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Optimum hybridization of wind turbines, heat pumps, and thermal energy storage systems for nearly zero-exergy buildings (NZEXB) using rational exergy management model

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Abstract

The definition Nearly-Zero Exergy Building (nZEXB) is a new concept regarding 4DE (Fourth-Generation District Energy) systems. It recognizes different energy exchanges between the district and the building at different exergy levels. In nZEXB systems, a multitude of sustainable systems and renewable energy resources are mobilized. Their optimum bundling in design and load allocation is a complex problem and must be based on a robust platform of a common objective, which in this case is the minimization of exergy destruction that eventually leads to additional CO_2 emissions. An optimization model was developed for a ground-source heat pump with thermal energy storage coupled to a wind turbine. Coefficient of performance (*COP*) and Primary Energy Ratio (*PER*) were redefined in terms of exergy to serve the objective function of minimizing exergy destructions. In developing the optimization model, the *Rational Exergy Management Model* (REMM) was employed, which aims to increase the balance among the supply and demand exergies and helps to establish a circular exergy flow. The impact of several design variables like the type of terminal units, the size and efficiency of the wind turbine, reservoir temperature, and the split of the turbine electricity between the heat pump and the building were investigated. It has been concluded that heat pumps play a major role in achieving nZEXB status provided that an optimum bundling with other sustainable systems and equipment is achieved.

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Nomenclature

Symbol	Description	Symbol	Description
a	Heat pump constant (Eq.2)	d, des	Demand, design, destroyed
C	Cost	E, EX, x	Exergy
COP	Coefficient of performance	f	Primary Energy Source (Fuel)
c, d	Coefficients (Eq. 3, Eq. 10, Eq. 5)	HP	Heat pump
E	Electrical power, W	o	Environment
E_x	Exergy, W	r	Reservoir, return
h	Overall heat transfer coefficient, W/K^z	ref	Reference
m	Exponent in Eq. 5	TH, H	Thermal, heat
n	Exponent in Eq. 3	p	Pump

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OF	Oversizing factor
OT	Operative temperature, K
PER	Primary energy ratio
PEXR	Primary exergy ratio
Q	Thermal power, W
T	Temperature, K
X	Electrical power split to the building and heat pump
y, z	Coefficients in Eq. 16 and Eq. 12
η, η_I	First-law efficiency
ΔP	Unit pressure drop due to oversizing
Ψ_R	REMM efficiency
ε	Unit exergy, W/W or kW/kW
Subscripts	
a	Comfort air
s	Supply
z	Temperature power for overall heat transfer coefficient, *h*
Acronyms	
AC, DC	Alternating, direct current
CHP	Combined heat and power
GSHP	Ground-source heat pump
HVAC	Heating, ventilating, air-conditioning
LowEX	Low-exergy
nZEXB	Nearly-zero exergy building
nZEB	Nearly-zero energy building
PV	Photo-voltaic
REMM	Rational Exergy Management Model
TES	Thermal energy storage
WT	Wind turbine

1. Introduction

Heat pumps are becoming particularly important for the net (or nearly) zero energy buildings. The main claim is their high Coefficient of Performance (*COP*) according to the First-Law of Thermodynamics but the Primary Energy Ratio must be factored in for a better understanding of their performance:

$$PER = \eta_{ITH} \cdot COP \qquad \{\text{Quantity flow of energy from the primary resource}\} \qquad (1)$$

This equation shows how much primary energy in terms of quantity is used for unit heat provided/extracted by a heat pump with a given *COP*, where η_{ITH} is the overall thermal efficiency covering the entire energy flow process starting from the fuel input to the thermal power plant and ending up to the point of use at the heat pump, including transmission and distribution losses. Eq. 1 is also useful in predicting the primary CO_2 emissions that is the responsibility of the heat pump. However, as will be shown later, there are other avoidable CO_2 emissions responsibility of the heat pump due to exergy destructions and mismatches in supply and demand exergies. The most recent EU28 average for η_{ITH} is about 0.43 [1] minus 0.081 for transmission and distribution losses of the power output at the plant (2013 World average) [2]. In many other countries, this figure is less than 0.3. Therefore, η_{ITH} at best is 0.4. If for example, seasonal average of a space heating ground-source heat pump COP_H is 3, then this heat pump hardly beats a condensing natural-gas boiler with an average efficiency of 0.90, because *PER* of the heat pump in this case is only 1.2.

