SHC 2015, International Conference on Solar Heating and Cooling for Buildings and Industry

Cost-optimal sizing of solar thermal and photovoltaic systems for the heating and cooling needs of a nearly Zero-Energy Building: design methodology and model description

Daniele Testi, Eva Schito, Paolo Conti

BETTeR (Building Energy Technique and Technology Research group), University of Pisa – DESTEC, Largo Lucio Lazzarino, 56122 Pisa, Italy

Abstract

This paper deals with the cost-optimal sizing of solar technologies for thermal and electrical needs of residential or tertiary nearly Zero-Energy buildings. The proposed design procedure is based on lifetime simulation of building loads and energy systems; therefore, according to proper cost-optimality considerations, it is possible to find the best sizing of both heat and electricity generators in the context of high-efficiency buildings (e.g. number of solar thermal and PV modules). The paper is divided in two parts. In this first part, we describe general features and principles of the methodology, together with the physical models of building-plant system. Building requirements of thermal and electrical energy are evaluated according to internal loads and external climate, while energy system operation is simulated by a full set of equations reproducing the coupled behavior of each piece of equipment. A preliminary application example referring to a nearly Zero-Energy Building is also illustrated: In the second part of the work, we will apply and discuss the overall simulation-based optimization procedure. Results show the notable benefits of the proposed design approach with respect to traditional ones, in terms of both energy and economic savings. Besides, the proposed methodology can be successfully applied in the more general framework of Net Zero Energy Buildings (NZEBs) in order to fulfill recent regulatory restrictions and objectives in building energy performances.

Keywords: Nearly Zero-Energy Buildings; cost-optimal design; solar thermal; photovoltaics; heat pumps; design method

1. Introduction

The design of building energy systems aims to figure out the best technological solution to match energy demand for services. A universal straightforward design procedure does not exist as any specific project has particular characteristics and objectives [1].

doi:10.1016/j.egypro.2016.06.191

Nomenclature

Acronyms

B.O.S.	balance of system
DHW	domestic hot water
H/C	heating and cooling system
HP	heat pump
PV	photovoltaic system
RF	radiant floor
ST	solar thermal system
TS	thermal storage

Symbols

C	global cost
C_0	installation cost
C_{dehum}	coil characteristic coefficient
COP	actual coefficients of performance in heating mode
COP_{id}	maximum theoretical COP in heating mode (i.e. Carnot efficiency)
E	energy
EER	actual coefficients of performance in cooling mode
EER_{id}	maximum theoretical EER in cooling mode (i.e. Carnot efficiency)
F_R	ST removal factor
H_{ve}	equivalent ventilation-thermal transmittance
I_{sol}	global irradiance at a given orientation
$I_{sol,o}$	extra-terrestrial global irradiance on the horizontal surface
K_{RF}	RF thermal output per surface unit
K_t	hourly clearness index
$NOCT$	nominal operating cell temperature
P	power
S	surface
T_{aqu}	aqueduct temperature
T_{DHW}	DHW delivery temperature
$T_{eva/cond}$	effective heat exchange temperature in HP evaporator or condenser
T_{ext}	outdoor temperature
T_{off}	switching-off temperature
T_{PV}	PV modules temperature
T_{TS}	thermal storage temperature
U_L	ST frontal losses coefficient
U_{wf}	water-floor thermal transmittance
UA	heat transmittance-surface product
V	volume

b_0	incidence angle modifier coefficient for single-cover ST collectors
c	specific heat capacity
c_0	unitary installation cost
f_p	primary energy factor
n	number of PV modules or ST collectors
n_{air}	air changes per hour
n_{RF}	emitter exponent of the radiant floor
s	thickness
x	humidity ratio

Greek letters

$\beta_{T,PV}$	PV penalization factor depending on PV technology
η	efficiency
ϑ	angle between the beam radiation and the normal to the ST collectors
λ	thermal conductivity
ρ	density
$(\tau\alpha)_n$	transmittance-absorptance product for normal-incidence irradiance
ϕ	time shift

Superscript

II	second-law parameter
$'$	next time step
$*$	sol-air temperature
TOT	cumulative value at the end of project lifetime

Subscript

CK	cooking service
LGT	lighting
OU	electric uses (household appliances, office devices)
dehum	dehumidification
des	design condition
el	electrical
grid	electrical grid
in	inlet conditions
inv	electronic converter (i.e. B.O.S.)
ls	losses
prod	production
ref	reference conditions
th	thermal
w	water
z	indoor

Traditional engineering design process is based on a sequential approach, starting from the definition of the project objectives, the choice of a main technological solution to address the set goals, the identification of technical and economic constraints, and the application of the "precautionary principle", to ensure that the project specifications are met. The latter point is commonly obtained by oversizing the main equipment, on the basis of the worse operative situation, and the installation of additional back-up devices. Consequentially, both technical and economic inefficiencies occur, resulting in an increasing of the energy consumption, operational costs and environmental impact.

