Numerical approach regarding functional and design optimization for a residential building heating system composed by heat pump and auxiliary

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Abstract. In this paper is presented the physical model and mathematical approach which describe the equation system used in system calibration and design optimization. The system proposed for study is built from heat pump, for energy demand delivery, together with auxiliary heating source to face in all low temperature days, when heat pump work at maximum load but the required demand by the building is higher. The paper present few of the common used systems in market for which the mathematical equation system will be proposed to come in help designers for in simulation and cost optimization. Simulation of proposed design is realized and results are delivered. The system construction, is optimized by comparison study of design and simulation data for each system type proposed. The comparison study is used for cost estimation of system and energy balance.

1 Introduction

Building heating and cooling research domain encounter no limits today, being a used subject in design, simulation or experimental study or together [1-4]. Methodologies to evaluate [5] and optimise [6] exist today. Agents for heating purpose are prepared by a wide diversity of individual heating equipment or hybrid systems [7-9]. Even the subject make part of multiple numbers of studies, the domain give more and more interesting direction for research and design. Newly types of heating equipment are also a research pillar due to high level of innovation and uncertainness when are designed by engineers. More and more tools for this type of design are available today to engineering and research. Basically, empiric technics in digital instruments, are mathematics and physics modelling of real equipment. Mechanical engineering has developed a wide range of equipment modelling which today results in a high efficiency heating equipment components with high accurate mathematical models. Comparative studies are done, results and discussions give a good direction in terms of design and optimisation. For those studies, geothermal heat pumps are driven the lower running costs in Europe, or, secondly, diesel oil fired boilers, used for electric heating [10]. On the other hand, an interesting study show the increasing of 33% in primary energy usage, respectively, 26% greenhouse gas emissions of geothermal heat pump compared with district heating system [11]. For heat pumps, experimental researches were driven to see the performances of different types of heat pumps. For two twin houses, an experimental investigation was launched, between a geothermal and air to water heat pumps. The results shown relatively constant performances for geothermal heat pump comparing air to water [12]. In practice, are found more and more complex heating systems, which are composed by at least two types of heating equipment. Hybrid systems, for heating, are generally built for delivery the necessary demand in worst conditions, or to improve the old equipment. Also, to decrease the overall primary heating demand and increase performances, hybrid systems are suitable to candidate. Therefore, will give the opportunity, to research field, to develop new methods of evaluation or to use separate methods for each systems and link them, by correlation. Literature, is giving numerous types of strategies to be used for evaluation regarding performances or optimisation of the heat pump. Wet bulb temperature and entering output temperature in evaporator, as input parameters for regressive curves are used to simulate the behaviour of the heat pump with EnergyPlus [13]. In this paper, the condensing waste heat recovery was evaluated by model. Energy savings encountered values between 47% and 64%, in cooling, comparing system with no heat recovery. Bi-Linear interpolation, of performances data, were used as modelling technique to evaluate heating and cooling capacities, heating and cooling electrical power demands, or heating and cooling coefficients of performance, using finite element design of a ground source heat pump (GSHP) boreholes transfer elements. Results are plotted for installation in London, for an building with total area of 10,500 m^2 [14]. Results of simulation shows decreasing in temperature of deep ground elements by a period of 30 years, which reveal the susceptibility of underground heat to decrease by time.

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Air source heat pump (ASHP) compressor performances, drive in general, the global performances of the heat pump. A parametric study, concentrate on compressor evaluation for efficiency, tacking into account pressure ratio, was done to evaluate dependency [15]. Cycle parameters are extracted from refrigerant state diagrams. In this study, it was shown an interesting influence of compressor isentropic efficiency by refrigerant state pressures, corresponding to temperatures at evaporator and condenser units. Two refrigerants were considered and tree types of compressors were simulated. Isentropic efficiency interval were found to be 0.53 to 0.77, corresponding to pressure ratios of 7.5 respectively 2.5 in case of R410 refrigerant with a scroll compressor.

In case of a water source heat pump (WSHP), with an exterior lake as heat source, an exponential dependent heat pump COPs, by water temperature, were used for heating and cooling demands [16]. This model consist by knowing the regressive coefficients of historical data, from experiments. Equations feet the curves of COP in heating and cooling, and electrical power demand has been evaluated with. Some of results conduct to an interval of 6˚C to 10˚C of source water temperature, in heating season, respectively 22˚C to 31˚C for cooling season.

By simulation, for an ASHP installed in Beijing, China, a maximum electrical power of 800W was measured for a maximum heating load of 30kW when outside temperature is -15˚C, in case of a residential building [17]. For those values, the electrical power of compressor is the 2.7% of maximum heating load demands.

By sizing, in case of 42.631kW maximum load power for domestic hot water production, was used a heat pump with maximum electrical power equals 8.2kW [20]. This conduct to 19.24% percentage of heat pump electrical power comparing maximum heat demand. Hot water demand is based calculated by a 10˚C inlet system temperature with 60˚C as set temperature value of daily hot water.

