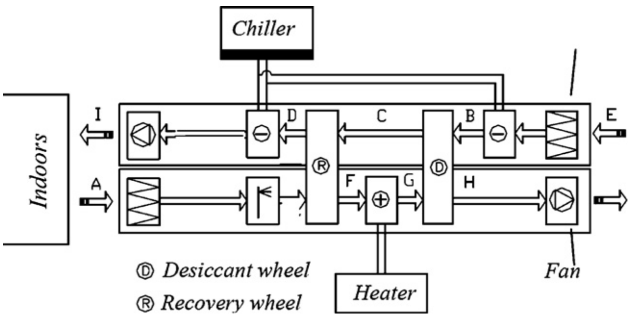

Fig. 9. Typical layout for desiccant cooling systems [37].

expressions for the single exergy efficiency of the *AHU* and overall exergy efficiency of the DEC system are shown.

According to the diagram, the exergy of the return airflow entering the *AHU* unit, ex_{ret}, would be equal to that of the air at the conditions in point *A*, and that of the supply air, $ex_{sup,}$ would be associated to the air at the conditions *I*:

$$Ex_{in,ove} = \frac{W_{circ}}{\eta_{elec}} + Ex_{chill} + Ex_{heat,gen} + m_{ret}ex_{ret} + m_{eva\,p}ex_w \quad (50)$$

$$Ex_{in,AHU} = W_{circ} + Ex_{cc} + Ex_{hc} + m_{ret}ex_{ret} + m_{eva\,p}ex_w \quad (51)$$

$$Ex_{out,AHU} = m_{\mathrm{sup}}ex_{\mathrm{sup}} \quad (52)$$

$$\Psi_{sngle,AHU} = \frac{Ex_{out,AHU}}{Ex_{in,AHU}} \quad (53)$$

$$\Psi_{ove} = \frac{Q_{cool}\left(1 - \frac{T_0}{T_{room}}\right)}{Ex_{in,ove}} \quad (54)$$

Marletta [37] performs a simplified steady state exergy analysis of a conventional air conditioning unit, including cooling and re-heating coils, and a desiccant-cooling system. In order to make the systems comparable, the same mass flows, inlet (outdoor ambient air) and supply air conditions are regarded in both cases. The reference state is defined as air at 35 °C with a humidity content of 21.4 g/kg. Under these conditions, the exergy efficiency of the *AHU* is 21.5%. Major causes for irreversibilities in the *AHU* are the processes occurring in the cooling coil, namely cooling and dehumidification of supply air. Irreversibilities here arise mainly due to relatively high temperature difference between the water and air sides of the cooling coil (water supply temperature of 8 °C and air inlet temperature of 35 °C were regarded). A summary of the main results is presented in Table 11.

In turn, using DEC systems which make advantage of chemical dehumidification, higher cooling coil temperatures can be allowed (15 °C are regarded in the example), significantly reducing irreversibilities in this component (see Table 11). Yet, DEC systems have some adiabatic but irreversible components, i.e. regenerator, desiccant wheel and evaporative cooler. Therefore, exergy efficiency of *AHU* with the DEC system is lower than that of the conventional air conditioning system, 13–19% depending on the regarded regeneration temperature chosen. These results are coherent with those found in Ref. [89]. Dincer and Rosen [89] perform an exergy analysis on an open-cycle air desiccant cooling system, using design conditions from an experimental desiccant cooling system built at the University of Gaziantep, Turkey. They find that the highest exergy losses occur in the desiccant wheel. Even if an infinitely large heating source at the output regenerator temperature is considered as source for the regeneration heat, large inherent irreversibilities are linked to the heating process. Evaporative cooling processes are also of importance for the performance of the whole cooling system, as can be seen in Table 11. Yet irreversibilities associated to them are inherent to evaporative cooling processes and, thus, there is not a great potential to reduce them.

Looking at the results from Refs. [37] and [89], it may be concluded that irreversibilities in the heating coils and regenerators of DEC systems represent a significant part of their overall irreversibilities (25–31%). These irreversibilities are mainly due to the temperature difference between the air and water side of the heating coil and regenerator heat exchanger. Similar conclusions can also be derived for the desiccant wheel. Therefore, significant improvements could be achieved in the performance of the

Table 11
Comparison of exergy analysis of different DEC and conventional air conditioning systems.

System	Location	Outdoor air (reference); T_0 (C) ω_0 (/kg)	Indoor air T_0 (°C); ω_0 (g/kg)	Heating coil T_{supply} (°C)	Cooling coil T_{supply} (°C)	ψ_{ove}[a] (%)	Irreversibility list and magnitude	Ref.
AHU + chiller	–	35; 21.4	26; 10.5	60	8	4.5	Cooling coil (59%) Condensate (24%) Heating coil (18%)	[37]
DEC	–	35; 21.4	26; 10.5	80	15	3.13	Heating coil (26%) Regenerator (25%) Cooling coils (21%) Desiccant wheel (21%) Condensate (4%) Evaporative cooler (3%)	[37]
DEC	Turkey	31.5; 9.5	17.3; 11.6	–	–	–	Desiccant wheel (34%) Heating coil (31%) Regenerator (18%) Evaporative cooling coils (17%)	[89]

[a] The authors did not include the efficiency of the electricity conversion in the definition of the overall efficiency.

systems if the heating and regeneration temperatures are further lowered.