In this regard, the ever-increasing demand of high-efficiency solutions, low production cost and strict environmental regulation have prompted operators to search for alternative sizing and control approaches. In addition, we note that modern systems are generally made of coupled subsystems, in which different technologies (e.g. renewable energy systems and traditional generators) cooperate for concurrent objectives. Therefore, modern design approaches shall not focus only on components sizing, but they have to seek the optimal size and management strategy of all different subsystems in order to maximize the synergy among reciprocal interactions.

To achieve these goals, rigorous techniques of decision making have to be employed, such as optimization methods. In particular, we refer to holistic approaches based on a simulation of the overall equipment, properly modeling all the involved physical mechanisms and including mutual interactions among different components. This approach is named *"simulation-based optimization procedure"* and it has been largely applied in building energetic studies in recent decades [2,3].

With regard to solar technologies, the cost-optimal [4] integration of thermal and electric technologies for matching the energy needs of a building is a paradigmatic case to apply the above-mentioned considerations [5–7]. Especially in Europe, where the Energy Performance of Buildings Directive [8] requires all new buildings to be nearly zero-energy by the end of 2020, advanced design methodologies that are both reliable and cost-effective are particularly needed. Among the others, simulation-based procedures are proving to be very effective and favorable for building energy systems (solar technologies included) [9–12].

The proposed methodology can be used to investigate the cost-optimal design of solar-assisted systems for heating, cooling, and domestic hot water production in free-standing residential buildings, employing nearly zero energy (NZEB). In that context, solar technologies are typically coupled with low-temperature heat terminal unit (e.g. radiant floor), thermal storage, and electrically-driven heat pump generator(s), as back-up. Additional back-up technologies can be used to cover the remaining energy requirements and modulate the power peaks.

In this work, we describe some simplified dynamic models of the above-mentioned equipment. All these expressions are included in a full set of equations, to reproduce the operative performance of the overall system. For the sake of clearness, we focus on a reference case study, though the proposed method and models can be easily extended to other similar cases. For a given building envelope, location, energy equipment characteristics, and expected occupancy schedules, the main results of the proposed routines are the dynamic and seasonal energy performance of each system component and of the whole system, together with the operative evolution of operative variables (e.g. storage temperature, PV energy production, temperature of heat terminal units).

As above-mentioned, energy simulation-based methods are favorably coupled to sensitivity analyses and optimization algorithms in order to find the best size of system components (viz. solar thermal collectors and photovoltaic modules). Besides, dynamic simulation of the integrated system can be used to determine the optimal control strategy of energy fluxes and the corresponding set-points. With reference to the described case study, optimization procedure and results will be discussed in [13].

2. System modeling

In this section, we describe the models and control strategy of a reference building loads and energy system components. We refer to a reference building (i.e. a farm hostel) equipped with PV generator, ST generator, HP unit, and thermal energy storage. Both thermal and electrical energy balances are evaluated through a discrete dynamic method over a standard year of operation. We use an hourly time step, thus, the following equations are calculated subsequently for every hour of the reference year. An application example and the main outputs of the proposed simulation method will be illustrated in section 4.

2.1. Building energy demand

Building energy demand consists in both sensible and latent loads, DHW production, thermal and electrical energy for cooking, electrical energy for lighting, electrical energy for other uses (food refrigeration, household appliances, and office devices).

2.1.1. Heating sensible load

Hourly heating load depends on outdoor temperature, geometry, and building thermo-physical properties. In this work, we evaluate $E_{th,H}$ as:

$$E_{th,H} = P_{th,des}\left(1 - \frac{\bar{T}_{ext} - T_{des}}{T_{off,H} - T_{des}}\right) \qquad [kWh] \tag{1}$$

$T_{off,H}$ is the nominal temperature at which building gains and losses are balanced and the heating system is subsequently switched off. $\bar{T}_{ext}$ is an effective outdoor temperature based on the actual evolution of T_{ext} and building properties. It is defined as the simple moving average of the previous $\bar{\phi}$ values of T_{ext}, being $\bar{\phi}$ the nearest integer of the effective time shift of the building. The latter value is evaluated as:

$$\bar{\phi} = \sum_i (UA)_i \phi_i \left/ \left[\sum_i (UA)_i + H_{ve}\right]\right. \qquad [h] \tag{2}$$

where $(UA)_i$ and ϕ_i are the surface-thermal transmittance product and the characteristic time shift (as defined by EN ISO 13786:2007 [14]) of the ith external wall, respectively. H_{ve} is the equivalent ventilation-thermal transmittance depending on air change rate. This model aims at considering the delay of the building thermal load with respect to the outdoor temperature evolution due to the envelope heat capacity.