For cooling purpose, two WSHPs used as water to water configuration, register a maximum electrical power of 8.00 kW to 9.00 kW. For the building where system is installed, two heat pumps provide 32,8kW of the cooling needs. Heat pump return the heat into a recirculated water from a ground loop using heat-exchangers. The study was done for Sydney weather. The heat pumps maximum instant power consumption, represent 24.3% - 27.4% of maximum building cooling heat demand [19]. Data delivered by study was estimated with DesignBuilder software, used in modelling of energy demands, heating and cooling space and systems modelling.

For different system configurations and multiple weather conditions, the heat pumps electrical power consumption reported to maximum heat load, in heating or cooling, is not overpass a 30% for case studies presented.

This paper will propose a mathematical model to be used for energy demands evaluation of building heating system. Optimisation of heat pumps systems, based on energy demand and heat pump configuration will be made using model parameters evaluation. Two types of heat pump, ASHP and WSHP are evaluated for 5 different cities of Romania, based on ISO weather data for this country through Meteonorm database.

2 Mathematical model

Water source heat pump considered for modelling, is based on relative quasi-constant temperature of underground or surface lakes water. Heat pump extract the heat from the water using different types of equipment such heat-exchangers or indirect immersed evaporators with respect of refrigerant pressure for unfreezing purpose. A schematic of the system configuration, for building heating demands, and water source based on lake water is presented in figure 1.

The system considered take into account the all elements, from the heat source to delivery plugs of the heating agent. The building is generally presented to reveal the internal design temperature set point.

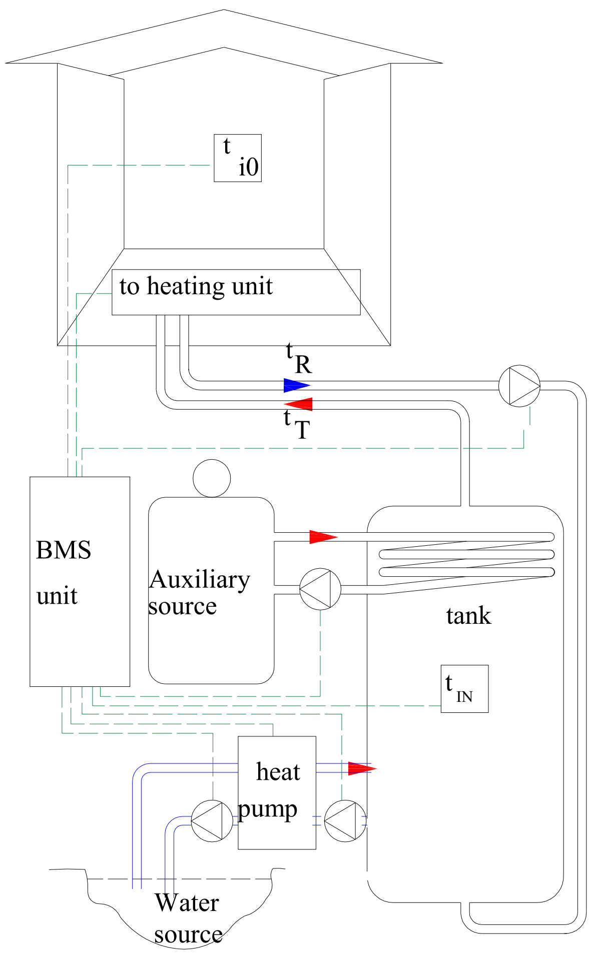

Fig. 1. WSHP system model considered.

For heating, all system components are controlled by the building management systems (BMS), to be more facile to implement the algorithm of function, regarding system, in case presented modelling is used. Water source is a surface lake with high enough water volume to consider a constant temperature over the period for performance evaluation. The auxiliary heat source is used for the additional heating energy necessary in worst outside environment considered for this study. Water tank is required by system to use his internal temperature as a reference for heat pump working temperatures at condenser side. All losses are not evaluated and are considered contained into heat demand of the building. For internally building heating equipment, a circulation pump will drive the heat agent from the tank, with right temperatures, evaluated through presented methodology.

Tank temperature, is always a temperature depending by turn and return temperatures of heating agent. By correlation with external worst scenario temperature, t_{e0}, and interior set temperature, t_{i0}, reference turn and return temperatures values (t_{T0} and t_{R0}), corresponding temperatures of the agent are calculated. For a specific exterior temperature (t_e), turn (t_T) and return (t_R) temperatures, are correlated with t_{T0} and t_{R0}, having both expression (1). Internal tank temperature, t_{IN}, is considered into this paper being the average of both, t_T and t_R temperatures.

$$\mathbf{t_T}=t_{i0}\cdot(t_{T0}-t_{e0})/(t_{i0}-t_{e0})-\mathbf{t_e}\cdot(t_{T0}-t_{i0})/(t_{i0}-t_{e0})$$
$$\mathbf{t_R}=t_{i0}\cdot(t_{R0}-t_{e0})/(t_{i0}-t_{e0})-\mathbf{t_e}\cdot(t_{R0}-t_{i0})/(t_{i0}-t_{e0}) \quad (1)$$