In Fig. 10 the percentage of irreversibilities for each component of the DEC systems analysed in Ref. [37] are shown for different regeneration temperatures. It is shown that for lower regeneration temperatures, irreversibilities in the dessicant wheel and the heating coil are reduced. However, irreversibilities increase for the pre- and post-cooling processes, as lower temperatures in the supply airflow are required to overcome the losses in the effectiveness of the dessicant wheel at lower regeneration temperatures. Yet, absolute values of the overall irreversibilities are lower if lower regeneration temperatures are used, being 6.85, 5.4 and 4.3 kW for regeneration temperatures of 90, 80 and 70 °C, respectively. Subsequently, lowering regeneration temperatures for the desiccant cooling system increases its exergy efficiency. Similar trend can be found in terms of the specific primary energy consumption, which is reduced for lower regeneration temperatures.

In Table 11 the overall efficiencies for the DEC and conventional air conditioning systems are shown. However, the authors of these papers disregarded the efficiency for the electricity conversion, η_{electr}, i.e. electricity is considered as a direct input into the cooling systems. Electricity is not an energy source, but an energy carrier. Therefore, when making an analysis on a general system level, the efficiency of its conversion from other energy sources should also be taken into account, η_{electr}, i.e. the respective primary energy factor for the electricity should be regarded. Thus, in the case of an electrical compression chiller or an electrical boiler, the exergy input into the chiller and boiler, respectively, should also be divided by the efficiency of the electricity conversion, η_{electr}, i.e. the respective primary energy factor for the electricity.

Alpuche et al. [20] perform a dynamic exergy analysis of open cooling system for hot humid climatic conditions corresponding to Tabasco (Mexico). The systems investigated are a conventional compression chiller for air conditioning, conventional air conditioning system with dehumidification and a DEC system. All systems have an air mass flow of 0.850 m^3/s and are examined and compared regarding their exergy efficiency and the number of hours that indoor comfort conditions are fulfilled.

It is important to notice that the authors include the exergy flowing outside the building envelope due to exfiltration, $ex_{m,out}$, in the definition of the exergy efficiency. This is equivalent as applying the "simple" exergy efficiency to evaluate the performance of the cooling system (see Section 2.3). The vapour compression conventional cooling system provided the smallest number of hours under comfort conditions (3850 h), and also shows the smallest exergy efficiency (2.5%). Adding a dehumidifier increases the number of hours with comfort conditions fulfilled

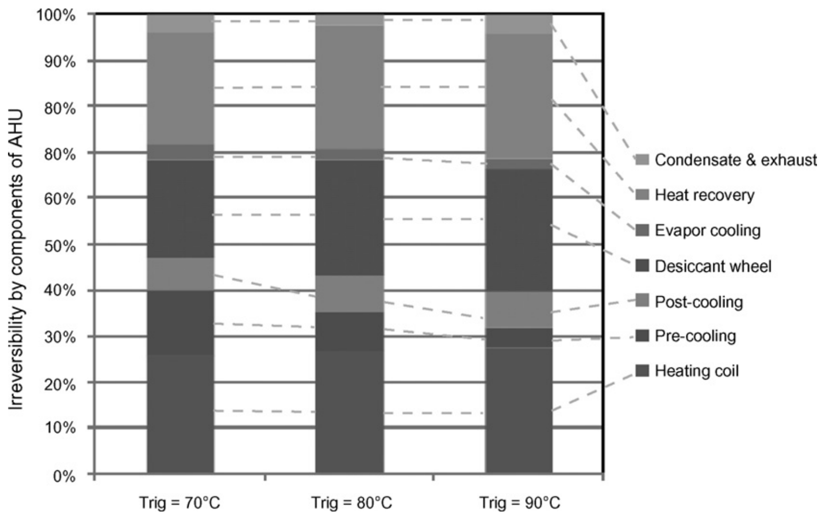

Fig. 10. Percentage of the total irreversibilities in the different components of the *AHU* of the DEC system for different regeneration temperatures, T_{rig} [37], modified.

(4519 h), but at the expense of reducing the exergy efficiency of the system (1.8%). In turn, the use of an open DEC system would almost double comfort hours (7687 h) and allows significant increases in the exergy efficiency of the cooling system (6.3%).