2.1.2. Cooling sensible load

Similarly to $E_{th,H}$, hourly cooling load is evaluated by considering the effects of the heat capacity of building. Furthermore, solar irradiation is included through the so-called "sol-air temperature", T^* [15].

$$E_{th,C} = P_{th,des,C}\left(1 - \frac{\bar{T}^*_{ext} - T^*_{des,C}}{T_{off,C} - T^*_{des,C}}\right) \qquad [kWh] \tag{3}$$

where $T_{off,C}$ is the nominal temperature at which the cooling system is switched off and $\bar{T}^*_{ext}$ is the simple moving average of the previous $\bar{\phi}$ values of the external sol-air temperature, T^*_{ext}.

2.1.3. Latent load in cooling season

Latent load is due to people, bathing and cooking activities. Thus, users schedule is required to evaluate vapor production per hour. In this work, we deal with only small-capacity dehumidifiers, typically used in residential applications (i.e. portable units). Dehumidification performance, x_{dehum}, depends on indoor air temperature and moisture, together with technological and geometrical characteristics of the device. Here, we refer to the model presented by [16,17] for wet surfaces. Energy and mass balances are evaluated through two main coefficients, that are:
- UA_{dehum}, that is the classical heat transmittance-surface product for heat exchanger analysis;
- C_{dehum}, known as "coil characteristic", used to determine the interrelated values of air enthalpy drop, wet surface temperature (i.e. coil temperature), and saturated air conditions in proximity of the water film on the coil surface (see [15] for more details on the treatise on dehumidifying coils).

Finally, mass balance of water vapor of indoor air reads:

$$x_z' - x_z = \frac{x_{prod} - x_{dehum}}{\rho_z V_z} + n_{air}\left(x_{ext} - x_z\right) \left[kg/kg\right] \tag{4}$$

where x_{dehum} is the hourly moisture removal. Here, we consider dehumidification (i.e. $x_{dehum} > 0$) when floor temperature $T_{f,RF}$ (see section 2.1.6.) is lower than internal dew point.

2.1.4. Domestic Hot Water demand

Similarly to users schedule, hourly profile of DHW demand is a required input. Corresponding energy demand reads:

$$E_{th,DHW} = \rho_w c_w V_{DHW}\left(T_{DHW} - T_{aqu}\right) \qquad \left[kWh\right] \tag{5}$$

where T_{aqu} can be considered equal to the annual average of the outdoor temperature.

2.1.5. Electricity demand for lighting load, cooking and other electric uses

Hourly schedules of lighting, cooking and other electric uses (household appliances, office devices) is needed to calculate electricity consumption in each time step. In this work, we consider electric cooking devices (i.e. induction cooking).

2.1.6. H/C system terminal: radiant floor model

In this work, we consider a radiant floor as heat terminal unit. We simulate the water and floor temperature evolution through a simplified resistance model (Eqs. 6). In particular, floor temperature is used to verify air-vapor condensation during cooling period and so the switching on of the dehumidifiers.

$$\Delta T_{RF} = T_{w,RF} - T_z = \Delta T_{RF,nom}\left(\frac{En_{th}}{S_{RF} K_{RF,nom}}\right)^{\frac{1}{n_{RF}}} \left[K\right] \tag{6.a}$$

$$T_{f,RF,H} = T_{w,RF,H} - \frac{En_{th,H}}{S_{RF} U_{wf}} \qquad\qquad T_{f,RF,C} = T_{w,RF,C} + \frac{En_{th,C}}{S_{RF} U_{wf}} \qquad \left[K\right] \tag{6.b}$$

where ΔT_{RF} is the difference between the mean temperature of the water circulating within the radiant floor, $T_{w,RF}$, and T_z. Floor temperature, $T_{f,RF}$, is evaluated through the thermal transmittance between circulating fluid and floor surface, U_{wf}. In this work, internal air temperature is assumed as constant and equal to room set-point.