Condensing temperature, θ_{CD}, at which refrigerant will discharge the latent heat, is tank temperature dependant and its value is the average value between t_{IN} and return temperature of the heating agent expressed by (2).

$$\mathbf{\theta_{CD}}=t_R+0.5\cdot(t_{T0}-t_{R0})\cdot(t_{i0}-\mathbf{t_e})/(t_{io}-t_{e0}) \quad (2)$$

Reporting electrical power consumption of heat pump compressor (W_{EL}) to maximum heating power demand (Q_{b_0}), in worst case, of entire building, we can establish the correlation between both terms by (3) [5], were electrical factor (f_{el}) is an dimensionless sub-unitary parameter. Also, the maximum heating demand is reported to overall heat transfer rate (H), of the building in the worst case and given by (4) [5].

$$W_{EL}=f_{el}\cdot Q_{b\_0} \quad (3)$$

$$Q_{b\_0}=H\cdot(t_{i0}-t_{e0}) \quad (4)$$

Expression (4) required, for modelling, the overall transfer rate of entire building which is evaluated or known by the beginning, with the desired method. For each external temperature, the heating demand (Q_b) is linearly dependant, and with expression (5), will be evaluated each heating demand.

$$\mathbf{Q_b}=Q_{b\_0}\cdot(t_{i0}-\mathbf{t_e})/(t_{io}-t_{e0}) \quad (5)$$

In this paper, electrical factor and heat agent reference temperatures will be evaluated, to understand the matter them change the behaviour of the system, how the heating delivered by heat pump satisfy the building demand and which part of the energy required will remain in charge of the auxiliary source, to be delivered.

Heat pump behaviour is based on required condensing power, which direct impacts the electrical power at compressor inlet plugs. Condensing power required, together with isentropic efficiency of compressor at the refrigerant pressure ratio and temperature difference between evaporator and condensing environments, is given by expression (6).

$$\mathbf{Q_{CD}}=\eta_{is}\cdot W_{EL}\cdot(\theta_{CD}+\Delta_{CD}+273.15)/(\theta_{CD}-\theta_{VP}+\Delta_{CD}+\Delta_{VP}) \quad (6)$$

In equation (6), evaporator and condensing temperatures of environments, are noted θ_{CD} and θ_{VP}, refrigerant state condensing and evaporation temperatures are different to be able to transfer the heat from, respectively to refrigerant. The refrigerant state temperatures are different with a small difference in both condensing and evaporation and noted by Δ_{CD} and Δ_{VP}. The compressor work with an isentropic efficiency, which denote that entire electrical power consumption is not transferred totally to condensing power. By different heat pumps topologies, evaporator and condensing temperatures together with temperatures difference of refrigerant states, values can be different.

Generally for ASHP refrigerant state, is definitively dependant of air temperature were evaporator (in heating) or condenser (in cooling) transfer the heat from, respectively to air.

In case of WSHP, the water source temperature is driving the refrigerant state parameters of evaporator (in heating) or condenser (in cooling) both being also dependant of equipment used, as example immersion coils, water to water or air to water heat exchangers.

In both situations, temperatures differences will conduct to evaporator or condenser efficiency.

By correlation, evaporator and condenser equipment efficiency, will conduct to an overall influence of entire heat pump system. This paper will also evaluate how the power given by heat pump is condenser and evaporator temperature difference dependant, for the targeted condensing temperature (in heating). This evaluation will conduct to appropriate values of equipment selection for desired heating power provided by heat pump by referring to auxiliary source.

Evaluation of energy, for entire heating period, is based on summing of power demand at each outside temperature, multiplied by number of hours at each temperature state. Each temperature power demand is evaluated with equation (5) and for external temperature will be used ISO weather data for 5 cities in Romania.

For ASHP heat pump will be used same entry data, for a similar system, presented in figure 2 were the only change is evaporator equipment, able to extract heat from outside environment, which have temperature corresponding external weather data. Regarding condensing temperature, this have the same calculation for the same reference building, evaluated through overall transfer heat rate.

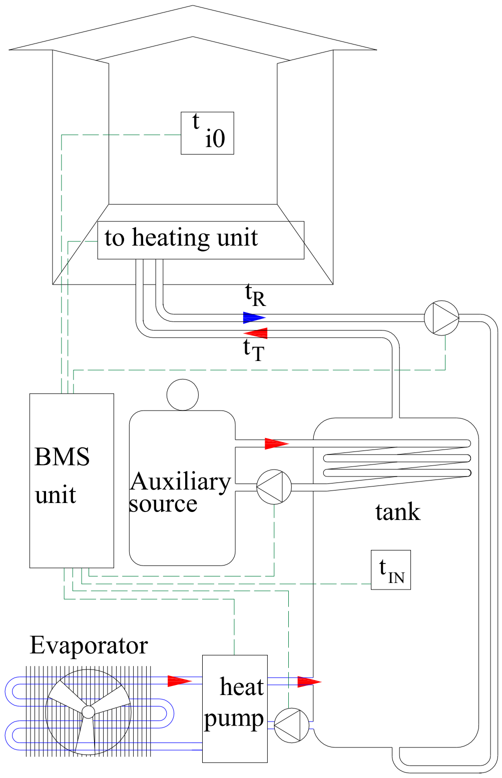

Fig. 2. ASHP system model considered.