The big difference in the efficiency of the DEC and conventional cooling units seems uncoherent with results from authors reviewed in Table 11. Therefore, for clarification some remarks need to be done:

- The systems in Marletta [37] provide the same conditions for the supply air, i.e. the same comfort range is achieved, making the systems directly comparable to each other. In turn, in Ref. [20] different ranges of humidity and temperature are supplied, thus making operative conditions of the system differ from each other and significantly difficulting the comparison among the systems;
- In Ref. [20] the simple exergy efficiency (instead of the rational efficiency) is applied, and subsequently the exergy due to exfiltration is considered as part of the "desired output" in the overall exergy definition used. The exergy of this mass flow is higher, the more different the indoor conditions differ from outdoor reference conditions, i.e. the higher the number of comfort hours. Subsequently, regarding this term as desired output causes a more significant increase in the overall exergy efficiency for the DEC system than for the conventional one;
- Alpuche et al. [20] included the efficiency for the electricity conversion, $\eta_{electr} = 0.48$, whereas, as already pointed out, Marletta [37] and Dincer and Rosen [89] did not. Assuming the same efficiency for electricity conversion overall efficiencies for the conventional and DEC systems in Ref. [37] would be 2.3% and 2.6%, respectively, and thereby the exergy performance of the DEC system would be similar to that of the conventional cooling unit.

A comparison of the systems is extremely difficult, for their operative conditions are not the same. However, it may be inferred that the strong difference between the exergy performances of DEC systems and conventional units found in Ref. [20] may be mainly due to the use of the overall simple exergy efficiency, i.e. including the exfiltration losses as desired output in the overall (simple) exergy efficiency. Further cause for the different results obtained from the comparison of the DEC and conventional *AHU* systems in Ref. [20] and [37] might be the choice of a dynamic analysis [20] or a steady-state approach [37]. This highlights again the strong necessity of a common method for exergy analysis and exergy performance definition.

4.3.1. Solar-driven desiccant systems

The overall efficiency for solar-driven DEC systems is defined similarly as for conventional DEC units, but including the exergy output from the solar thermal systems as a further input into the overall cooling system.

Marletta [37] and [90] carried out an analysis of solar coupled DEC systems. The exergy input from the solar thermal system is regarded following a similar approach as the "physical boundary" (see Section 2.7). However, instead of regarding the exergy of the heat output from the collector field, the incident solar radiation onto collector surface is regarded as heat at the maximum temperature in the collectors. This is shown analytically in the following equation:

$$Ex_{coll} = \frac{Q_{coll}}{\eta_{coll}}\left(1 - \frac{T_0}{T_{max,coll}}\right) \quad (55)$$

As already mentioned in Section 4.3, according to Marletta [37], overall exergy efficiency of the conventional air conditioning *AHU*

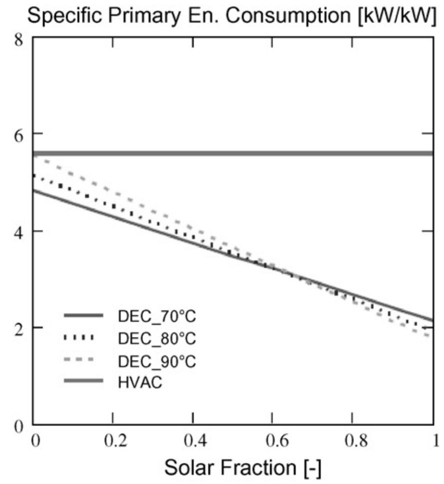

Fig. 11. Specific primary energy consumption for a conventional compression cooling unit and desiccant systems with different regeneration temperatures and solar fractions [37].

including the boiler and the compression chiller required to supply the heating and cooling water to the coils, is 4.5%. This results from the combined single efficiency of the *AHU* (21.5%) and that of the boiler and chiller (20.8%), so-called "sources efficiency" by the author. The "sources efficiency" could easily be increased by using environmental or low temperature heat instead of fossil-fuels or electricity for feeding the heating coil. For this aim, the use of solar systems coupled with DEC is an adequate alternative. The author showed that increasing the solar fraction allows obtaining overall exergy efficiencies of up to 8% (see Fig. 12).

Furthermore, in Ref. [37] the conventional air conditioning system presented in Section 4.3 is also compared to a DEC system with different solar fractions. The specific primary energy consumption (SPEC) and exergy efficiency were the parameters chosen for the comparison of the systems. From an energy perspective desiccant open systems have a good performance, so that relatively low solar fractions (below 10%) would be sufficient to be advantageous against a conventional air-conditioner. Similar results can also be found in Ref. [90].