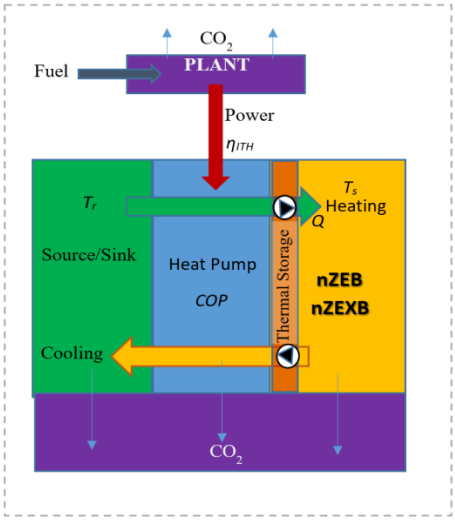


Fig. 1. Embedding boundaries of a heat pump and emission points [5]

PER further decreases if a building demands higher heating fluid temperatures. Therefore, the trend is to use

LowEx (Low-Exergy) buildings [3], with moderate demands for comfort heating and cooling temperatures. A logical approach is to eliminate distribution losses by generating mechanical power on site and drive the heat pump directly, and better to mention is, to use on-site cogeneration systems [4]. These points remind that in evaluation of the performance of a heat pump may not be considered isolated from the source and the demand points. Figure 1 shows the proper boundary selection of a heat pump operation (dotted lines) [5]. Fig. 1 shows that a heat pump is functionally embedded between the thermal source (in heating mode)/sink (in cooling mode) and the application point like a nearly-zero energy building (nZEB) or nearly-zero exergy building (nZEXB). In order to normalize the heat supply and demand rates plus storing energy in order to time shift energy obtained from interrupted renewable energy resources and systems like wind turbines and PV systems, an energy storage system is essential between the building and the heat pump. Furthermore, the same heat pump is linked to the power plant using primary energy resources and linked by the power transmission and distribution lines. The entire system has CO_2 and other harmful emission responsibilities not only due to energy inefficiencies but also the exergy destructions at several points. CO_2 emissions are greatly reduced in LowEx buildings with high *COP* heat pumps and radiant panel systems [6]. This is especially true if an optimum operating temperature T_s is calculated [7]. In this token, radiant panel systems are exergetically the best couple to heat pumps in net zero energy buildings [8]. *COP* of a heat pump in heating is directly related to the supply temperature of the heat pump. In an approximated form over a relatively small range of $35^oC \leq T_s \leq 70^oC$: $COP_H = \frac{a}{(T_s - T_r)}$ (2)

It is evident that for a given resource (reservoir) temperature T_r, like the ground temperature for a shallow geothermal system, COP_H increases with decreasing T_s, so that space heating with low supply temperatures are becoming more and more important for sustainable and green buildings. On the other hand, the heat output of terminal equipment, Q decreases with a decrease in T_s:

$$Q = c(T_s - T_a)^n \tag{3}$$

Here c and n are the characteristics of the specific equipment used. T_a is the indoor comfort temperature. To be more precise, T_a may be replaced by *OT*, namely the operative temperature, because most of the terminal equipment types, like panel heating and cooling systems transfer heat with both convection and radiation in different proportions depending upon the type of the equipment. Eq. 3 clearly shows that heating capacity of a given terminal equipment of any type with a power of n decreases if T_s decreases. This conflict, i.e. while trying to decrease T_s in order to increase the heat pump COP_H, reduction of the heating capacity of the terminal equipment the building may be resolved simply by oversizing the equipment [7, 9]. If the terminal units are operated at a different temperature than their design value T_s, namely T_{sd}, then the oversizing factor, *OF* is:

$$OF = \left[\frac{(T_{sd} - T_a)}{(T_s - T_a)} \right]^n \tag{4}$$

Equipment oversizing may be done either by series extension or parallel addition of equipment [4]. In series oversizing, the hydronic circuit length becomes longer (i.e. the length of the tube coils in a panel heating or cooling system), thus the head loss becomes proportional to the *OF*. However, according to a specific pump curve, the flow rate decreases and therefore the average fluid flow across the tube length further decreases, which ends up in a further decrease of the heat output while the average fluid temperature decreases [9, 10], requiring a larger *OF*. This is especially important in long circuits and district heating systems [11]. Additional pressure losses due to singularities of oversizing makes the relationship between the pressure head increase ΔP and *OF* different than a linear one. In parallel oversizing, the flow rate, V increases proportionally with *OF*, thus the relationship between ΔP and *OF* becomes primarily a parabolic one or similar:

$$\Delta P = d \cdot OF^m \tag{5}$$

Here, d and m depend upon the type of oversizing, the equipment used, and specific application of oversizing. Equations 3 and 4 show that terminal equipment with the power n close to one, like panel heating systems are preferable to other type of units like fan-coils with n about 1.4, because if n is high then the need for oversizing will be higher. An economical optimization methodology was developed for determining the most economical solution by oversizing the heat pump and/or oversizing the equipment [7]. A further parametric study for design

optimization of nZEB buildings was carried out [10]. Yet economical and First Law analyses may not be sufficient to describe the actual performance of renewables and heat pumps and their true impact on the environment by applying the principles of the Second Law of Thermodynamics. Exergy analysis is becoming more critical and descriptive for systems operating at low temperatures and using low-enthalpy renewable energy resources with or without heat pumps. In order to accomplish such a task, an exergy-based methodology for design optimization and analysis have been developed in this study. So far in the literature, almost all performance analyses have been mostly based on the First Law of Thermodynamics. Especially with the advent of district energy systems, connected buildings are not only exchanging power but also several levels of heating, cooling, and domestic water exergy. Because these exergies may not match with the exergy supply of the district, an nZEB building may not be an nZEXB building [12]. An example follows in Fig. 2. A green building exchanges both electrical and thermal power with a district energy system. On an annual basis, the building receives 10,000 kW-h of AC electrical and provides 10,000 kW-h AC electrical energy with the same quality. At the same time, the building receives 15000 kW-h of heat in the form of hot water annually from the district at an average supply temperature of 353 K (80ºC) and provides 15000 kW-h of thermal energy at an average temperature of 343 K (70ºC) annually. Electrical and thermal power is generated in the premises of the green building (solar PV, solar collectors, wind turbine, heat pump, biogas-operated combined heat and power). According to the current practice, this building is an ideal net-zero energy building, because on an annual basis it receives and returns the same quantity of electrical and thermal energy. This conclusion is based on the First Law of Thermodynamics, without taking into account the exchange of the quality (Exergy) of heat. In fact, the temperature of hot water received is higher than the hot water supplied to the grid.

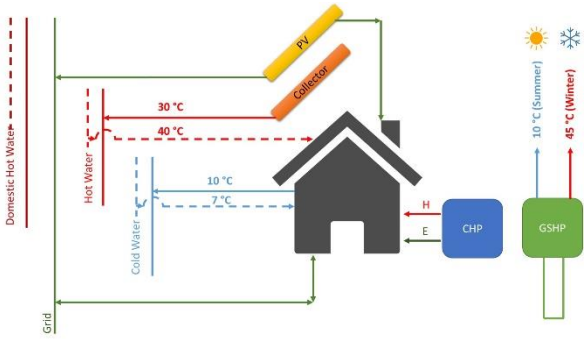


Fig. 2. Thermal and power relation between nZEB and district [12]

The quality of heat supplied from the grid, namely the unit exergy of the 1 kW-h of the supply heat, ε_s according to the ideal Carnot Cycle must be considered:

$$\varepsilon_s = \left(1 - \frac{T_{ref}}{T_s}\right) \times (1\,\text{kW - h}) \tag{6}$$

T_{ref} is the equilibrium environment reference temperature. In this example, it is taken to be 283 K (Average ground temperature at shallow ground loops of the heat pump). The green building receives annually15000 kW-h. The total supplied exergy E_{xs} is:

E_{xs} = (1-283/353) · 15000 = 2974,5 kW-h.

Repeating the same calculation for the exergy of the annual heat supplied to the grid from the green building:

E_{xr} = (1-283/343) · 15000 = 2623,9 kW-h.

