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Optimizarea termodinamica a unui sistem hibrid inovativ care utilizeaza captatoare solare si pompa de caldura pentru date climatice zilnice

Thermodynamic optimization of a novel hybrid system using solar collector and heat pump based on daily weather data

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Rezumat. Lucrarea de fata prezinta un sistem termic hibrid inovativ compus din captatoare solare, robinet de reglaj proportional, rezervor de acumulare si pompa de caldura conectate hidraulic in serie, pentru a satisfice necesarul de caldura al unei cladiri. Procedura evalueaza valoarea temperaturii de iesire a robinetului de reglaj, pentru a obtine temperatura proiectata la iesirea captatorilor solari, astfel incat energia acumulata peste zi in interiorul rezervorului de acumulare, sa satisfaca necesarul energetic la vaporizatorul pompei de caldura pentru a putea genera la condensator toata energia necesara pentru incalzire. Ecuatiile matematice sunt astfel propuse, precum si un algoritm de functionare, pentru a calcula temperatura de iesire a robinetului de reglaj in schema hidraulica propusa. In capitolul de concluzii, sunt prezentate avantajele acestui algoritm, prin plotarea rezultatelor simulate in diverse conditii climatice si de necesar termic de incalzire.

Cuvinte cheie: sistem hibrid, captatoare solare, necesar termic

Abstract. The paper presents a novel hybrid heating system composed by solar collectors, mixing valve, heat storage tank and heat pump hydraulically connected in series, to satisfy the heating demand of a building. The procedure evaluates the temperature set-point value of the mixing valve, to obtain the design output temperature of solar collector, thereby the energy accumulated over the day inside storage tank should satisfy required heat pump evaporator energy to deliver at its condenser building demand. Mathematical equations are thus proposed and algorithm to establish mixing valve temperature set-point is presented. In the conclusion of the paper are described advantages of this algorithm based on plotted simulation results in several weather conditions and building heating demands.

Keywords: hybrid system, solar collectors, thermal requirement

1. Introduction

Solar energy usage as an alternatively to hydrocarbons start increase over past years. As a renewable resources as indirect energy source as solar radiation and indirect usage as environmental storage in air, rivers, lakes, underwater or ground, solar energy is part of our daily life. Responsible to drive all living creatures sun is used today for thermal comfort of humans during hot or cold season or for daily hot water preparation. Wide range of technologies to transfer or transform the solar energy to be used in heating or cooling demands are developed by today research and exploitation fields. Thermal solar collectors are ones must used equipment when solar radiation is the primary energy driven the system. In parallel, once vapor compression technology was developed, pressurized refrigerant properties are used by heat pumps to deliver thermal agents at a desired temperature potential useful to provide thermal comfort required by buildings.

By time, where proposed by research field, to be used a mix of technologies used in different thermal configurations, to satisfy the newly or rehabilitated buildings heating/cooling demand, to deliver a more efficient and sustainable thermal energy in a nature friendly mindset. This mix of technologies are commonly described today by the term hybrid systems, where at least two types of energies are used to satisfy building energy demand.

The mix of technologies helps decrease as much as possible equipment physical parameters and design requirements which can otherwise resulting in a more powerful equipment to satisfy the worst environment conditions demand but with high implementation cost and for a shortest period of time, when external weather conditions have very low apparition during the season. Thus, combining two or more technologies, can be drastically decreased the power of each equipment to deliver same amount of required energy to assure thermal comfort of final consumer.

2. System description

This paper presents a classical solar system composed by solar collectors and thermal storage tank for energy accumulation over the day. The novelty of this hydraulic diagram is the proportional mixing valve in the solar loop. With this connection, the output of solar collector thermal agent temperature can be controlled.