In Fig. 11 the efficiency for the electricity conversion, η_{electr}, is regarded as 0.37. In turn, in Fig. 12 electricity is regarded as a direct

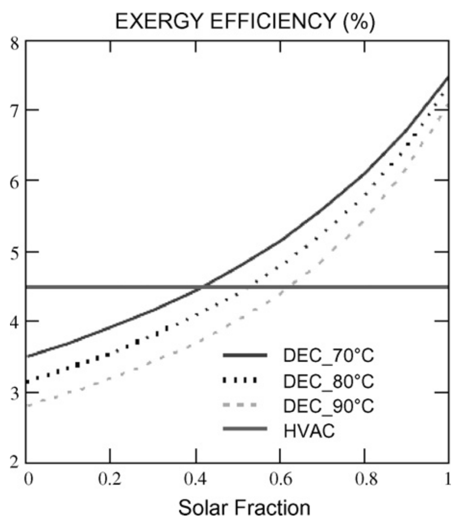

Fig. 12. Exergy efficiency for a conventional compression cooling unit and desiccant systems with different regeneration temperatures and solar fraction [37].

input into the building systems. Thus, different boundaries for the energy and exergy analysis were chosen, making significantly difficult the comparison of the results from energy and exergy analysis. If the same efficiency for electricity generation would be regarded, the exergy efficiency of the conventional system would be 1.8% rather than 4.5% and that of a DEC system without solar fraction and with regeneration temperature of 80 °C would be 2.3% rather than 3.1%. Exergy efficiencies for the DEC systems are also reduced, but the effect on these units is lower as for the conventional unit, for lower electrical energy for the chiller is required if a desiccant unit is used. Therefore, regarding the same boundaries for the SPEC and overall exergy efficiency of the unit, i.e. including in both cases the efficiency for the electricity conversion, $\eta_{electr,}$ similar conclusions could be withdrawn from energy and exergy analysis.

However, as discussed in Section 2.5, the exergy performance of cooling units is very sensitive to varying outdoor conditions and also to the efficiency of the electricity conversion regarded. Thus, for a detailed comparison of the performance of the systems further dynamic analysis of the units would be desirable.

5. Conclusions

From the critical comparison among methodologies and results of the reviewed studies applying exergy analysis to heating and cooling systems based on renewable energy sources, the following concluding remarks may be derived:

- Some authors have pointed out that results from exergy analyses of climatisation systems may be more sensitive to the reference environment than for other energy conversion systems, especially when and where the indoor conditions are near to the reference ones. Nevertheless, a general agreement on the proper choice of the dead state is not found in the reviewed literature. Actually, most of the papers analysed perform exergy analysis following a steady-state approach. Moreover, some of them assume standard state values for the reference temperature, some others adopt a local design value, others again a local value representative of the period covered by the analysis. Some other authors have found that a careful choice of the properties defining the reference environment under steady state analysis would lead to small differences in the results compared with dynamic approach, avoiding thus the necessity to face time consuming analyses. However, which statistical value (e.g. mean, mode, median, etc.) for the properties of the reference would lead always to the best agreement with dynamic analysis over the same period of time has neither been identified. Regarding the humidity content of the dead state, it is found that most papers disregard the chemical exergy of the outdoor air. This assumption may lead to relevant inaccuracies in warm and humid climates. In those studies where chemical exergy of the air is taken into account (e.g. studies on evaporative cooling processes), two opposite approaches have been highlighted: assuming a reference humidity ratio equal to the actual outdoor humidity value or equal to saturated air.
- It is clear that all these different methodological approaches have an influence on the results achieved and tend to complicate the possibility to compare them and to derive shared assessments on systems and components. More investigations are then necessary to further clarify the limits of the steady state approach and the benefits of dynamic calculations, as well as the more suitable choice of the reference environment (from a temperature and humidity point of view) in both cases.
- Most of the papers devoted to exergy analysis reviewed here use the so-called rational exergy efficiency for characterising the performance of the energy systems analysed. Yet, in some of them simple exergy efficiencies can be found. In many cases the exergy efficiency definition chosen by the authors can be deduced from the analytical expression of the exergy efficiency. However, the choice between simple or universal and rational or functional exergy efficiencies is often not clearly stated by the authors. Agreement on a common applicability framework for these different definitions of the exergy efficiency would be of great interest, and significantly ease comparisons among results from different analysis.
- Exergy analysis can be used to optimise the performance of energy systems both on a component or system levels. It has been shown that the optimization of the most irreversible process of a component does not necessarily lead to the optimisation of the system, i.e. the exergy losses in the whole system are not minimized or reduced and might even increase. Therefore, exergy is a more powerful and appropriate tool for optimising the performance of system or component when performed from a holistic view, regarding the influence of improving or changing one process or component on the overall performance of the energy system regarded. For this aim, parametric studies have proven to be of great interest. Furthermore, a holistic methodology for exergy analysis regarding this interdependence of the energy processes in an energy system has been reported. By classifying the exergy losses into endogenous/exogenous and avoidable/unavoidable, deeper insight can be gained on the interdependence on the energy processes involved and best choices for reducing exergy losses in each component and the energy system as a whole can be identified. Although the method is significantly more complex than conventional exergy analysis, its application to different energy systems would allow to gain further insight on their real optmisation potential. Thus, further research and studies on this direction would be of great interest.
- Exergy analysis in the papers reviewed is performed both on a component level (e.g. solar collector or absorption cooling machines) and system level (energy systems for space heating and cooling applications). However, often the boundaries for exergy analysis are not clearly stated by the authors. Ambiguities on the definition of the boundaries are typically found both at the primary energy conversion step and at the heat exchange process within the building. Indeed, electricity might be regarded as an energy carrier (following an end-energy approach) or the energy efficiency for its conversion from energy sources might be regarded (following a primary-energy approach). Final output might be regarded as the output from the emission systems or the indoor air energy and exergy demands. The choice of the boundaries for performing exergy analysis might significantly influence results and insights gained from it. This is obviously true also for energy analysis, but it is even more for an exergy analysis, due to the fact that exergy destruction would happen at each step of the energy chain even if energy losses were ideally zero. Therefore, the boundaries chosen should always be clearly stated. Particularly, for the analysis of direct solar systems, the choice of the boundaries for exergy analysis (physical or technical approaches) would greatly influence the results and conclusions obtained. The approach followed should also be always clearly stated. Furthermore, it would be very useful to find a common framework for analysis of those systems which may depend on the purpose of the analysis.
- A common agreement on the methodologies for the exergy analysis of renewable energy-based climatisation systems is also the mandatory condition for any proposal dealing with the application of exergy indicators in a normative framework. Only a few proposals in this regard have been found, although it is