Energy needed from auxiliary source, is nothing else than additional energy required by the heating system, to deliver the necessary building heating energy (E_{req}), given by equation (7).

To evaluate both, provided heat pump energy, by equation (8) and auxiliary source by equation (9), will be used hourly data given by ISO weather format through Meteonorm standard tool for climate database. By interpolation between weather stations near interest cities, weather data are able to be used as reference in research about energy evaluation.

$$\mathbf{E_{req}} = \Sigma \mathbf{Q_{b\_t}} \cdot \mathbf{n_{h\_t}} \cdot 10^{-3} \tag{7}$$

$$\mathbf{E_{HP}} = \Sigma \mathbf{Q_{CD\_t}} \cdot \mathbf{n_{h\_HP\_t}} \cdot 10^{-3} \tag{8}$$

$$\mathbf{E_{aux}} = \mathbf{E_{req}} - \mathbf{E_{HP}} \tag{9}$$

Where, n_{h_t} is number of hours at temperature t_e, Q_{b_t} and Q_{CD_t} are building heating demand for external temperature t_e, respectively heat pump delivered heating power for evaporator temperature θ_{VP}.

For ASHP, evaporator environment temperature is external temperature t_e, but for WSHP, the evaporator temperature is water temperature, considered constant during heating season, in this study.

By equation (7) and (8), the energy evaluation return values in kWh. Referring to a surface of building, if needed, the results make possible establishment of the necessary heating energy for square meter.

3 Results and discussions

Optimisation of heat pump for building heating demand was lunched based on electrical factor from equation (3) and for heating power (10). Evaluation were driven to see dependence of heating provided by heat pump, over a fixed range of electrical factor, f_{el}, and t_e, external temperature.

$$\mathbf{Q_{CD\ aw}} = \eta_{is} \cdot \mathbf{W_{EL}} \cdot (\theta_{CD} + \Delta_{CD} + 273.15)/(\theta_{CD} - \mathbf{t_e} + \Delta_{CD} + \Delta_{VP})$$
$$\mathbf{Q_{CD\ ww}} = \eta_{is} \cdot \mathbf{W_{EL}} \cdot (\theta_{CD} + \Delta_{CD} + 273.15)/(\theta_{CD} - \mathbf{t_w} + \Delta_{CD} + \Delta_{VP}) \tag{10}$$

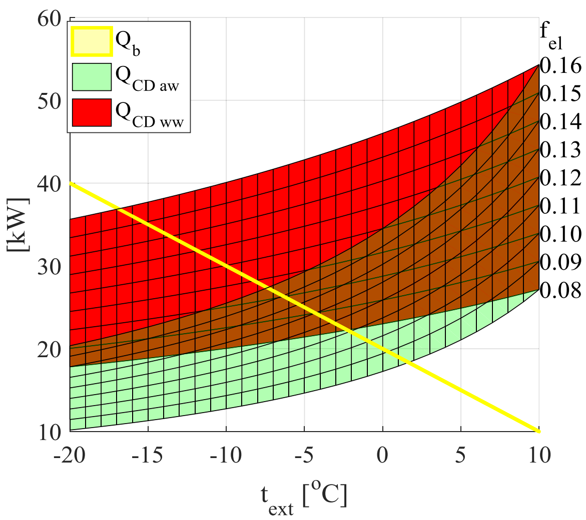

Fig. 3. Building heating demand, ASHP and WSHP condenser power for external temperature and electrical factor variation.

In equation (10) both heating power given by ASHP and WSHP are calculated. In this equation, $Q_{CD\ aw}$ and $Q_{CD\ ww}$ represent ASHP, respectively WSHP provided power at condenser side.

For an interval of 0.08 to 0.16 of electrical factor and for the interval of -20˚C to 10˚C of external temperature, can be seen in Figure 3 at which external temperature and for what electrical factor the building energy demand is satisfied. Is visible in the diagram that in case of ASHP, comparing WSHP, the heating delivered by heat pump it is drastically decreased; for same conditions, the ASHP cannot sustain entire building heating demand below -7.5˚C, for an electrical factor of 0.16 or 16% from total heating demand.

Evaporator and condenser temperature difference is considered, in all paper evaluation, 5˚C. A big influence in the heat pump performances is the value of temperature of water at condenser side, make it water dependence behaviour [21].

