DOI: 10.2507/31st.daaam.proceedings.xxx

APPLICATION OF AIR-WATER HEAT PUMP IN REFURBISHED BUILDING WITH EXISTING RADIATOR SYSTEM

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This Publication has to be referred as: Teskeredzic, A[rmin]; Blazevic, R[ejhana] & Dzino, S[anita] (2020). Application of air-water heat pump in refurbished building with existing radiator system, Proceedings of the 31st DAAAM International Symposium, pp.xxxx-xxxx, B. Katalinic (Ed.), Published by DAAAM International, ISBN 978-3-902734-xx-x, ISSN 1726-9679, Vienna, Austria
DOI: 10.2507/31st.daaam.proceedings.xxx

Abstract

Strategic renovation of existing building stock is strongly promoted in Energy Efficiency Directive and Energy Performance of buildings directive. Measures at the envelope are usually combined with replacement of existing heating system. This paper deals with situation where existing radiator system are connected to heat pump instead of previously installed old boiler for heating. Designers of these systems rely on bin method which is well designed for new systems. However, in case of existing systems several factors determine the available power from heat pumps such as: oversized radiators, over-or under flow of hot water in existing system – both influencing the return temperature or inlet temperature to heat pump. It is demonstrated that in some cases available and declared capacity of the pump is overestimated which is result of neglecting previously mentioned factors. In this paper the adaptation of existing bin method according to standard EN 14825 is proposed which fits any existing system. Calculations are performed and results are presented for the baseline condition and three scenarios – two with radiator system and one with underfloor heating. Advances of improved methodology for calculation are highlighted and the conclusions are presented at the end.

Keywords: heat pump; bin methodology; bivalent temperature; seasonal coefficient of performance

1. Introduction

Energy balances in the EU member states have shown that there is a continuous increase in final energy consumption and that the buildings sector participate with 40% in the overall consumption [1]. It is also known that the increase of the living standard results in higher energy consumption for heating and cooling. Due to its high share in overall consumption, the building sector is recognized as the area that has the greatest potential for reducing energy consumption at the national level. Energy efficiency in the building sector, as well as promotion of sustainable buildings, are becoming extremely important instruments for achieving energy consumption reduction. Also, the Energy Efficiency Action Plan [2] and a several number of directives, which aim is to promote the improvement of the energy performance of buildings [3], underline the importance of energy efficiency in buildings as well as the importance of energy management in buildings.

In the past years, the technology of heat pumps has been improved, so today air heat pumps are increasingly used as primary heating systems. Additionally, in the Directive 2009/28/EC on the promotion and use of renewable energy

sources [4], heat pump technology is recognized as necessary for using renewable energy sources from air, water and ground. Air-water heat pumps are especially suitable for the replacement of existing heating system, because they are relatively cheap, easy to install and air source is available everywhere. Coefficient of performance and heat capacity of these pumps are strongly dependent on the outside air temperature, inner temperature and type of the heating system. As the coefficient of performance of the heat pump changes primarily depending on the climatic conditions, many researchers have analyzed influence of the outside air temperature on the seasonal performance of the heat pump. Analysis conducted in [5] was used to investigate the influence of the outside climate on the seasonal performances of different kinds of heat pumps coupled with different buildings. In order to model the outdoor climate, researchers were utilized bin method. Also, in [6] authors proposed different procedures for the analysis of the seasonal performance of mono-compressor, multi-compressor and variable speed compressor air-to-water heat pumps. The obtained results showed the influence of the effective operating mode of the heat pumps on the SCOP value.

A new method which is used for determining the optimal values of the balance-point temperature and of the thermal storage volume of air-to-water heat pumps for heating is presented in [7] where the seasonal COP of the heat pump system is analyzed as a function of the bivalent temperature and of the storage volume. The results showed that the optimal value of the bivalent temperature is independent of the storage volume. In [8] the effects of heat capacity modulation of heat pump in order to meet variable hot water demand are considered. For this purpose, an energy analysis was performed of an air heat pump with variable speed compressor was carried out. It has been noted that the heating capacity has an approximately linear and proportional relationship with the speed of the compressor.