expected that further efforts will be carried out in order to include exergy into national or local energy regulations for the building stock.

- Exergy analysis should always come in parallel with the energy analysis. The energy performance of the analysed systems is not always clearly stated, resulting in partial and often not entirely understandable results. On the other side, whenever energy and exergy analyses bring to different or even opposite conclusions, it is not always well stated how the authors would combine the different indications derived from the two approaches. This lack of final synthesis may represent a limit to the widespread of the exergy analysis and to the comprehension of its relevance. In this sense, it would be of great interest if the usefulness of the obtained results would be more evident.
- Renewable sources are not necessarily low exergy sources. Among the renewables analysed here, solar energy should be considered as a high exergy source following the technical boundary approach or a low exergy source if a physical boundary is adopted. Biomass is comparable to fossil fuels and then it should be classified as high exergy. In turn ground and similar natural sinks that might be used as reservoirs in heat pumps may be considered as low exergy. It is an open question whether it is more important to save primary energy (that means using as much as possible renewable sources) or to save primary exergy (that means using renewables and non-renewables in the most efficient way). Thus, further debate on which should be the final objective of a combined energy and exergy analysis of climatisation systems should be necessary.

As a final conclusion, the authors expect that the critical view reported here would be useful for the development of further studies using the exergy approach as one of the tools to investigate the most rational use of energy sources within the built environment.