2.2. Heat pump generator

We consider air to water electrically-driven HPs with variable capacity control units, namely units able to change the output thermal power without significant penalization of COP. HP nominal capacity is chosen equal to the peak load of the building in order to ensure the matching of the building thermal load in any conditions (i.e. the maximum value between $P_{th,des,H}$ and $P_{th,des,C}$). As suggested by several technical standards (see for instance [18]), the so-called *second-law efficiency* (or *task efficiency*) is used to evaluate HP performances. The latter method reads:

$$COP = \eta_{HP,H}^{II} COP_{id} = \eta_{HP,H}^{II} \frac{T_{cond}}{T_{cond} - T_{eva}} \qquad EER = \eta_{HP,C}^{II} EER_{id} = \eta_{HP,C}^{II} \frac{T_{eva}}{T_{cond} - T_{eva}} \qquad (7)$$

According to manufacturer's data, η_{HP}^{II} can be assumed as a constant with sufficient accuracy. T_{eva} and T_{cond} values depend on the particular service provided, namely $T_{w,RF}$ and $T_{TS,\text{setpoint}}$ (see Fig. 1). An average temperature drop of 5 K is considered at the indoor HP heat exchanger (water side) between $T_{eva/cond}$ and secondary fluid. Similarly, a drop of 10 K is assumed at the outdoor HP heat exchanger (air side). During the heating season, when $T_{ext} < 2 \ °C$, a defrost cycle is performed by means of an electrical resistance (COP=1) for 30 minutes.

2.3. PV generator

We simulate PV performances through the model provided by [19,20], which is based on the work by Evans [21].

$$E_{el,PV} = n_{PV} \left(\eta_{inv} \eta_{PV} S_{PV} I_{sol,PV} \right) \qquad [kWh] \qquad (8.a)$$

$$\eta_{PV} = \eta_{PV,ref} \left[1 - \beta_{T,PV} \left(T_{PV} - T_{ref,PV} \right) \right] \quad [-] \qquad (8.b)$$

$$T_{PV} - T_{ext} = \left(219 + 819 K_t \right) \frac{NOCT - 20}{800} \ [°C] \qquad (8.c)$$

where $K_t = I_{sol}/I_{sol,o}$ is the hourly clearness index [22], evaluated through TMY data from Italian CTI [23]. $I_{sol,o}$ is evaluated through classical literature formulations (see for instance [19]).

2.4. ST generator

ST generator is simulated through the classical model illustrated in technical standards [22,24]. It reads:

$$E_{th,ST} = n_{ST} \left(\eta_{ST} S_{ST} I_{sol,ST} \right) \qquad [kWh] \qquad (9.a)$$

$$\eta_{ST} = F_R \left(\tau \alpha \right)_n \left(1 - b_0 \left(\frac{1}{\cos \vartheta} - 1 \right) \right) - \frac{F_R U_L \left(T_{ST,in} - T_{ext} \right)}{I_{sol.ST}} \quad [-] \qquad (9.b)$$

where $T_{ST,in}$ is assumed to be equal to the value of thermal storage temperature (see Fig. 1).

2.5. Thermal storage

In this work, we consider classical thermal storages for tertiary applications. Internal energy variation is calculated through a simplified lumped component model. It reads:

$$V_{TS} \rho_w c_w \left(T_{TS}' - T_{TS} \right) = E_{th,ST} + E_{th,HP,TS} - \left(E_{th,DHW} + E_{th,TS,H} \right) - E_{th,TS,ls} \ [kWh] \qquad (10)$$

TS energy losses, $E_{th,TS,ls}$, depend on TS insulating characteristics and storage room temperature, $T_{ext,TS}$.

$$E_{th,TS,ls} = S_{TS} \frac{\lambda_{TS}}{s_{TS}} \left(T_{TS} - T_{ext,TS} \right) \qquad [kWh] \qquad (11)$$

3. Overall energy balance and control strategy

3.1. Thermal energy balance

The control strategy of the above-described thermal energy fluxes is a function of T_{TS} evolution. The latter one depends on four control parameters: $T_{TS,\text{setpoint}}$, $T_{TS,up}$, $T_{TS,\text{down}}$, $T_{TS,\text{max}}$. The first one is the nominal set-up temperature of the storage, the second one is the TS temperature for switching on/off the heat pump from direct heating mode, the third one is the minimum temperature at which the heat pump is used to reheat the storage, and the forth one is the maximum thermal storage water temperature. In short, TS control strategy can be summarized as follows:

- if $T_{TS} > T_{TS,up}$ both DHW and heating loads are delivered by removing thermal energy from the storage, i.e. $E_{th,TS,H} > 0, E_{th,DHW} > 0, E_{th,HP,TS} = 0$ (see Fig. 1.a)
- if $T_{TS,\text{down}} < T_{TS} < T_{TS,up}$ only DHW load is delivered by removing thermal energy from the storage; heating load is delivered by the HP unit without removing heat from the storage, i.e. $E_{th,TS,H} = 0, E_{th,DHW} > 0, E_{th,HP,TS} = 0$ (see Fig. 1.b)
- if $T_{TS} < T_{TS,\text{down}}$ the HP is used to rise TS temperature to $T_{TS,\text{setpoint}}$, $E_{th,TS,H} = 0, E_{th,DHW} > 0, E_{th,HP,TS} = 0$ (see Fig. 1.c)
- if $T_{TS} > T_{TS,\text{max}}$ no additional thermal energy is stored in TS, hence $T_{TS} = T_{TS,\text{max}}$

During the cooling season, the control strategy of the TS is the same as the heating one. For cooling service, HP is coupled directly with the radiant floor, without using storage systems (see Fig. 1.d).
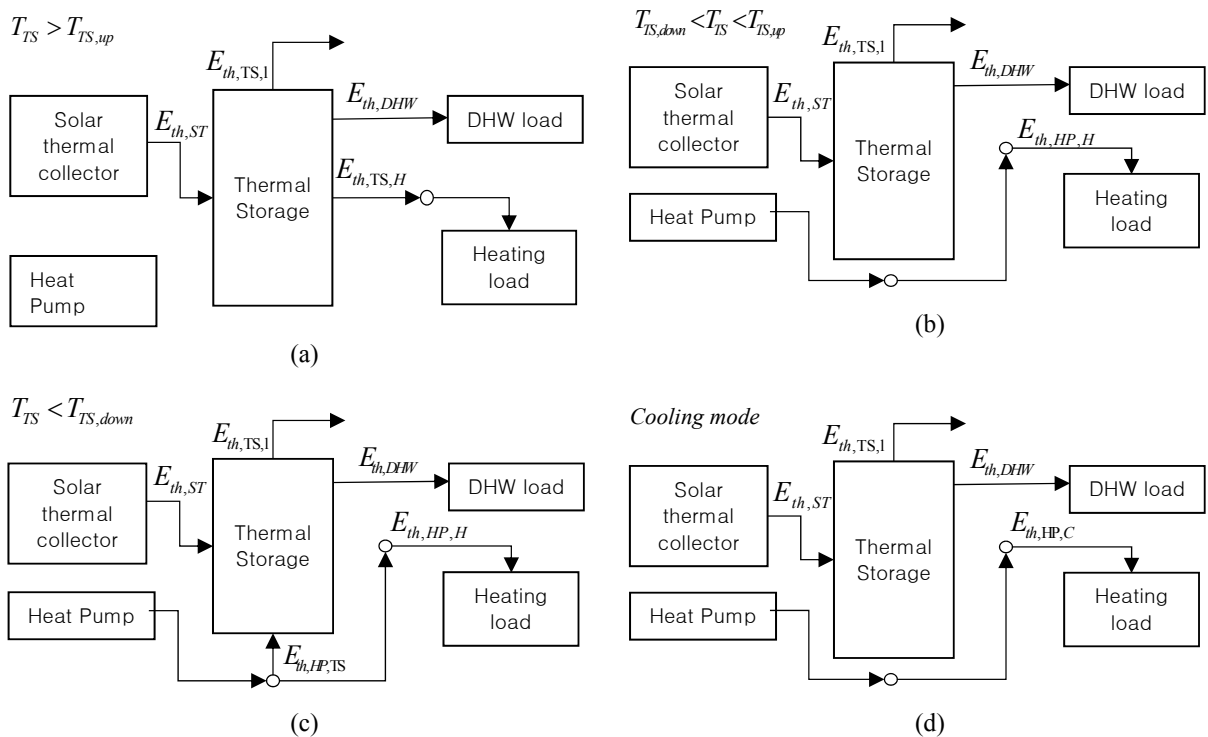

Fig. 1. Schematics of modes of operation.

3.2. Electrical energy balance: grid exchanges

In this work, we do not deal with electrical storage elements, therefore any surplus or deficit in electricity production is exchanged with the grid. The electrical energy balance reads:

$$E_{el,grid} + E_{el,PV} = E_{el,HP} + E_{el,LGT} + E_{el,CK} + E_{el,OU} \qquad [kWh] \qquad (12)$$

Net primary energy consumption at the end of the project lifetime is simply given by the following expression:

$$E_{in}^{TOT} = f_{p,el} \sum_{lifetime} E_{el,grid} \qquad [kWh] \qquad (13)$$

where $f_{p,el}$ is assumed to be 2.5, in accordance with the value proposed by the European Directive on energy efficiency 2012/27/EU.