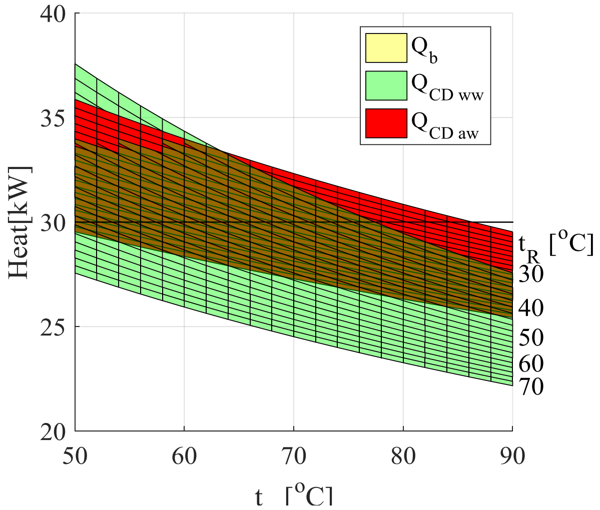

Fig. 4. Building heating demand, ASHP and WSP condenser power for a specific range of turn and return temperatures.

Also by modelling, the condenser water temperature at output plug, is directly influencing the heating power of the heat pump. To reveal dependence of this paper model, into heat pump performances, was conducted an evaluation based on same reference building but for a constant heating demand at -10˚C external temperature.

Heating demand of the building, for a constant value, when is plotted together with delivered heating power of condensers in both situations, ASHP respectively WSHP, reveal the influence of turn and return temperatures range in heat pump performance. In Figure 4 is shown that for 0.15 electrical factor of WSHP and -10˚C external temperature, heat pump can deliver all heating demand if turn temperature below 65˚C respectively 45˚C for return temperature. ASHP for this evaluation were conducted in case electrical power is 1.5 times electrical power of the WSHP or 0.225 electrical factor.

Isentropic efficiency of the compressor is an equipment based parameter, which conduct to overall heat pump performance. By refrigerant type heat pump performances can register different performances [21]. The equipment behave differently by type of compressors and components used for evaporation, condenser, or expansion valve. Normally, all those will affect the isentropic efficiency and to see his influence in heat pump performances, an optimisation evaluation was lunched. For this, isentropic efficiency for different external temperature, when condensing temperature is in a fix range, is plotted in Figure 5. To compare data, heating building demand over external temperature is available on graph. For various isentropic efficiency, ASHP respectively WSHP heating power, delivered at their condenser units, are plotted to show in which conditions heat pump is able to deliver required power for heating of the building. Electrical factor used is 0.11 and same reference building with overall heat transfer rate 1000W/K. Evaporator temperature for ASHP represent external temperature and refrigerant state have 5˚C temperature difference.

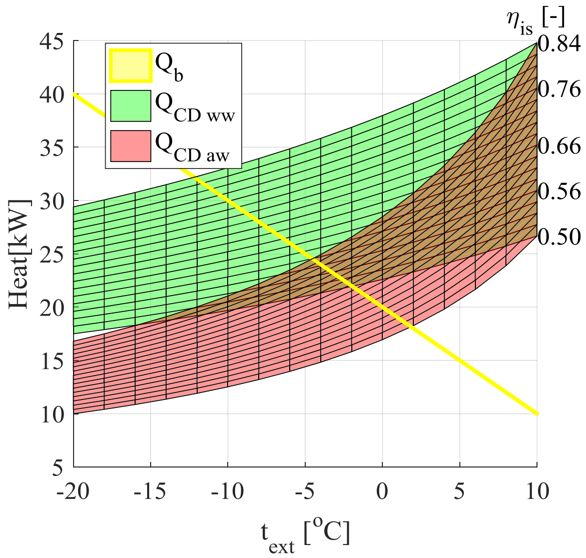

Fig. 5. Building heating demand, ASHP and WSHP condenser power for a specific range of external temperature and isentropic efficiency.

Regarding condenser, temperature difference, in both heat pumps topologies, is also 5˚C. For WSHP, evaporation environment temperature is constant 10˚C, in all range of external temperature.

Figure 5 show that, for a specific isentropic efficiency of the heat pump compressor, condenser delivered heat to the reference building can sustain or not the required heating. Based on that, can be correlated isentropic efficiency with building demand and heat pump condenser heating. For isentropic efficiency, below 0.6, WSHP can deliver required heat of the building over -6˚C, respectively -0.5˚C for ASHP. For results of Figure 5, values of heat agent are, by design, t_T/t_R = 50/30, temperature difference each, condenser and evaporator 5˚C, and internal design and exterior temperatures reference design, 20˚C respectively -20˚C.

4 Case study

Romania have various climatic areas in terms of heating periods and worst case external temperature with warm environment at Black Sea or cold environment on mountains. Heating period is different for each city selected in this study, due to external temperature average, when its value is below 10˚C. In Table 1 are defined heating periods for all 5 cities selected.