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References

[1] IEA ECBCS Annex 37. Low Exergy Systems for Heating and Cooling, www.ecbcs.org/annexes/annex37.htm.
[2] IEA ECBCS Annex 49. Low Exergy Systems for High-Performance Buildings and Communities, www.annex49.com.
[3] COSTeXergy Analysis and Design of Innovative Systems for Low-EXergy in the Built Environment, www.costexergy.eu.
[4] F. Calise, A. Palombo, L. Vanoli, Design and partial load exergy analysis of hybrid SOFC-GT power plant, Journal of Power Sources 158 (2006) 225–244.
[5] A. Corrado, P. Fiorini, E. Sciubba, Environmental assessment and extended exergy analysis of a "zero CO_2 emission", high-efficiency steam power plant, Energy 31 (2006) 3186–3198.
[6] A. Dagdas, R. Ozturk, S. Bekdemir, Thermodynamic evaluation of Denizli Kizildere geothermal power plant and its performance improvements, Energy Conversion and Management 46 (2005) 245–256.
[7] S. Douvartzides, F. Coutelieris, P. Tsiakaras, Exergy analysis of a solid oxide fuel cell power plant fed by either ethanol or methane, Journal of Power Sources 131 (2004) 224–230.
[8] M. Kanoglu, A. Bolatturk, Performance and parametric investigation of a binary geothermal power plant by exergy, Renewable Energy 33 (2008) 2366–2374.
[9] M. Kanoglu, I. Dincer, M.A. Rosen, Understanding energy and exergy efficiencies for improved energy management in power plants, Energy Policy 35 (2007) 3967–3978.
[10] M. Kopac, A. Hilalci, Effect of ambient temperature on the efficiency of the regeneration and reheat Catalagzi power plant in Turkey, Applied Thermal Engineering 27 (2007) 1377–1385.
[11] M.A. Rosen, Energy and exergy-based comparison of coal-fired and nuclear steam power plants, Exergy, An International Journal 1 (3) (2001) 180–192.
[12] M.A. Rosen, R. Tang, Improving steam power plant efficiency through exergy analysis: effects of altering excess combustion air and stack-gas temperature, International Journal of Exergy 5 (1) (2008) 35–51.
[13] G.P. Verkhivker, B.V. Kosoy, On the exergy analysis of power plants, Energy Conversion and Management 42 (2001) 2053–2059.
[14] M.J. Moran, H.N. Shapiro, Fundamentals of Engineering Thermodynamics, 3rd ed., John Wiley & Sons, New York, USA, 1998.
[15] A. Bejan, G. Tsatsaronis, M. Moran, Thermal Design and Optimization, John Wiley and Sons, New York, USA, 1996.
[16] A. Bejan, E. Mamut (Eds.), Thermodynamic Optimization of Complex Energy Systems, Kluwer Academic Publishers, The Netherlands, 1999, pp. 73–92.
[17] A. Bejan, Advanced Engineering Thermodynamics, John Wiley and Sons, New York, USA, 1997.
[18] M. Shukuya, A. Hammache, Introduction to the Concept of Exergy—For a Better Understanding of Low-temperature-heating and High-temperature-cooling Systems, VTT research notes 2158, Espoo, Finland, 2002.
[19] M.A. Rosen, I. Dincer, Effect of varying dead-state properties on energy and exergy analyses of thermal systems, International Journal of Thermal Sciences 43 (2004) 121–133.
[20] M.G. Alpuche, et al., Exergy analysis of air cooling systems in buildings in hot humid climates, Applied Thermal Engineering 25 (4) (2005) 507–517.
[21] T.T. Chow, et al., Energy and exergy analysis of photovoltaic-thermal collector with and without glass cover, Applied Energy 86 (2009) 310–316.
[22] A. Angelotti, P. Caputo, The exergy approach for the evaluation of heating and cooling technologies; first results comparing steady-state and dynamic simulations, in: Proceedings of the 2nd PALENC and 28th AIVC Conference, vol. I, Crete Island, Greece, (2007), pp. 59–64.
[23] H. Torio, D. Schmidt, Exergetic assessment and contribution of solar energy systems to the energy performance of buildings, in: Proceedings of the Nordic Symposium of Building Physics 2008, vol. 2, Copenhaguen, Denmark, (2008), pp. 637–644.
[24] P. Sakulpipatsin, Exergy Efficient Building Design, PhD Thesis, Department Building Technology, University of Delft (the Netherlands) (2008).
[25] R. Chengquin, L. Nianping, T. Guanfa, Principles of exergy analysis in HVAC and evaluation of evaporative cooling schemes, Building and Environment 37 (2002) 1045–1055.
[26] W.J. Wepfer, R.A. Gaggioli, E.F. Obert, Proper evaluation of available energy for HVAC, ASHRAE Transactions 85 (1) (1979) 214–230.
[27] ISO 13790: 2008, Thermal performance of buildings. Calculation of energy use for space heating and cooling (2008).
[28] DIN 4701-10: 2001, Energy Efficiency of Heating and Ventilation Systems in Buildings. Part 10: Heating, Domestic hot Water, Ventilation. German National Standard. Berlin: Deutsches Institut für Normung e.V. (2001).
[29] EnEV:2007, Verordnung über energiesparenden Wärmeschutz und energiesparende Anlagentechnik bei Gebäuden (Energieeinsparverordnung–EnEV) (2007).
[30] DIN V 18599:2007. Energy efficiency of buildings–Calculation of the net, final and primary energy demand for heating, cooling, ventilation, domestic hot water and lighting. German National Standard. Berlin: Deutsches Institut für Normung e.V. (2007).
[31] D. Schmidt, Design of low exergy buildings—method and pre-design tool, The International Journal of Low Energy and Sustainable Buildings 3 (2004) 1–47.
[32] D. Favrat, F. Marechal, O. Epelly, The challenge of introducing an exergy indicator in a local law on energy, Energy 33 (2008) 130–136.
[33] T. Morosuk, G. Tsatsaronis, A new approach to the exergy analysis of absorption refrigeration machines, Energy 33 (2008) 890–907.
[34] R.L. Cornelissen, Thermodynamics and sustainable development—the use of exergy analysis and the reduction of irreversibility. Doctoral Thesis, Technical University Twente, Enschede, the Netherlands (1997).
[35] G. Tsatsaronis, Thermoeconomic analysis and optimization of energy systems, Progress in Energy Combustion Science 19 (1993) 227–257.
[36] E.C. Boelman, P. Sakulpipatsin, Critical analysis of exergy efficiency definitions applicable to buildings and building services, in: Proceedings of the PLEA 2004, The Netherlands, 2004.
[37] L. Marletta, Exergy analysis of solar cooling—a worked example, Report for the IEA SHC Task 38, January 2008.
[38] H. Esen, et al., Energy and exergy analysis of a ground-coupled heat pump system with two horizontal ground heat exchangers, Building and Environment 42 (2007) 3606–3615.
[39] A. Hepbasli, M. Tolga Balta, A study on modelling and performance assessment of a heat pump system for utilizing low temperature geothermal resources in buildings, Building and Environment 42 (2007) 3747–3756.
[40] D. Schmidt, H. Torio, C. Sager, Exergiebewertung für Gebäude, in: Deutsche Klima-Kälte Tagung, Hannover, Germany, 2007.
[41] G. Tsatsaronis, Definitions and nomenclature in exergy analysis and exergoeconomics, Energy 32 (2007) 249–253.
[42] J. Abraham, L. Suganthi, Entropy added taxing methodology: a case study with domestic refrigerators in India, International Journal of Exergy 3 (2) (2006) 150–163.
[43] G. Hirs, Exergy loss: a basis for energy taxing, in: P.A. Pilavachi (Ed.), Energy Efficiency in Process Technology, Elsevier Applied Science, Amsterdam, 1994.