Weather compensation is type of the heating control which is the most often used to improve the efficiency of the heating systems [9], and can also be used in combination with the heat pump system. Influence of the water return temperature value in the heating system operation is neglected in EN 14825 standard. In this paper, a calculation model and advanced algorithm based on the bin method according to the valid standard for heat pumps EN 14825 have been developed. The main difference between standard bin method according standard EN 14825 [10] and developed algorithm in this paper is that developed algorithm takes into account the water return temperature in heating system. So, this paper presents a comparison of the results obtained using the standard bin method with the results obtained using the developed Excel algorithm, which, taking into account the operation of the complete heating system, represent an improved bin method. Also, in this paper an analysis of the heat pump application in building with different refurbishment level is performed. Two cases of the heating system are considered. First analysed case is with the existing radiator heating system and second case is with new underfloor heating system installed.

2. Formulation of the problem

Energy efficiency programs and plans in the building sector are in the first step focused on the decrease of the energy need of the building by interventions at the envelope of the buildings. This means putting additional thermal insulation at outside walls, roofs and floors usually combined with replacement of old windows with new, efficient ones. Implemented measures at the building envelope impose new conditions for the existing heating systems, which become oversized for refurbished building. The temperature levels should be decreased in order to avoid overheating of conditioned spaces.

When advanced scenarios of energy efficiency measures are applied – existing heating system is replaced with the new, efficient system which better fits decreased building needs. In that sense and in combination with promotion of renewables in buildings, installation of heat pumps becomes very attractive. National programs are designed to support promotion of renewables in buildings and different financial mechanisms are available. In Bosnia and Herzegovina two entity Long term strategies for the refurbishment of the existing building stock promote the use of the heat pumps as the primary heating systems after the deep refurbishment of existing buildings.

In this part of Europe, in most cases, the heating systems were designed with high temperature levels such as 90/70/20. In this case, the application of the heat pumps is out of question. However, when the existing buildings are deeply refurbished, existing radiators become oversized and the supply temperature can be decreased significantly which opens the possibility for application of the heat pumps. In this paper the heat pump manufacturer data are used in combination with different levels of the existing building refurbishment and adaptation of the existing bin method for calculation of the seasonal COP of the heat pump is proposed. Systematic of the work presented in this paper is as follows:

- Representative building from the TABULA building stock in Bosnia and Herzegovina has been chosen
- Heat demand for the representative building is estimated for original state of the building including two scenarios of refurbishment,
- Application of the heat pump according to EN standard 14825 and the bin method
- Analysis of the deficiencies in existing standard and proposal how to overcome potential problems
- In house developed tool for calculation of quasi-steady-state.

Producers of the heat pumps in their catalogues provide the data on available heat power and COP based on the following parameters:

- Ambient temperature,
  Hot water supply temperature.
- Maximum supply temperature.

The latest is often neglected by the design engineers, leading to poor estimates of the available heating power, especially in cases where for extremely low outside temperatures supply temperature has to be larger than the maximum heat pump supply temperature.

During preliminary calculations, the water supply temperature is usually taken as constant out of three available values (35°C, 40°C, 45°C) and design engineers only need to interpolate the available heat power and COP based on the outside temperature. This would however mean that the system operates with constant supply temperature providing the control of the heating system by variations of the mass flow rate in the system. In case of the weather compensated control, the mass flow rate should be constant while the control is provided by changing the water supply temperature. In this case, the two-side interpolation should be implemented. Namely, according to the existing outside temperature the first level of interpolation is to find available heat powers for three characteristic supply temperatures based on the known outside temperature. Once these heating powers and COPs are found, the second level of interpolation is made according to the known supply temperature from the controller. Two level interpolation results in the known heating power and COP for any given outside and supply temperatures.

The described process seems to be accurate and straightforward at the first sight but, in almost all cases, the design engineers neglect two extremely important parameters such as:

- the return temperature from the existing system and
- maximum temperature limit of the heat pump supply temperature.