4. Preliminary application example and results of the proposed method

In this section, we illustrate some relevant outputs evaluated through the application of the just-described simulation methodology to the reference case (i.e. a farm hostel). Building energy demands and energy equipment characteristics are shown in Table 1. Possible outcomes refer to both evolution of operative variables and energy balances over chosen periods. For instance, Figs. 2.a, 2.b, and 2.c show the temperature evolution of the TS, the COP of the heat pump in heating mode, and the PV output with respect to the electrical demand of the building, respectively. Regarding integral evaluations, Fig. 2.d presents the share of the various services on the total electricity consumption over one-year operation.

Table 1. Building energy demands and energy equipment characteristics.

Building energy loads	Radiant floor	PV generator
Heating: 777.2 kWh/y	$\Delta T_{RF,nom} = 20\ K$	$S_{PV} = 1.5\ m^2$
Cooling: 422.0 kWh/y	$S_{RF} = 320\ m^2$	$n_{PV} = 20$
DHW: 483.1 kWh/y	$K_{RF,nom} = 60\ W/m^2$	$\eta_{inv} = 0.85$
Cooking: 82.1 kWh/y	$n_{RF} = 1.1$	$\eta_{PV,ref} = 0.13$
Lighting: 203.8 kWh/y	$U_{wf} = 6\ W/(m^2 K)$	$\beta_{T,PV} = 0.004\ 1/K$
Other electric uses: 134.0 kWh/y		$T_{ref,PV} = 25\ °C$
		$NOCT = 45\ °C$
ST generator	**Thermal storage**	**Heat pump generator**
$S_{ST} = 3\ m^2$	$V_{TS} = 2\ m^3$	Nominal capacity: 16.5 kW
$n_{ST} = 4$	$T_{TS,set-point} = 50\ °C$	$\eta_{HP,H}^{II} = 0.45$
$F_R = 0.8$	$T_{TS,up} = 60\ °C$	$\eta_{HP,C}^{II} = 0.35$
$(\tau\alpha)_n = 0.7$	$T_{TS,down} = 42\ °C$	
$U_L = 5\ W/(m^2 K)$	$T_{TS,max} = 90\ °C$	
$b_0 = 0.1$	$\lambda_{TS} = 0.04\ W/(mK)$	
	$s_{TS} = 0.08\ m$	

Depending on design objectives, one can investigate each design variable in order to maximize proper economic or energy performance indexes. The minimization of total lifecycle cost is one of the most common goals in professional activity; thus, in section 5, we will introduce the mathematical formulation of the optimization problem for C^{TOT} value.
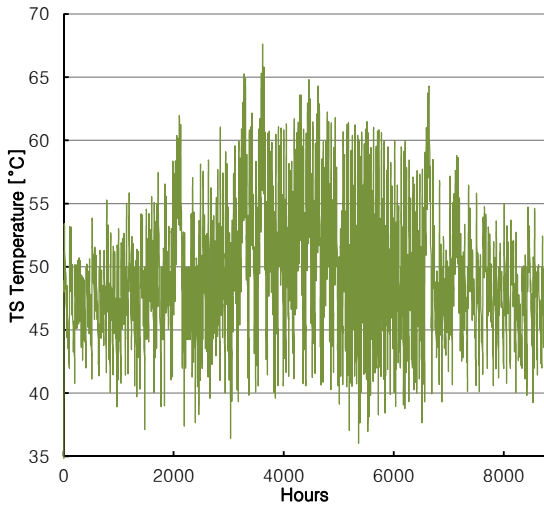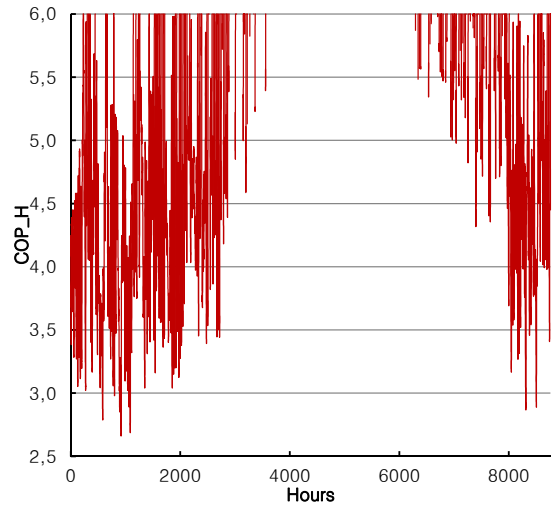
Fig. 2.a. TS temperature evolution.