[44] R. Nishikawa, M. Shukuya, Numerical analysis on the production of cool exergy by making use of heat capacity of building envelopes, in: Proceedings of the 6th International IBPSA Conference, Kyoto, Japan, (1999), pp. 129–135.
[45] J. Szargut, The Exergy Method—Technical and Ecological Applications, Renewable Wiley Interscience, 2005.
[46] N.U. Ukidwe, B.R. Bakshi, Industrial and ecological cumulative exergy consumption of the United States via the 1997 input-output benchmark model, Energy 32 (2007) 1560–1592.
[47] J. Szargut, Anthropogenic and natural exergy losses (exergy balance of the earth's surface and atmosphere), Energy 28 (11) (2003) 1047–1054.
[48] G. Wall, M. Gong, On exergy and sustainable development. Part 1. Conditions and concepts., Exergy, An International Journal 1 (3) (2001) 128–145.
[49] G. Chen, Exergy consumption of the earth, Ecological Model 148 (2–4) (2005) 363–380.
[50] H.T. Odum, Handbook of Emergy Evaluation Folio 2: Emergy of Global Processes, Center for Environmental Policy, University of Florida, Gainesville, Florida, 2000
[51] W.A. Hermann, Quantifying global exergy resources, Energy 31 (12) (2006) 1685–1702.
[52] E. Sciubba, S. Ulgiati, Emergy and exergy analyses: complementary methods or irreducible ideological options? Energy 30 (2005) 1953–1988.
[53] B. Sørensen, Renewable Energy: Its Physics, Engineering, Use, Environmental Impacts, Economy and Planning Aspects, Third ed., Elsevier Academic Press Inc., USA, 2004.
[54] H. Torio, D. Schmidt, Framework for analysis of solar energy systems in the built environment, from an exergy perspective, Submitted for publication.
[55] S.M. Jeter, Maximum conversion efficiency for the utilization of direct solar radiation, Solar Energy 26 (1981) 231–236.
[56] R. Petela, Exergy of undiluted thermal radiation, Solar Energy 74 (2003) 469–488.
[57] B. Sandnes, Energy efficient production, storage and distribution of solar energy. PhD Thesis, University of Oslo, Norway (2003).
[58] H. Gunerhan, A. Hepbasli, Exergetic modelling and performance evaluation of solar water heating systems for building applications, Energy and Buildings 39 (5) (2007) 509–516.
[59] C. Koroneos, T. Spachos, N. Mossiopoulos, Exergy analysis of renewable energy sources, Renewable Energy 28 (2003) 295–310.
[60] A. Bejan, Extraction from exergy from solar collectors under time-varying conditions, International Journal of Heat and Fluid Flow 3 (2) (1982) 67–72.
[61] I. Luminosu, L. Fara, Determination of the optimal operation mode of a flat solar collector by exergetic analysis and numerical simulation, Energy 30 (2005) 731–747.
[62] M. Izquierdo Millán, F. Hernández, E. Martín, Available solar exergy in absorption cooling process, Solar Energy 56 (6) (1996) 505–511.
[63] W. Xiaowu, H. Ben, Exergy analysis of domestic-scale solar water heaters, Renewable and Sustainable Energy Reviews 9 (2005) 638–645.
[64] S.K. Chaturvedi, D.T. Chen, A. Kheireddine, Thermal performance of a variable capacity direct expansion solar-assisted heat pump, Energy Conversion Management 39 (3) (1998) 181–191.
[65] M. Meir, On creating thermal comfort without energy use, PhD, University of Oslo, Norway (2002).
[66] E. Torres-Reyes, M. Picon-Nunez, J. Cervantes-de Gortari, Exergy analysis and optimisation of a solar-assisted heat pump, Energy 23 (4) (1998) 337–344.
[67] O. Ozgener, A. Hepbasli, Modeling and performance evaluation of a ground source (geothermal) heat pump system, Energy and Building 39 (2007) 66–75.
[68] A. Dikici, A. Akbulut, Exergetic performance of heat pump systems having various heat sources, International Journal of Energy Research (2008) DOI: 10.1002/er.
[69] P.G. Charalambous, et al., Photovoltaic thermal (PV/T) collectors: a review, Applied Thermal Engineering 27 (2007) 275–586.