For selected cities, was used same configuration of the heat pump equipment in terms of working parameters. In right down area of Figure 6, are described system parameters, tacked into consideration for model entry data. The five cities selected, are from 5 types of weather areas, in term of outside average temperature. From the Table 1 can be noted that for Constanta, comparing with Miercurea Ciuc, the heating period start with a delay of 26 days, globally, this will be seen into energy demand of the building, Q_b. On graphs can be seen that also the maximum required heating demand of the building is decreasing for specific cities from warm areas like Slatina and Constanta.

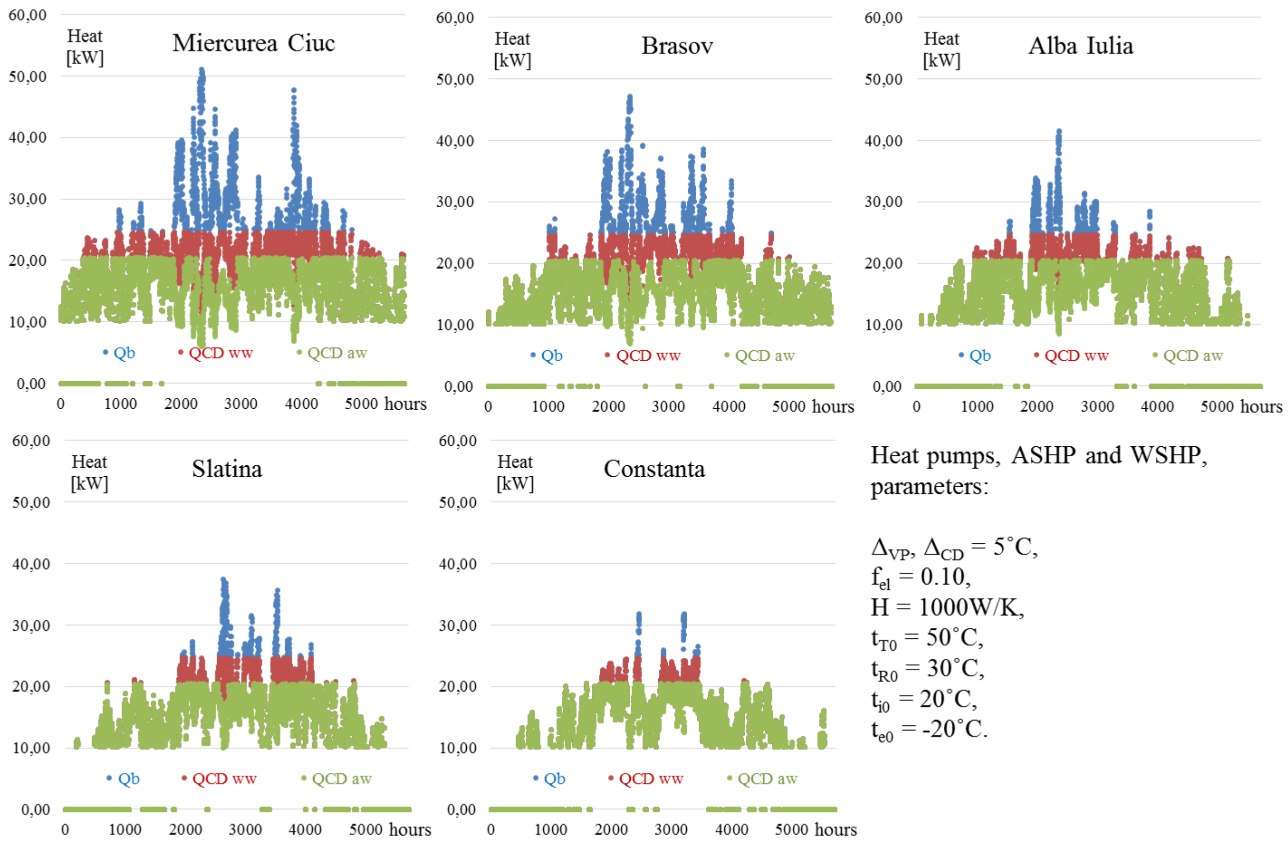

Figure 6. Building heating demand, ASHP and WSHP condenser power for 5 cities of Romania.

For Miercurea Ciuc, heating demand, in deep cold season, don't have off periods, comparing with warmly environment, as Slatina and Constanta, for which exist hourly specific frames, were heating is not required. A maximum building heating power requirement is obtained for Miercurea Ciuc, with a value of 51.1kW. In the opposite side, in Constanta, maximum required heating power represent 31.90kW, with 37% less comparing Miercurea Ciuc. For all 5 cities, can be noted that, the same compressor configuration, in term of electrical factor, maximum delivered power, at condenser side of the heat pumps, in case WSHP is bigger comparing ASHP. A correlation between both, ASHP and WSHP should be driven, or in case ASHP the electrical factor to be increased. In the worst conditions, ASHP work with air temperatures, were values of condenser power correspond with temperatures around -20°C, for which, the performances are lower comparing WSHP, were at same external temperatures, evaporation temperature is at the water source level. The heat pumps can sustain, in warm conditions, a maximum power bigger than building requirement. For specific outside temperatures, condenser heating power is bigger due to a low requirement at building side, or reduced losses, comparing with cold periods. Because of configuration, and for energy evaluation, the amount of power above requirement are not necessary to be tacked into account. Heat pumps generally work when heat is required.