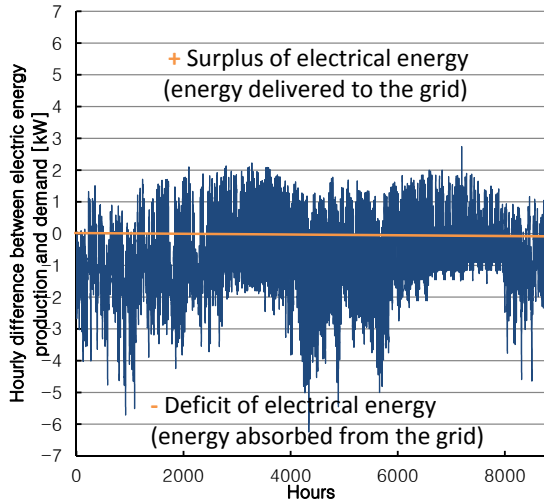Fig. 2.b. Hourly COP evolution in heating mode.

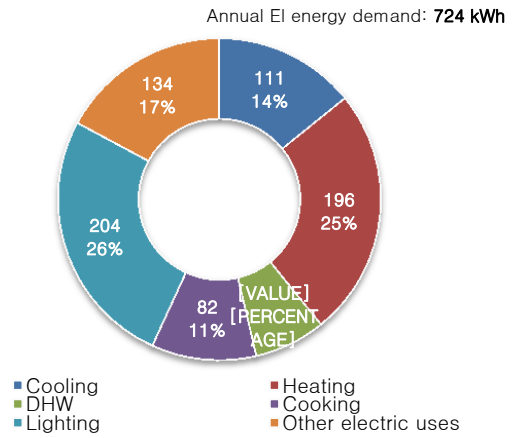

Fig. 2.c. Electrical energy exchanges with the grid.

Fig. 2.d. Share of electrical energy consumption for each service.

5. Formulation of the optimization problem

According to the above-described models and control strategy, a simple optimization problem can be formulated, with the aim of minimizing the global cost of the system, evaluated as:

$$C^{TOT} = c_{0,PV} n_{PV} + c_{0,ST} n_{ST} + c_{0,TS} V_{TS} + C_{0,HP} + \\ + \sum_{lifetime} \left\{ c_{el,in} \max\left[0; E_{el,grid}\right] + c_{el,out} \min\left[0; E_{el,grid}\right] \right\} \qquad [\text{€}] \qquad (14)$$

where n_{PV}, n_{ST}, V_{TS}, and $T_{TS,up}$ are the optimization variables of the problem. By investigating those values, we can obtain the best sizing and control strategy depending on the economic parameters c_0, $C_{0,HP}$, $c_{el,in}$, and $c_{el,out}$. We stress that, according to current regulation, there is a upper limitation on the year-round incomes due to the selling of electrical energy. In other words, on a yearly base, no revenue is provided for the amount of energy delivered to the grid that exceeds the amount of the taken one.

Starting from the just-found best configuration, we can investigate the nearly optimal configurations in order to assess the cost-optimality of the solution. Possible indicators are energy savings or installation costs (i.e. the size of the investment). This further analysis is recommended as different installation expenditures can lead to the very same C^{TOT} or E_{in}^{TOT} values.

An alternative optimization strategy deals with the well-known Pareto optimality: we can test different design and control solutions to find a Pareto frontier in terms of C^{TOT} and E_{in}^{TOT}, allowing cost-benefit considerations. We will come back to these points in the second part of the work [13].

6. Conclusions

In the present work, we introduced a plain and innovative approach to the design of solar thermal and photovoltaic systems in a typical nearly Zero-Energy Building, based on the simulation of the entire lifecycle of the system. We proposed simplified models and set of equations for performance prediction of both building and system components. We dealt with heating, cooling, and dehumidification building requirements, together with the efficiency of low-temperature heat emitter (i.e. radiant floor), thermal storage, and typical thermal and electrical generators for NZEBs (viz. PV panels, ST collectors, thermal storage, and heat pumps). Some examples of the procedure outputs were shown in section 4. Results show as the proposed model can be considered as an effective tool for advanced-energy design; indeed, it is able to predict the effect of different design alternatives on both single component operation and final overall system performances. Furthermore, due to the reduced computational effort of the model, it can be easily coupled to optimization routines in order to find the optimal values of design parameters according to energetic and economic cost-benefit considerations.