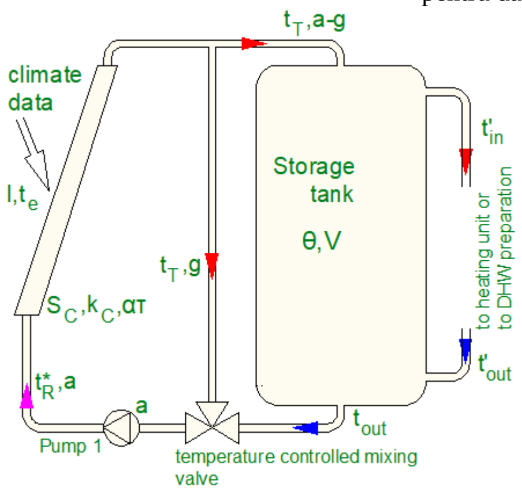

Fig 1

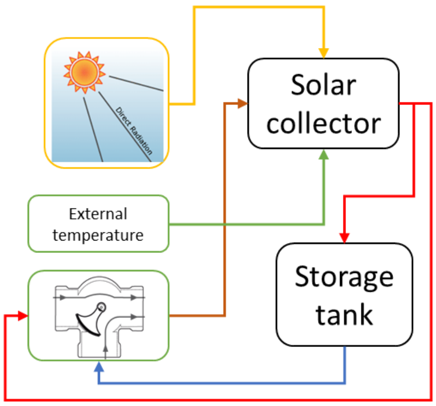

Fig 2

In figure 1 it is presented the hydraulic connection for solar loop in which solar collector input flowrate is constant over the time due to recirculating pump 1. Because the recirculating pump is installed after the proportional mixing valve, the flow in both valve inputs circuits is assured with only one pump. This is an advantage of the schematic which reduce the implementation and exploitation costs over the time. As can be seen on the notations, flow “a” is a constant value and g changes over time. Proportionality between the two inputs is assured by an electronic controller on which set temperature at output of mixing valve can be changed with values evaluated by mathematical model presented in this paper.

$$g = a \cdot \frac{t_R^* - t_{out}}{t_T - t_{out}} \tag{1}$$

Flow rate, g, in the high temperature input line of mixing valve is it directly evaluated knowing both inputs temperature of the proportional mixing valve. Thus, the hot input pipeline flow rate is evaluated with equation 1, where hot input line temperature, t_T, is the design temperature of solar collector output, cold line temperature, t_{out}, is the output temperature value from the bottom of the thermal energy storage tank, and proportional mixing valve temperature, t_R^*, being the set temperature for solar collector input.

Solar collector mathematical model design temperature is given by equation 2. Functionality of solar system witch collector is described by equivalent temperature, given by equation 4, and mechanical parameters evaluated with the thermal module, E, evaluated with equation 3.

$$t_T = t_E + (t_R^* - t_E) \cdot E \tag{2}$$

$$E = e^{-\frac{F' \cdot k_{\Sigma} \cdot S}{a \cdot \rho \cdot c}} \tag{3}$$

$$t_E = \frac{\alpha \cdot \tau}{k_c} \cdot I_{sol} + t_e \tag{4}$$

Setup temperature of mixing valve is the input temperature of solar collector, its value, external temperature value and solar radiation giving the inputs coefficients of solar system described by model in figure 2. For a given set of daily data, external temperature and solar radiation will assure the design temperature. This is possible by a specific input temperature, t_R^*, evaluated based on known data and design temperature at the solar collector output. Thus, rewriting equation 2 by equation 3, we can evaluate input temperature based on known external data and design temperature.

$$t_R^* = \frac{t_T - t_E * (1 - E)}{E} \tag{5}$$

Inside energy storage tank the temperature follows a thermocline profile with hot temperature at the top of the tank and cold temperature at the bottom of it. By design, hot temperature is design temperature, t_T, of solar collectors, and cold temperature, t_{out}, is the initial temperature before starting the solar collector functional period. The simulation of thermal storage tank is realized with a pre-trained neural network with 90 neurons on hidden layer, analyze presented before in the research. Condition of storage tank functionality is energy saturation inside tank should not be realized, thus, the output temperature of bottom volume of tank is constant over the time and cold input line of proportional mixing valve it easy to simulate.