[70] J.S. Coventry, K. Lovegrove, Development of an approach to compare the "value" of electrical and thermal output from a domestic PV/thermal system, Solar Energy 75 (2003) 63–72.
[71] H. Saitoh, et al., Field experiments and analyses on a hybrid solar collector, Applied Thermal Engineering 23 (2003) 2089–2105.
[72] T. Fujisawa, T. Tani, Annual exergy evaluation on photovoltaic-thermal hybrid collector, Solar Energy Materials and Solar Cells 47 (1997) 135–148.
[73] M. Bosanac et al., Photovoltaic/Thermal solar collectors and their potential in Denmark, Final Report EFP Project 1713/00-0014, 2003.
[74] A. Hepbasli, O. Akdemir, Energy and exergy analysis of a ground source heat pump system, Energy Conversion and Management 45 (2004) 737–753.
[75] E.K. Akpinar, A. Hepbasli, A comparative study on exergetic assessment of two ground-source (geothermal) heat pump systems for residential applications, Building and Environment 42 (2007) 2004–2013.
[76] H. Torio, Comparison of ventilation and solar thermal systems for residential buildings on the basis of their energy and exergy flows, Master Thesis, Ossietzky University Oldenburg (Germany) (2007).
[77] J.G. Cervantes, E. Torres-Reyes, Experiments on a solar-assisted heat pump and an exergy analysis of the system, Applied Thermal Engineering 22 (2002) 1289–1297.
[78] DIN V 18599:2007, Energy efficiency of buildings–Calculation of the net, final and primary energy demand for heating, cooling, ventilation, domestic hot water and lighting, Part 1. General balancing procedures, terms and definitions, zoning and evaluation of energy sources, German National Standard. Berlin: Deutsches Institut für Normung e.V.
[79] N. Diefenbach, et al. (Eds.), Wärmeversorgung für Niedrigenergiehäuser–Erfahrungen und Perspektiven, Institut für Wohnen und Umwelt (IWU), Darmstadt (Germany), 2005.
[80] D. Schmidt, Benchmarking of "LowEx" buildings, in: Proceedings of the Nordic Symposium of Building Physics 2008, vol. 2, Copenhaguen, Denmark, (2008), pp. 621–628.
[81] Institut Wohnen und Umwelt (IWU) (Eds.), Deutsche Gebäudetypologie–Systematik und Datensätze. Darmstadt, Germany, December 2003.
[82] M. Pons, et al., Thermodynamic based comparison of sorption systems for cooling and heat pumping, International Journal of Refrigeration 22 (1999) 5–17.
[83] A. Sencan, A.K.A. Yakut, S.A. Kalogirou, Exergy analysis of lithium bromide/water absorption systems, Renewable Energy 30 (2005) 645–657.
[84] D. Boer, et al., Exergy analysis of a solar absorption cooling system, in: Proceedings of the 2nd OTTI International Conference in Solar Air-Conditioning, Tarragona, Spain, 2007.
[85] A. Khaliq, R. Kumar, Exergy analysis of solar powered absorption refrigeration system using LiBr–H_2O and NH_3–H_2O as working fluids, International Journal of Exergy 4 (1) (2007) 1–18.
[86] T.S. Ravikumar, L. Suganthi, A.A. Samuel, Exergy analysis of solar assisted double effect absorption refrigeration system, Renewable Energy 14 (1–4) (1998) 55–59.
[87] M. Izquierdo, et al., Compressors driven by thermal solar energy: entropy generated, exergy destroyed and exergetic efficiency, Solar Energy 72 (4) (2002) 363–375.
[88] N.B. Ezzine, et al., Solar cooling with the absorption principle: First and Second Law analysis of an ammonia–water double generator absorption chiller, Desalination 168 (2004) 137–144.
[89] I. Dincer, M. Rosen, Exergy—Energy, Environment and Sustainable Development, First ed., Elsevier Publication, Oxford, UK, 2007.
[90] L. Marletta, G. Evola, F. Sciurella, Second law analysis of solar-assisted desiccant open cycles: a comparative analysis, in: Proceedings of the 2nd OTTI International Conference in Solar Air-Conditioning, Tarragona, Spain, 2007.
[91] H.M. Henning (Ed.), Solar-Assisted Air-conditioning in Buildings—A Handbook for Planners, Springer-Verlag, Vienna, New York, 2003 (printed in Vienna).