E_{xr} is less than E_{xs} and the exergy balance is in deficit. These exergy-based results conclude that, although the building seems to be a net-zero energy building, it is not when both the first and second laws of thermodynamics are considered. This consideration is also very important for CO_2 emission calculations. Therefore, besides the definition of LowEx building NZEXB and nZEXB, these definitions must follow:
NZEXB: Net-zero Exergy Building is a building, which is connected to a district energy system and on an annual basis, supplies an equal amount of the sum of thermal exergy at different temperatures and power exergy to the district to the total amount that receives from the district ..

nZEXB: Nearly- zero Exergy Building is a building, which satisfies 80% of the NZEXB target.
LoWEXB: Low-Exergy Building is a building, which can satisfy all thermal loads between 40°C and 17°C.

$$PEXR = \frac{E_s}{E_f} = \frac{\varepsilon_s \cdot Q_s}{\varepsilon_f \cdot \frac{Q_s}{(\eta_{ITH} \cdot COP)}} = \frac{\left(1-\frac{T_{ref}}{T_s}\right)}{\left(1-\frac{T_{ref}}{T_f}\right)} \cdot (\eta_{ITH} \cdot COP) = \frac{\left(1-\frac{T_{ref}}{T_s}\right)}{\left(1-\frac{T_{ref}}{T_f}\right)} \cdot PER \tag{7}$$

Here, ε_s is the unit exergy of the heat supplied by the heat pump at a temperature of T_s. The heat pump demands electrical power at a rate of $1/COP$ per unit heat output power. This power is supplied by the primary fuel by an overall efficiency of η_{ITH} at an adiabatic flame temperature T_f. Defining COP_{EX} in the following format,

$$COP_{EX} = \frac{\left(1-\frac{T_{ref}}{T_s}\right)}{\left(1-\frac{T_{ref}}{T_f}\right)} \cdot COP \tag{8}$$

$$PEXR = \eta_{ITH} \times COP_{EX} \quad \{\text{Quality flow of energy from the primary resource}\} \tag{9}$$

PEXR deals with the *quality* flow of primary energy resource up to the demand point, while *PEX* deals with the *quantity* flow. It is worth to mention that *PEXR* takes into account the type (exergy) of the fuel input in terms of its unit exergy, which enables a more comprehensive analysis, where *PER* is insensitive to the type or quality of fuel. Thus, the type of fuel, whether it is renewable or not is not distinguishable, leaving the analysis of the performance of a heat pump incomplete. For the same example given earlier in this paper (η_{ITH} = 0.4, COP_H = 3, *PER* = 1.2), if T_f is 2000 K, T_d is 333 K, and T_{ref} is 283 K, then the corresponding *PEXR* from Eq. 7 is:

$$PEXR = \frac{\varepsilon_s}{\varepsilon_f} \cdot PER = \frac{\left(1-\frac{283}{333}\right)}{\left(1-\frac{283}{2000}\right)} \cdot (0.4 \times 3) = 0.209$$