Heat pump model consist of isentropic efficiency applied to Carnot efficiency between evaporation and condensing temperatures based on equation 6.

$$Q_{cd} = \eta_{el} \cdot f_{cd} \cdot \frac{t_{cd} + \Delta t_{cd} + 273.15}{t_{cd} - t_T + \Delta t_{cd} + \Delta t_{vp}} \cdot W_{comp} \tag{6}$$

Condensing equivalence factor, f_{cd}, is a form of Carnot efficiency evaluated by simulation and consist in polynomial regression calculation of its value. In the paper was presented a 3x2 regression coefficient matrix representation and two more matrixes, one used for evaporator temperature and the other for condensing temperature. Thus, in a mathematical simulation tool like Matlab, a matrix form of equation coefficients are convenient to be used. Compressor electrical engine efficiency, η_{el}, reduce compressor power applied to condensing Coefficient of Performance, COP_{cd}, based on manufacturer datasheets. Heat pump evaporator temperature is the designed temperature of solar collector temperature with its value conditions presented in this paper. Condensing and evaporation temperature differences, Δt_{cd} and Δt_{vp}, represent design values of this coefficient, to satisfy heat transfer demand of equipment. Generally, those values are comprised between 3 and 5 Celsius degrees.

Condensing power demand is evaluated for each external temperature conditions over the functioning period of the system. For each hourly weather temperature value, building demand is evaluated with global insulation coefficient, GN, evaluation of it can be found in National Evaluation Methodology MC001 or Romania. This coefficient is evaluated over the building for specific type of construction by height levels and distribution by surface of its volume. The thermal losses of the building are evaluated using temperature difference of internal building design temperature and each step exterior temperature. Therefore, the compressor power required by the system to deliver power demand at heat pump condenser is synthetized by equation 7.

$$W_{comp} = \frac{GN \cdot V_{build} \cdot (t_{i0} - t_e)}{\eta_{el} \cdot f_{cd} \cdot \frac{t_{cd} + \Delta t_{cd} + 273.15}{t_{cd} - t_T + \Delta t_{cd} + \Delta t_{vp}}} = \frac{P_{cd}}{COP_{cd}} \tag{7}$$

Condensing temperature value, t_{cd}, is a question of heating equipment of the building. If direct expansion internal units are used, as very efficient thermal system, with low refrigerant values required due to an improved thermal transfer coefficient because of its forced convection, the value of condensing temperature is at its minimum. Refrigerant condensing temperature for internal heating units with forced convection can be set in interval 29-32^{o}C.

In equation 7 now is possible to evaluate compressor electrical power at each external condition to satisfy internal comfort temperature or, as noted in this paper, design temperature, t_{i0}. For evaluation, hourly or daily average temperatures and solar radiation can be used. In this study, hourly typical meteorological year (TMY) data is used. In European area those data can be found easily and free of charge on European Commission, in EU science HUB website for photovoltaic geographical information system (PVGIS) [7]. Temperature and direct beam solar radiation are two input meteorological data used in solar collector evaluation and heat storage tank energy evaluation over the day. At each time step, heat storage tank charging power can be calculated with equation 8.

$$Q_{tank} = (a - g) \cdot \rho \cdot c \cdot (t_T - t_{out}) \tag{8}$$

The functionality of the system is assured when storage tank energy accumulation is sufficient to deliver heat pump evaporation energy required to satisfy building demand. Thus, in the calculation algorithm evaporation power is necessary. For this purpose, evaporation refrigerant state is evaluated with evaporation equivalence factor, electrical motor efficiency and compressor power given by equation (7). In equation 9,

$$Q_{evap} = \eta_{el} \cdot f_{vp} \cdot \frac{t_T - \Delta t_{vp} + 273.15}{t_{cd} - t_T + \Delta t_{cd} + \Delta t_{vp}} \cdot W_{comp} \tag{9}$$