These two problems are negligible if the nominal water supply temperature is lower than the maximum allowed temperature. However, in cases where the supply temperature should be even slightly increased (meaning +5°C) and if these two values were not taken into account, then the problems could be expected during exploitation of the system. Declared available heat pump power would not be really available since the return temperature is high and the mass flow rate through the heat pump would not be sufficiently high to achieve declared heat power. This fact is illustrated with numerical example in the results subsection.

3. Mathematical model and heat pump data

In this subsection a mathematical model for the problem is outlined as well as the producer's data on available heat pump heating power and COP for given outside temperatures and water supply temperatures. For all cases studied in this paper the weather compensated control has been applied. In general, according to the standard [10], the seasonal heat pump performance calculation is based on outside and water supply temperatures for each bin. In this case, as already mentioned, the return temperature from the system is neglected. In cases where the supply temperature has to be even slightly higher than the maximum supply temperature, problems could be expected. Namely, it is not only that additional heater will increase the supply temperature according to the needs than the return temperature will be increased as well.

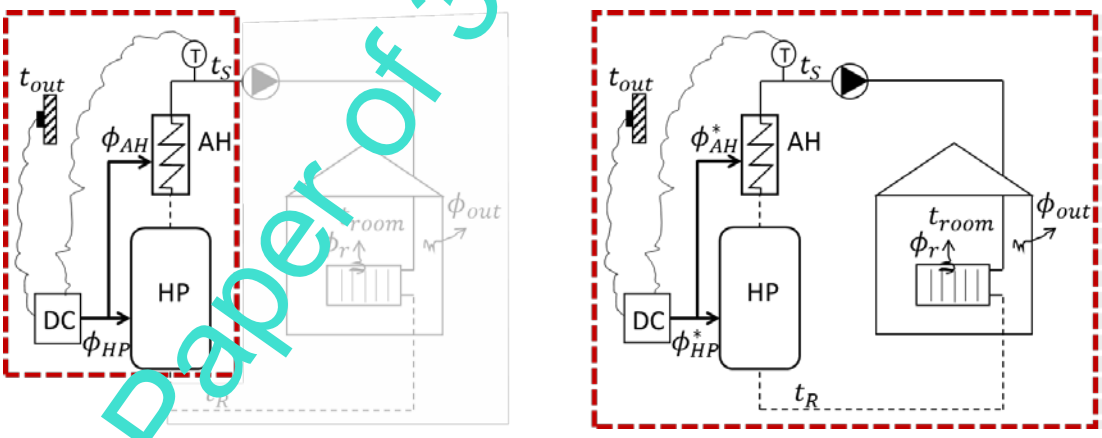

Fig. 1. Control Volume marked in red dashed rectangles for: standard application of the bin method (left) and improved method proposed in this paper (right).

Figure 1 shows control volumes for calculations for two cases, on the left-hand side is the standard case which takes into consideration the desired supply temperature and the results of calculations are the heat fluxes ϕ_{HP} and ϕ_{AH} or heat pump and additional heater flux, respectively. On the right-hand side proposed improvement outlined in this paper expands the control volume over the consumer and takes into account the radiator system and the return temperature, so that all values presented in figure can be calculated. Before the explanation of both procedures is given, it is necessary to explain how the water supply temperature is determined (weather compensated control) and how the heating power from the HP producer's data is obtained.

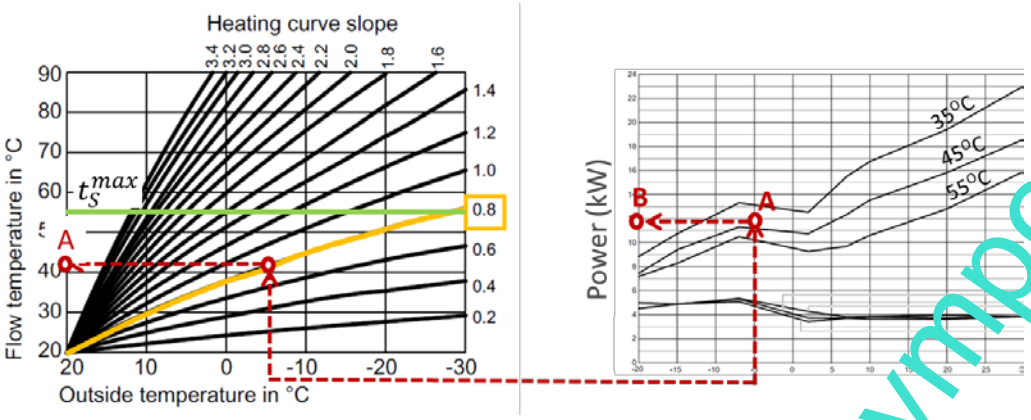

Fig. 2. Weather compensated control (left) and producer's data sheet on heating power versus outside temperature (right).