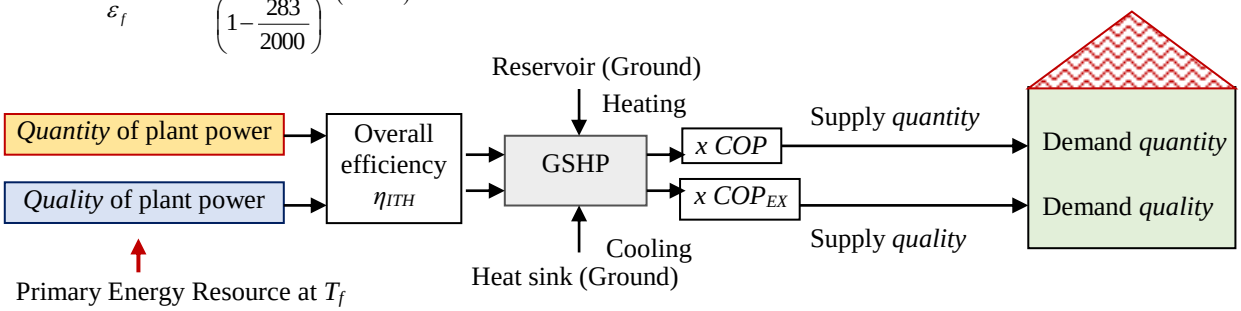


Fig. 3. Energy and exergy flow chain for a heat pump performance

Comparison of this value with, *PER*, which is 1.2 shows that *PER* alone may be misleading for designers and energy planners. PER may only be useful to a certain extent in economic analyses, but not in environmental and technical issues. Yes, on one side this ground-source heat pump provides 20% points more heat output (utilizing reservoir heat) than the primary energy resource used, but on the other side, it more importantly uses a high-exergy fossil fuel with a unit exergy of about 0.86 W/W (ε_f) but produces only hot water (low-exergy) at 333 K, corresponding to a unit exergy of 0.15 W/W (ε_s). This means that there exists a big exergy destruction in the process, which at the same time translates into additional emissions [13]. If this system needs to have a break-even value for *PEXR*= 1, then the required *COP* may be calculated. From the above arithmetic, this *COP* has to be 14.3. With the current technology, this is a hardly achievable value compared to practically maximum *COP* value of 6. Heat pumps need to be driven with on-site or close-to-site renewable energy resources and systems for better environmental performance. For example, if the fossil fuel is replaced with solar power having ε_f = 0.49 W/W [4], then *PEXR* improves to 0.37. Furthermore, if the heat demand temperature is reduced from 333 K to 313 K considering a LowEX building, the result becomes complicated due to the fact that, while the supply temperature is reduced, supply exergy is also reduced. This reduces *PEXR*, but at the same time the same

reduction in the supply temperature increases the heat pump *COP*. For example, if the *COP* increases from 3 to 4.5, then *PEXR* becomes 0.35, hardly equals the previous value. One may conclude that the use of LowEX building technology may not be sufficient to improve the exergy performance of a heat pump system. A more effective solution is to use on-site renewable sources. For example, if η_{ITH} increases from a grid power value of 0.4 to 0.7, then *PEXR* increases to 0.61. Therefore, we suggest the future priorities:

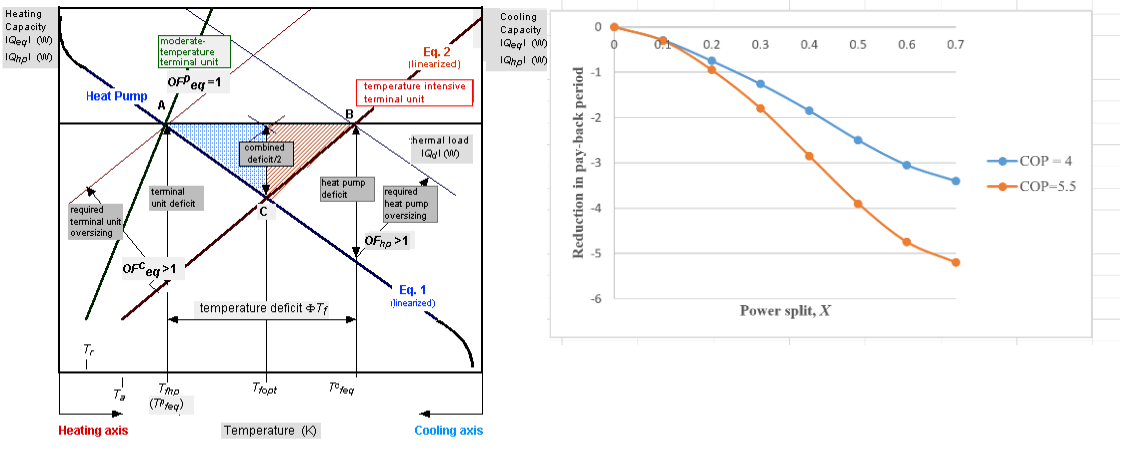
1. Use local power
2. Use renewable energy resources (wind, solar, etc.) for power production on-site
3. Use high *COP* heat pumps
4. Use LowEX buildings
5. Try deep geothermal rather than shallow geothermal resources to increase T_r in ground-source heat pumps, while the additional costs are taken into account and a careful optimization is made.