A carefully selection of heat pumps should be done, for cold environment. If we look inside Figure 7, only 91.5MWh from the required 105MWh of the building, in case water to water heat pump, for Miercurea Ciuc. This represent 87.1%, from the total heating energy demand, according Figure 8.

Table 1. 5 cities heating periods.

	Start of heating	End of heating
Miercurea Ciuc	19.09.2017	14.05.2018
Brasov	27.09.2017	10.05.2018
Alba Iulia	11.10.2017	24.04.2018
Slatina	14.10.2017	13.04.2018
Constanta	15.10.2017	09.04.2018

For the ASHP, the situation is worse than WSHP. Only 73.7% of the building energy demand, E_{req}, or 77.4MWh from the overall requirement. Evaluation of the energy, for ASHP, respectively WSHP are made by equations (7) and (8). In case of auxiliary energy, obtained from the auxiliary source, only 0.4MWh is required for Constanta city, were is encountered a warmly environment. Contrary, for Miercurea Ciuc, in proposed conditions and heat pumps parameters, is required much more energy. In case WSHP, is required from auxiliary source, 13.5MWh, respectively 27.6MWh for ASHP. More suitable in case a WSHP selection, is due to its potential to deliver, for this city, enough heat comparing ASHP, for which is required double auxiliary energy, comparing water based heat pumps.

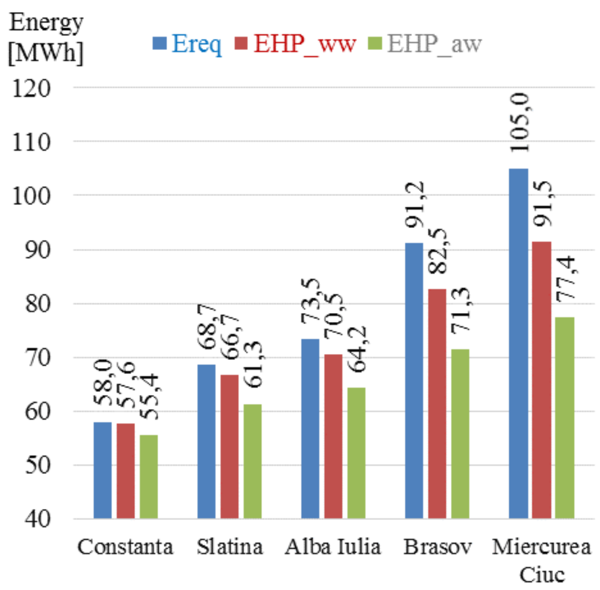

Figure 7. Building heating energy required and obtained by ASHP, respectively WSHP energy for 5 cities of Romania.

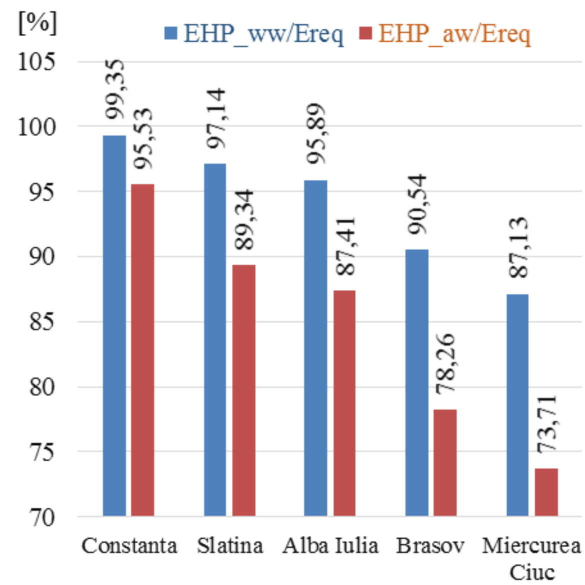

Figure 8. Obtained overall energy ratio for ASHP respectively WSHP, for 5 cities in Romania.

For the first 3 weather areas of Romania, in conditions selected for this study, energy obtained from the water source heat pump, represent more than 95% from the required building heating energy. With a maximum of 73.5MWh, the city from middle of Romania, required heating from auxiliary source, is 3%, from total, in case WSHP or 9.3% if ASHP is selected.

For each period and for each city, energy demand of the same reference building, with a specified overall transfer heat rate equal 1000W/K, is evaluated based on equation (7).

For selected methodology, the evaluation take place in different environment from weather point of view, were warmly, respectively cold cities are selected from a various types of cities in Romania, regarding their different external temperatures and distribution over the year. This type of study, by comparison, can generally give an overall opinion about design and energetic efficiency of hybrid heating by revealing the percentage of heat obtained from common heat pumps and necessary auxiliary energy when various type of outside conditions, from a building point of view, are plotted. More than that, the energy efficiency is given by the local condition of installation, for different water source temperature, of lake or rivers, or even underground water. By type, the heating system can be configured in summer to give required energy for cooling. Combining with cooling, the indicators for auxiliary source can change, and with them, the balance of designing equipment dimensions.