Evaporation energy, E_{evap}, required over the day is equal with discharged storage tank energy, E_{tank}, over the day. Adding compressor energy used to recirculate refrigerant and assure its evaporation and condensing pressures, condensing energy to satisfy building demand is also easy to evaluate. Equations 10-14 are used to calculate all energies driven in the system.

$$E_{tank} = 10^{-3} \cdot \sum_{i=1}^{24} Q_{tank(i)} \cdot \Delta\tau_{(i)} \tag{10}$$

$$E_{evap} = 10^{-3} \cdot \sum_{i=1}^{24} Q_{evap(i)} \cdot \Delta\tau_{(i)} \tag{11}$$

$$E_{el} = 10^{-3} \cdot \sum_{i=1}^{24} W_{comp(i)} \cdot \Delta\tau_{(i)} \tag{12}$$

$$E_{build} = 10^{-3} \cdot \sum_{i=1}^{24} GN \cdot V_{build} \cdot (t_{i0} - t_{e(i)}) \cdot \Delta\tau_{(i)} \tag{13}$$

$$E_{cd} = 10^{-3} \cdot \sum_{i=1}^{24} COP_{cd(i)} \cdot W_{comp(i)} \cdot \Delta\tau_{(i)} \tag{14}$$

To satisfy functionality condition of the system, design temperature, t_T, should be evaluated. For this scope an iterative algorithm is lunched. Starting point of calculation is done by given a randomly initial design temperature and condition to increase or decrease it being the comparison of both evaporation and heat accumulated in the tank. The algorithm diagram is presented in chart of figure 3. Stop of the iteration is given by selecting desired error, e_{rr}, this error being the accuracy between storage tank energy and evaporator energy.