Literature survey

Almost all research deal with the heat pump alone. Even heat pump systems with wind turbines or other renewable systems are treated separately from each other. Furthermore, most of the literature are based on the quantity of the primary fuel consumption and analyze the associated CO_2 emissions accordingly. The results are only useful for economic evaluations like the pay-back period. Fig. 4 on the left-hand side shows such an approach, where the operating fluid temperature is optimized for a compromising situation for equipment versus heat pump oversizing in order to minimize the cost of installation and operation [14, 15]. According to this graph, the required heating capacity of the heat pump versus terminal unit outputs are in conflict and this conflict may be resolved at an optimum oversizing split at the optimum operating fluid temperature. The right-hand side graph in Fig. 4 shows the reduction in the pay-back period of a specific example in terms of the power split (*X*) between the heat pump and the building (1-*X*) received from the on-site wind turbine [16]. According to this graph dedication of the entire power to the heat pump (*X*=1) seems to be economical. However, from the exergy point of view it has been shown above in terms of *PEXR* that once the electrical power is generated it must stay as power instead of converting it to thermal power. This shows that one needs a common base by converting exergy to cost or vice versa [4]: The cost of exergy destruction per unit supply exergy may be embedded into cost equations, like life cycle cost analysis optimizations [17].

$$\Delta C_{EX} = c\frac{\sum \mathcal{E}_{des}}{\mathcal{E}_s} \tag{10}$$

Here, *c* is the unit cost in Euros. A recent approach is the GreenHP initiative, which aims at developing a new, highly efficient urban heating system based on retrofitting of buildings [18]. This is indeed a very promising initiative, because the energy performance of buildings in Europe are quite poor [19]. In this respect, Dietle [20] presented an optimization approach on a components to system synthesis, based on the first-law. Tsaros et al. [21] studied on exergy analysis of heat pumps. They gave a good account of heat pumps alone by considering exergy transports between the components, to the loads, and to the environment. Regarding exergy and energy analyses about wind energy-coupled heat pumps, Ching-Song J. et.al. analyzed direct wind turbine and solar collectors coupled heat pump with emphasis on the First-Law [22]. Performance of a rooftop wind solar hybrid heat pump in terms of the Second-Law was investigated for homes and was concluded that solar systems have low exergy efficiency and therefore wind energy must be prioritized. In this respect, Qin-Yi Li et al. [23] performed an exergetic analysis of a domestic-scale system combining a 5 kW wind turbine and an array of 11.4 m^2 copper flat-plate solar collectors as well as two heat pumps, capable of producing space heating, hot water, cooling and electricity [24]. They found that wind power can provide 7.6% of the yearly heat pump power demand to satisfy the thermal loads of a 198 m^2 residential building in Beijing. The system can yearly reduce 31.3% carbon dioxide emission compared with conventional energy systems. Dincer and Rosen discussed wind turbine driven heat pumps giving fundamental equations regardless of analyzing their mutual energy and exergy relations [25].



Power split, *X* [16]

5]

Fig. 4. Economy-based optimum fluid temperature and power split analysis

Development of the REMM-based optimization model

In this model, there is a wind turbine system (WT), which provides electric power to the ground-source heat pump (GSHP). It also provides electricity to the nZEXB. The power split is denoted by *X*, which is the ratio of the power dedicated to the GSHP at design conditions to the total power generation capacity of the wind turbine at design conditions. Excess power is stored in the form of heat produced by the GSHP in TES. The model is shown in Fig. 5, which is divided into three sections, namely the supply, storage, and demand. The demand side is the building with floor heating system and is going to be oversized by adding more parallel circuits. Inverter exergy destructions are eliminated by using DC power in the building and the entire system (lighting, appliances, pumps etc.). Assuming that the TES tank is properly insulated, two major exergy demand points are identified namely the circulation pumps, namely P1 and P2. In the optimization process P3 is not included, because it is fixed at design conditions. Thus, the model reduces to optimize the operating fluid temperature, which affects *COP*, COP_{EX}, equipment (floor panel) oversizing, and circulation exergy demands. In this model, the oversizing related *exergy* demands are identified by the following expressions derived for unit heating load in nZEXB:
Pump P2: Based on Eq. 4 and Eq. 5,

$$E_{P2} = \left(\frac{d}{\eta_{P2}}\right)\left[\frac{(T_{sd} - T_a)}{(T_s - T_a)}\right]^{2n} \qquad \{\text{Here } E \text{ is exergy}\} \tag{11}$$

Here, T_{sd} and T_a are the design values for the equipment supply temperature and the indoor comfort air temperature, respectively. h and T_o are given values for a specific design and environment.