With equations (7), (8) and (9), are established auxiliary source energy, extracting from the building heating demand the heating provided by heat pumps, at their condensers. Reporting to seasonal building energy, WSHP and ASHP delivered energy, are obtained the percentages of delivered from total demand. This indicator is helping in selection of electrical factor of the heat pump. When specific percentage, for overall energy is required, this correspond to a specific electrical factor.

Increasing the compressor power, the overall energy obtained from WSHP and ASHP condensers, is more appropriate with building demand, with higher value in case WSHP. Can be seen in Figure 9 for electrical factor bigger than 0.12, more than 95% of energy required by building is delivered from heat pump, in water source configuration. In case of warmly environment city, a small WSHP in term of electrical factor of compressor can be used with good performances. For such cities, a WSHP is suitable to be used in heating system of the building. In the other hand, a WSHP for heating of the building is suitable when a value over 0.12, of electrical factor, and when the most important heating equipment of a combined hybrid system is required to be the heat pump.

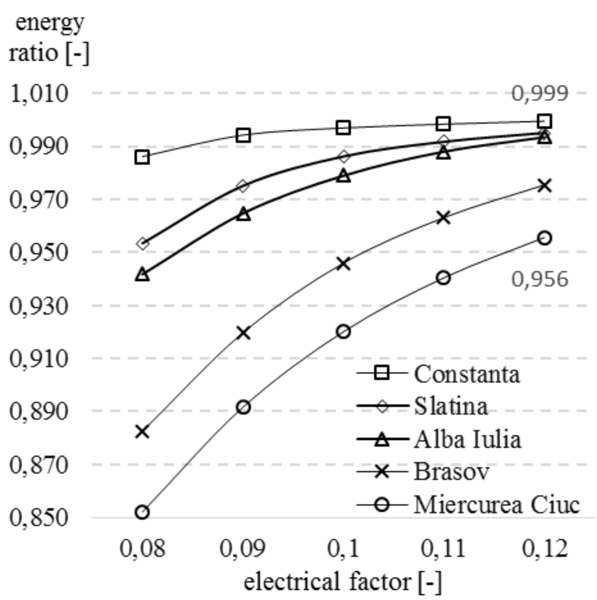

Figure 9. Obtained overall energy ratio for WSHP when electrical factor is changed, for 5 cities in Romania.

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For a desired percentage and with the proper auxiliary source, a hybrid system can be a good solution when efficiency improvement, of classical source, is purposed. Provided diagrams for heating power delivered by ASHP and WSHP, using presented methodology, are suitable data in selection, by design, of the proper heat pump dimension to be used. Also, when calibration is possible, in working parameters of heat pumps, this study reveal for each temperature differences between refrigerant and evaporator, respectively condenser environment, are suitable to be. Energy balance presented can reveal the way designers select the dimension of heat pump, in term of compressor power, for a desired overall energy ratio.

Dependency of heat pump performances with heating temperatures designed for cold season, can help engineers to correlate heating agent temperatures with heat pump parameters but also with external temperature or required efficiency. Plotted graph regarding isentropic efficiency is an example based indicator. This only reveal the way compressor isentropic efficiency behave when evaporator environment temperatures are varied. With low isentropic efficiencies, electrical power can be high enough to be tacked into account. For this, is suitable to correlate the compressor power directly with overall energy ratio and less with external temperature.

Delivered energy by equipment of the heat pump, in heating system, can help designers in evaluation criteria of building energy efficiency. By correlation with building surface, a specific square meter energy load, for entire season, is an interesting and used indicator in energy evaluation.

Direct influence of exterior temperature in delivered heating energy of air source heat pump is a key indicator of system performances, together with temperatures distribution per heating season. Knowing that a level of outside temperature is picked only few hours in entire season, make possible if an air source heating pump have relevance in heating system selection for a specific area. Together with global heat transfer of entire building, in heating, the heat pump electrical energy consumption will be the second indicator of system feasibility. Having the presented methodology together with Meteonorm data base, evaluation algorithm proposed in this section, can be a starting tool to show the CO_2 released, for building heating demands. With the heating powers at condenser for ASHP and WSHP, and for ON/OFF compressors typologies, COP is just a rapport between W_{EL} and Q_{CD}, being possible to be obtained for each step of calculation. Seasonal energy efficiency ratio, is the same, an indicator possible to be evaluated through presented equations, having hourly ISO weather data provided from Meteonorm database. Comparing with ground source heat pumps (GSHP), ASHP and WSHP present the advantages of using a heat sources, which in time, didn't lose the capacity of providing heat, as was shown by literature study of this paper. In case experimental studies are lunched, key system parameters evaluation, can deeply change the efficiency of the method, in case isentropic efficiency is known, for evaporator and condenser corresponding temperatures, together with their temperature differences.

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