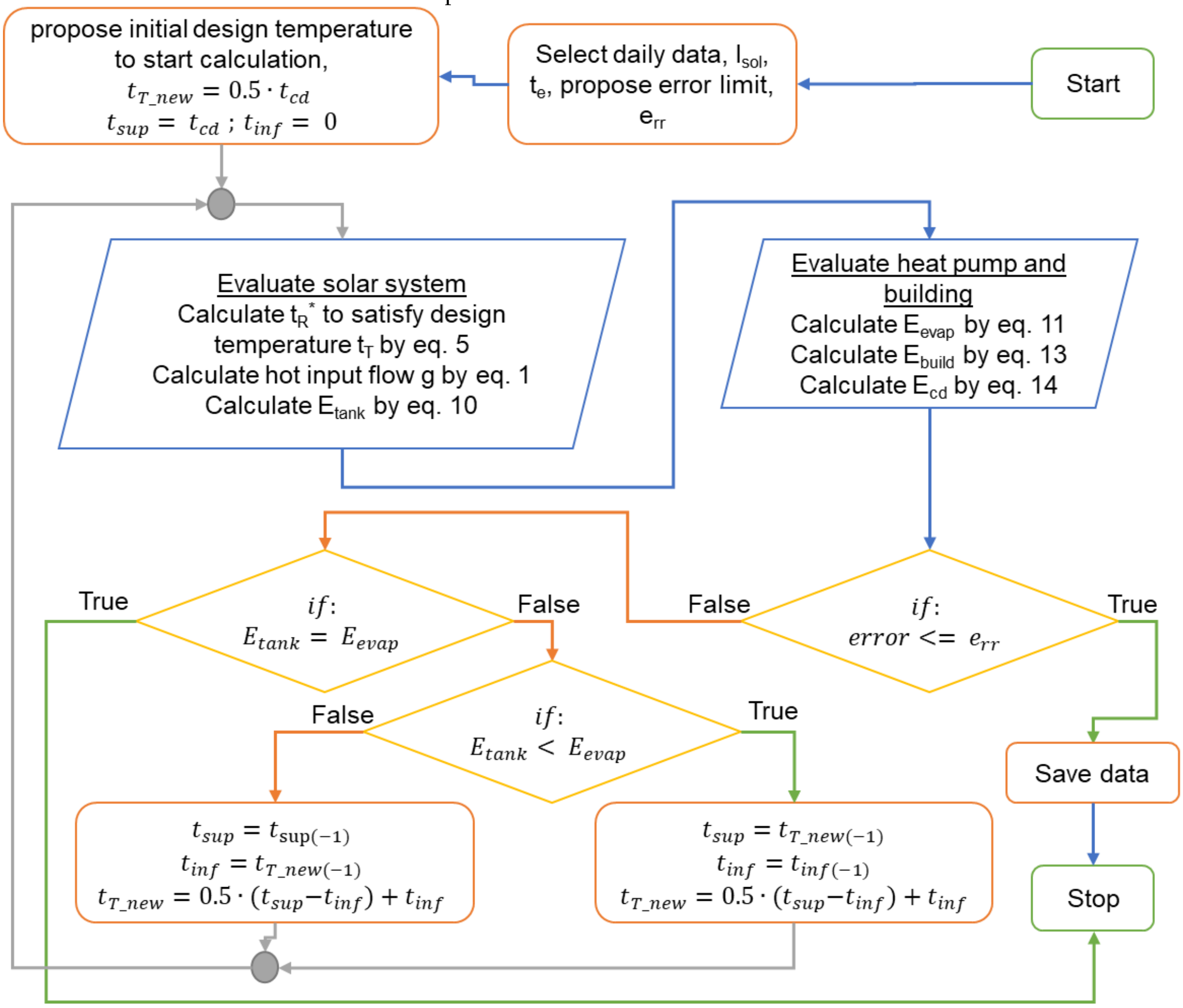

Fig 3

Algorithm state chart follows the programming tools logic, when a variable is updated by an equation using its old value, noted in the chart with index (-1). Thus, in the evaluation of the new design temperature, t_{T_new}, is updated by the average value in the middle interval of last used temperature and actual value. Depending on difference between tank energy and evaporation energy, the required design temperature should be greater comparing with old value if tank energy is less than evaporation demand and vice-versa when stored energy is higher. This state is true because once design temperature increases, for same weather conditions, storage tank energy decrease. In same time, evaporation power is increased, caused by a thermal agent with higher thermal potential.

The algorithm will return by calculation the proper design temperature to reach the maximum COP of the system when in the storage tank is stored sufficient energy for evaporation of refrigerant. Building demand clearly should be satisfied by condensing power, thus being assured the functionality of system at required external conditions. By addition to this, algorithm can be used to evaluate the system power design of each component to respond in the worst conditions required by final customer. This methodology evaluates as worst condition the minimal equivalent temperature and not the lowest external temperature. Thus, a pre-evaluation with data mining should be done.

3. System simulation

Using MATLAB software to evaluate a proposed system, a simulation session has been released. 2 March TMY Bucharest weather data is presented in figure 4.