TES: TES has a heat loss with an overall heat transfer coefficient of h, to its environment at a temperature T_o.

$$E_{TES} = h(T_s - T_o)^{\tilde{r}} \cdot \left(1 - \frac{T_{ref}}{T_s}\right) \tag{12}$$

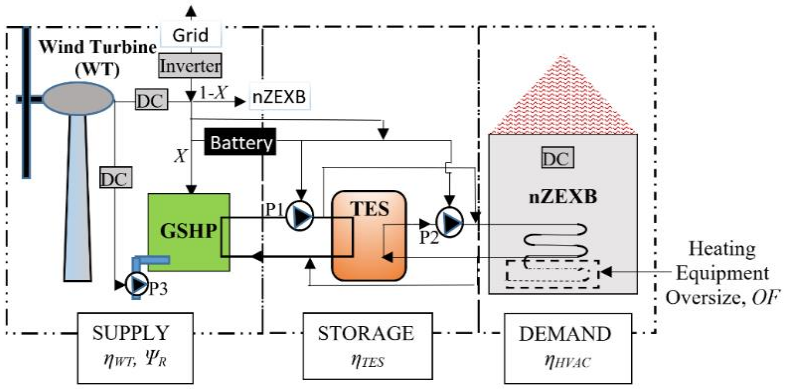


Fig. 5. nZEXB with GSHP and wind turbine

Pump P1: This pump follows P2 and in thermal energy balance between the heat pump and nZEXB, E_{P1} is Equal to E_{P2}, with the exception that it must take into account of η_{TES}. The reservoir pump P3 has a fixed operation (practically independent of T_s) so that it is not taken into account for the optimization process.

$$E_{P1} = E_{P2} / \eta_{TES} \tag{13}$$

Heat Pump

$$E_{HP} = \left(\frac{1}{a \cdot \eta_{HVAC}} \right) (T_s - T_r) \tag{14}$$

Then the partial objective function becomes:

$$\Sigma E = E_{P1} + E_{P2} + E_{TES} + E_{HP} \qquad \{\text{minimize}\} \tag{15}$$

Here E is exergy. The performance of the wind turbine for unit power output depends on T_s in terms of REMM efficiency Ψ_{RWT}. This value may be added to Eq. 15 with a weighing factor, y for complete optimization.

$$\Sigma E = E_{P1} + E_{P2} + E_{TES} + E_{HP} + y/\Psi_{RWT} \qquad \{\text{minimize}\} \tag{16}$$

$$\psi_{RWT} = 1 - \frac{\left(1 - \frac{T_{ref}}{T_s}\right)}{\eta_{WT}} \qquad [4] \tag{17}$$

For a given design, Eq. 15 is a function of the single variable T_s, so that Eq. 18 gives the optimum T_s.

$$dE/dT_s = 0, \tag{18}$$

Case study

A specific design example similar to the arrangement given in Fig. 5 has been optimized. The design inputs and other equipment parameters are: n = 1.3, d =1, η_{P2} = 0.9, η_{TES} = 0.85, η_{HVAC} = 0.9, h = 20, a = 2, z = 1.1, $T_a = T_o$ = 293 K, T_{sd} = 333 K, T_{ref} = 283 K, η_{WT} = 0.40, y = 5, T_r = 288 K, y = 10. Using these values in Equations 16 and 17 with a simple search yields the optimum T_s. Results are given in the next section.

Sample Results

A parametric study for a limited number of variables were carried out. It is possible to analyze the impact of all other variables of the model. Here, the impact of T_s on ΣE is analyzed first, which reveals the optimum T_s value. From Fig. 6, the optimum value of T_s is found to be 308 K (35°C) for an equipment with n = 1.3.