In this first part, we also introduced suitable objective functions and indicators for the cost-optimality of the solution. In particular, we suggest comparing different design solutions depending on both global cost at the end of project lifetime, installation costs, and energy performances. Future works will be aimed at applying simulation-based optimization techniques to relevant case studies.

References

[1] Evins R. A review of computational optimisation methods applied to sustainable building design. Renew Sustain Energy Rev 2013;22:230–45.

[2] ASHRAE. Energy Estimating and Modeling methods. ASHRAE Handb. - Fundam., Atlanta (GA): American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE); 2009, p. 19.1–19.39.

[3] Nguyen A-T, Reiter S, Rigo P. A review on simulation-based optimization methods applied to building performance analysis. Appl Energy 2014;113:1043–58.

[4] BPIE. Cost Optimality – Discussing methodology and challenges within the recast Energy Performance of Buildings Directive 2010:40.

[5] Carrilho da Graça G, Augusto A, Lerer MM. Solar powered net zero energy houses for southern Europe: Feasibility study. Sol Energy 2012;86:634–46.

[6] Eshraghi J, Narjabadifam N, Mirkhani N, Sadoughi Khosroshahi S, Ashjaee M. A comprehensive feasibility study of applying solar energy to design a zero energy building for a typical home in Tehran. Energy Build 2014;72:329–39.

[7] Thiers S, Peuportier B. Energy and environmental assessment of two high energy performance residential buildings. Build Environ 2012;51:276–84.

[8] European Parliament and Council of the European Union. Directive 2010/31/EU on the Energy Performance of Buildings (Recast) 2010.

[9] Hamdy M, Hasan A, Siren K. A multi-stage optimization method for cost-optimal and nearly-zero-energy building solutions in line with the EPBD-recast 2010. Energy Build 2013;56:189–203.

[10] Kapsalaki M, Leal V, Santamouris M. A methodology for economic efficient design of Net Zero Energy Buildings. Energy Build 2012;55:765–78.

[11] Charron R, Athienitis AK. Design and Optimization of Net Zero Energy Solar Homes. ASHRAE Trans 2006;112:285–95.

[12] Horowitz S, Christensen C, Anderson R. Searching for the optimal mix of solar and efficiency in zero net energy buildings. Am. Sol. Energy Soc. - Sol. 2008, Incl. Proc. 37th ASES Annu. Conf., 33rd Natl. Passiv. Sol. Conf., 3rd Renew. Energy Policy Mark. Conf. Catch Clean Energy Wave, vol. 8, 2008, p. 5016–44.

[13] Testi D, Schito E, Conti P. Cost-optimal sizing of solar thermal and photovoltaic systems for the heating and cooling needs of a nearly Zero-Energy Building: the case study of a farm hostel in Italy. SHC 2015, Int. Conf. Sol. Heat. Cool. Build. Ind., Istanbul (TR): 2015

[14] EN ISO 13786-2007 Thermal Performance of Building Components - Dynamic Thermal Characteristis. Brussels: European Committee for Standardization (CEN); 2008.

[15] O'Callaghan PW, Probert SD. Sol-air temperature. Appl Energy 1977;3:307–11.

[16] Air-Cooling and Dehumidifyng Coils. ASHRAE Handb. 2008 —HVAC Syst. Equip., Atlanta (GA): American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE); 2008, p. 22.1–22.16.

[17] Stoecker WF, Jones JW. Refrigeration and Air Conditioning. New York (NY): McGraw-Hill Publishing Company; 1983.

[18] EN 15316-4-2, Heating systems in buildings - Method for calculation of system energy requirements and system efficiencies - Part 4-2: Space heating generation systems, heat pump systems. Brussels: European Committee for Standardization (CEN); 2008.

[19] RETScreen® International. Photovoltaic project analysis. Minister of Natural Resources Canada; 2004.

[20] Riffonneau Y, Bacha S, Barruel F, Ploix S. Optimal Power Flow Management for Grid Connected PV Systems With Batteries. IEEE Trans Sustain Energy 2011;2:309–20.

[21] Evans DL. Simplified method for predicting photovoltaic array output. Sol Energy 1981;27:555–60.

[22] Duffie JA, Beckman WA. Solar Engineering of Thermal Processes. 4th ed. John Wiley & Sons, Inc.; 2013.

[23] CTI. Typical Meteorological Year. Milan (IT): Italian Committee of Thermotechnics (CTI); 2012.

[24] EN 12976-2, Thermal solar systems and components – Solar collectors – Part 2: Test methods. . Brussels: European Committee for Standardization (CEN); 2006.