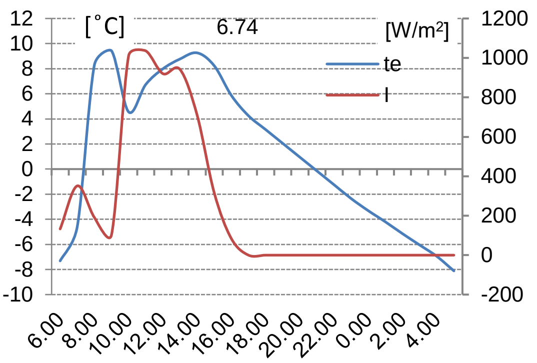

Fig 4

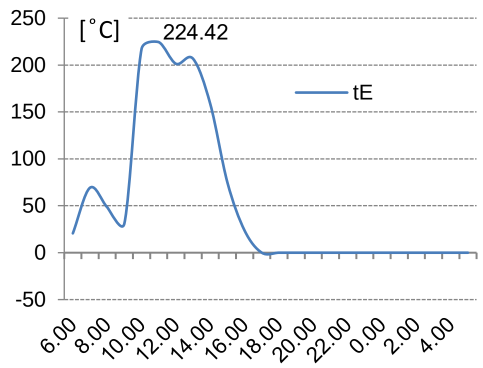

Fig 5

Associated equivalent temperature data is plotted on figure 5. This temperature is evaluated over daily hours when solar radiation is available and can be used in equation 4. As can be seen on the graph, equivalent temperature reaches high values and thermal potential of external weather being transferred to thermal agent flowing into solar system. Selected day is characterized by medium exterior temperatures and only a small interval with good solar radiation potential from 10:00am to 2:00pm, opportunity for analyzed system to deep charge of solar tank. This can be easily observed on figure 7, at which thermal agent flow with design temperature potential, transfer heat from collectors to tank.

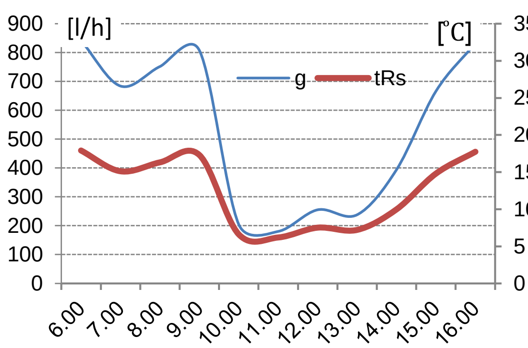

Fig 6

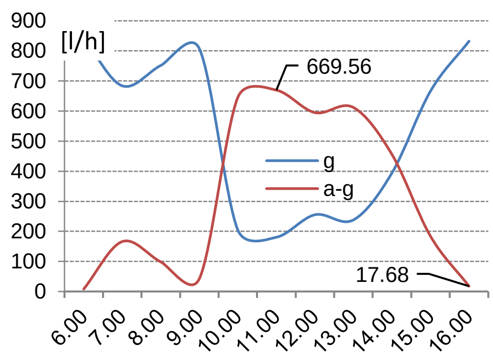

Fig 7

With a minimum 17.68 liters per hour, the system is capable to deliver energy at required design temperature in the worst external weather conditions. Figure 4 flow rate is evaluated for the proper heat tank energy, equal to evaporation demand based on state-chart of calculation algorithm in figure 3. In same time, the flow in mixing valve high potential line input increase, to deliver required temperature to obtain a thermal agent at solar collector output at the required design temperature based on procedure of this paper.