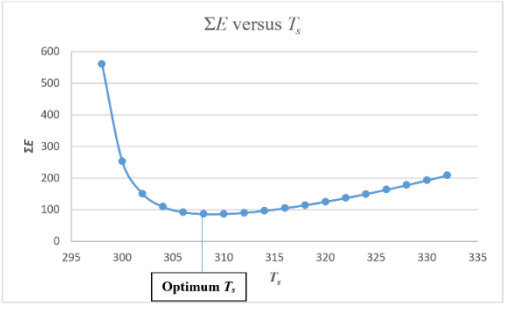


Fig. 6. Variation of the total exergy demand with T_s due to oversizing for the case study, n = 1.3

By varying the equipment exponent (n) between 1 and 1.4, the optimum temperature change has also been determined and shown in Fig. 7. Here, the minimum T_s is achieved for an equipment with n = 1, which

approximately corresponds to a radiant floor panel in heating. The variation of *PEXR*, which is the most important parameter in this model is maximum for n = 1 and T_s = 305 K, which is shown in Fig. 8.

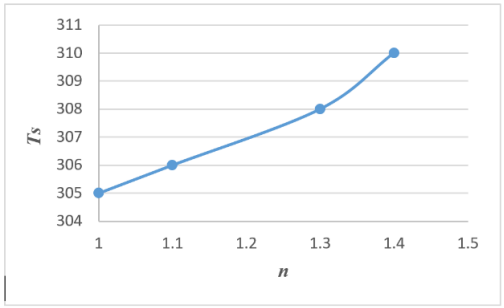


Fig. 7. Variation of the optimum temperature with equipment (n).

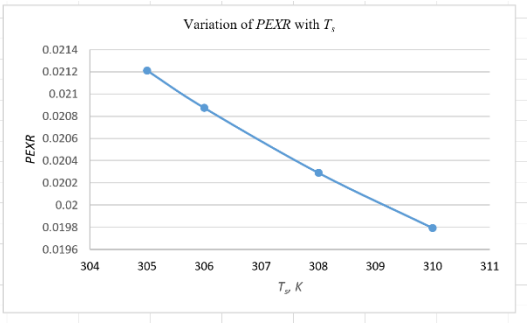


Fig. 8. Variation of *PEXR* with T_s

Discussion of results and conclusions

This study has presented a new model in order to provide a tool for optimizing the operating fluid temperature of a GSHP coupled to a wind turbine, which collectively serve an nZEXB building. In this model the energy and exergy interactions among the supply, storage, and demand are fully recognized and expressed in the objective function, all reduced to the operating temperature. Such a model, which encompasses thermal energy storage (TES) also enables to optimize the energy storage for wind energy in the form of heat, both in TES (daily or weekly) and in the ground (in cooling: seasonal storage). Yet it is debatable to whether converting electrical energy into heat just for comfort purposes is rational or not. From an exergy point of view, it is not quite rational. Instead, electricity must be used in more value adding applications in the built environment, especially in district energy systems, rather than space heating, which can be accomplished by waste heat or solar heat in the district. Even PVT systems may be mobilized, which add on power generation capacity over the wind turbine supply. The use of solar power generation system in general and wind generation system together assures a more daily sustained power supply, because there is about six hours of time difference between peak solar and peak wind periods. This study demonstrated that the new *PEXR* and COP_{EX} definitions, which bring a new performance metrication scheme, supplementing the First Law is instrumental in optimally hybridizing the renewable energy bundles and sustainable systems. If the absolute purpose is to use wind energy both for appliances, lighting etc. in nZEXB and also to heat and cool the building in a remote area (off the grid in particular) an optimum power usage split may be the first step to make the wind energy use in space comfort purposes more acceptable, provided that the operating fluid temperature is optimized. The case study presented herein shows that the terminal units for space heating must have their n values close to one (panel heating). Fig. 7. demonstrates this rule: if n increases above one, T_s increases, which reduces *PEXR* (Fig. 8). Other important design parameters are the pump efficiency, TES insulation, TES efficiency, *COP* of the heat pump, and the efficiency of the wind turbine. More these values are improved, then at an optimum split of wind power (X), GSHP use for comfort purposes in an nZEXB building may become reasonable and admissible. These arguments also apply for other renewable systems like cogeneration units running on biogas, absorption systems, and ORC [25]. the model developed in this study lets the designer to optimize any design variable to any sort of objective function selected.

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