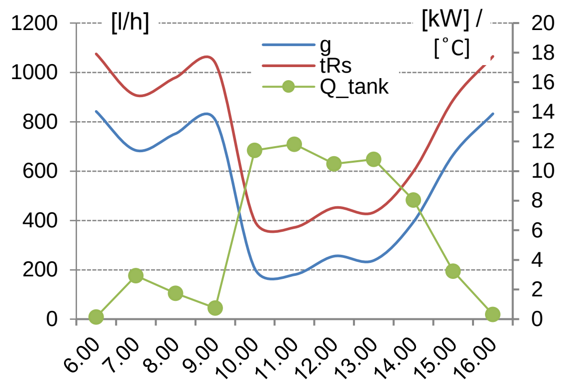

Fig 8

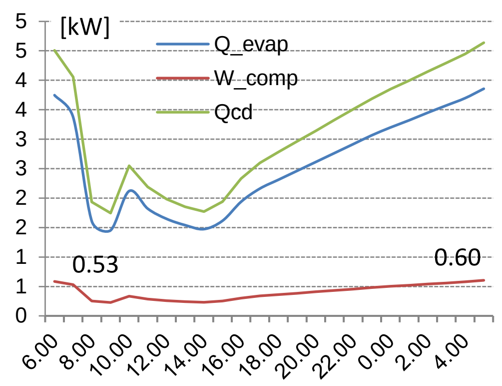

Fig 9

Return temperature, from equation 5 is proportional with equivalent temperature and external weather condition. Based on its value, dependency with flow of mixing valve high temperature input can be observed on graph of figure 8. As can be seen, its minimum value requires less flow from the output plug of solar collector. Based on this behavior, once the system function in good external weather, the solar loop discharged tank power is at its maximum. For the higher external conditions, plotted in figure 9, at 09:00am and 3:00pm, energy demand of the building is at its minimum, causing a less energy consumption of compressor electrical motor and evaporation. Renewable energy of this system can be associated to evaporation energy after extraction of electrical energy and all auxiliaries. Based on this paper equations, associated energies are plotted on figure 10, in kWh for the entire day of 2 March.

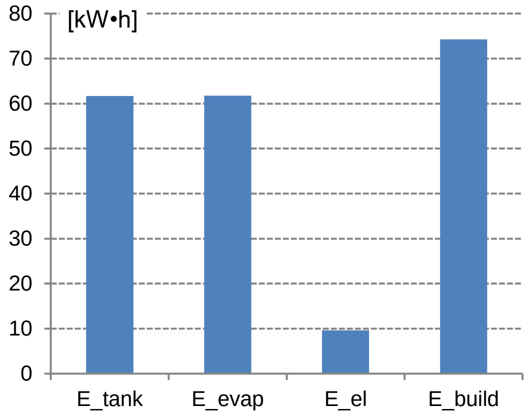

Fig 10

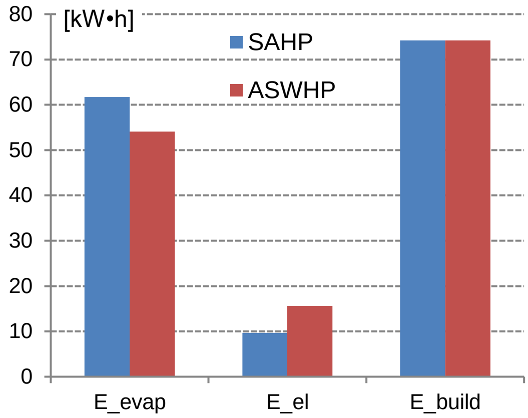

Fig 11

For the design temperature evaluated by algorithm from state-chart in figure 3 and by comparison with an air source heat pump, the advantages of solar assisted heat pump, plotted in red columns can be observed on figure 11. For the same building demand, SAHP deliver more evaporation energy for less electrical energy consumption.

Convergence of algorithm is done fast based on capability of Matlab calculation. Thus, after 10 steps design temperature is 18.1°C with an associated error less than 8.5Wh between evaporation and tank storage energy. The system analyzed is composed by 17 m^2 solar area, 300 m^3 building with 0.55 $W/m^3/K$ global insulation coefficient, and compressor power of heat pump plotted in figure 9 with its maximum at 0.6kW.

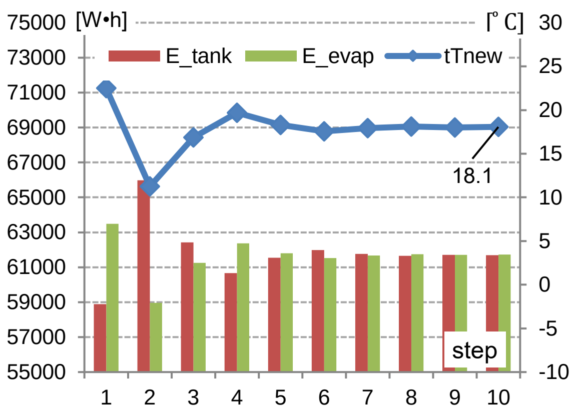

Fig 12

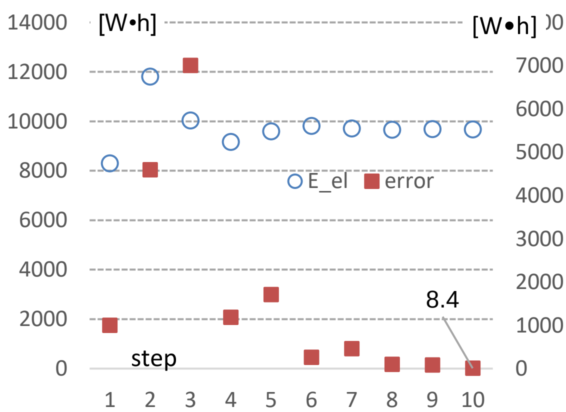

Fig 13

Electrical power of heat pump compressor corresponds to 2 March external conditions, for higher energy demand, causing higher compressor electrical power. Energies plotted in figure 2 reveal capacity of the proposed system to deliver more tank energy as can be seen in step2 for example but at this condition, evaporation temperature of tank volume is not at required level, causing a gap between required and delivered energy. This will affect the overall building thermal comfort.

The system designed as describe in this paper is capable to deliver energy demand using a series connection of solar assisted heat-pump. In case design temperature or output of solar collector is not at the required value, heat pump cannot deliver required energy and an auxiliary source being necessary.

Nomenclature:

t_{out} – thermal energy storage tank bottom output temperature, ^oC;
t_R^* – solar colector input temperature, ^oC;
t_T – solar collector output design temperature, ^oC;
t_{T_old} – last temperature used for evaluation, ^oC;
t_{T_new} – new temperature used for evaluation, ^oC;
t_{cd} – condensing temperature, ^oC;
t_{vp} – evaporator temperature, ^oC;
t_{i0} – design internal temperature, ^oC;
t_e – external temperature, ^oC;
t_E – equivalent temperature, ^oC;
I_{sol} – solar radiation, W/m^2;
S – solar collector equivalent surface, m^2;
k_C – thermal loss global coefficient, W/m^2K;
k_Σ – thermal transfer coefficient of inside tank coil, unitary in this paper due to open loop, W/m^2K;
a – solar collector flow rate, m^3/s;
ρ – thermal agent density, kg/m^3;
c – thermal agent specific heat, J/kgK;
α – solar radiation absorption coefficient of collector surface, -;
τ – transparency coefficient of solar collector cover glass, -;
$\Delta\tau_{(i)}$ – step time, h;
F’ – equivalence solar surface coefficient, - ;
E – solar collector thermal module, - ;
V_{build} – building volume, m^3;
GN – global insulation coefficient, W/m^3K;
NTU_C – numarul de unitati de transfer termic aferent suprafetei de captare solara, -;

NTU_S – numarul de unitati de transfer termic aferent schimbatorului de caldura al buclei solare, -;
Q_{cd} – condensing power, W,
Q_{evap} – evaporation power, W,
Q_{tank} – heat storage tank charging power, W,
f_{cd} – condensing equivalence factor, -,
f_{vp} – evaporation equivalence factor, -,
η_{el} – compressor engine electrical efficiency, -,
W_{comp} – compressor electrical power consumption, W,
E_{tank} – solar storage tank energy, kWh ;
E_{evap} – evaporation energy, kWh ;
E_{el} – electrical energy, kWh ;
E_{cd} – condensing energy, kWh ;
E_{build} – building demand energy, kWh ;
Δt_{vp} – evaporator temperature difference, ^{o}C,
Δt_{cd} – condenser temperature difference, ^{o}C,
SAHP – Solar Assisted Heat Pump,
ASHP – Air Source to Heat Pump,
WSHP – Water Source Heat Pump.

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