Figure 2 shows an illustration of how the water supply temperature is obtained marked with letter A (left) and how the available power from the heat pump producer's data sheet marked in letter B (right) is read. For the known outside temperature and previously chosen slope of the heating curve (in this case 0,8), the flow temperature in Figure 2 (left) can be obtained. For the same outside temperature and obtained water supply temperature from the heat curve, an interpolation is required to get the available power from the heat pump – see the line connecting point A and B in the Figure 2 (right). Note also that the maximum supply temperature is also given by the producers and in this case is shown with the green line in Figure 2 (left). If the required water supply temperature is higher than the t_S^{max} an additional heater will provide heat flux ϕ_{AH}.

Now when the water supply temperature can be obtained, as well as the available heat power from the heat pump, we can write the basic equation valid for, what we here call the Standard approach, as following:

$$\phi_{HP} + \phi_{AH} = K_B(t_{room}^{SET} - t_{out}) \tag{1}$$

where ϕ_{HP} is the available heat pump heating power according to HP data (point B in Figure 2 right), ϕ_{AH} is the additional heater heating power, K_B is the building loss coefficient (constant), t_{room}^{SET} is the desired room temperature and t_{out} is the outside temperature. Note that equation (1) is valid for any outside temperature.

The procedure for determination of required parameters is the following:

1. Outside temperature (t_{out}) is known and the water supply temperature (t_S) is obtained as shown in Figure 2 (left).
2. The room temperature (t_{room}^{SET}) is chosen and the right-hand side of equation (1) is known.
3. Based on the known values from the steps 1 and 2, ϕ_{HP} is obtained from the heat pump producer's data for the previously chosen type of HP,
4. Additional heater heating power (ϕ_{AH}) is calculated from equation (1) having in mind that it is only unknown value.

As already mentioned, this approach neglects the temperature level in the heating system and in special cases can lead to wrong results. This is true for the systems where the additional heater is necessary to obtain the required supply temperature which can happen in two general cases. One is due to lack of heating power available from the heat pump for the given outside temperature or second if the water supply temperature is higher than the t_S^{max}. The second case is dangerous since the presented standard procedure and equation (1) does not address increased temperature level in the system. Therefore, in this work an improved procedure is proposed which takes into account the control volume presented in Figure 1 (right). On the top of that, the proposed procedure also allows some additional calculation possibilities such as:

- Over or under-sized radiators or in general heating emitter bodies,
- Over or under-supply of circulation of hot water in the system in case of lack of hydronic balancing.

These two additional improvements of the existing procedure enable the calculations for all practical problems when the envelope is refurbished and the system has to be retuned, either by changing the slope of the heating curve or by installation of the heat pump which will be used as a new heat generator in the system.

The proposed procedure solves the following equation:

$$\underbrace{\phi_{HP}^*}_{1} + \underbrace{\phi_{AH}^*}_{2} = \underbrace{f_f \cdot \dot{m}c(t_S - t_R) = f_r \cdot \phi_r^N \left[\frac{\left(\frac{t_S + t_R}{2}\right) - t_{room}}{\Delta t_m^N}\right]^n}_{3} = \underbrace{K_B(t_{room} - t_{out})}_{4} \tag{2}$$

where ϕ^*_{HP} and ϕ^*_{AH} are recalculated values of available heat pump power and additional heater power, respectively, f_f and f_r are the factors of over-flow and oversized radiators, respectively, $\dot{m}$ is the mass flow rate, c is the specific heat of water, n is the exponent of the radiator or any other heating body and Δt^N_m is the temperature difference between average temperature of radiator and room for nominal conditions, known as soon as the nominal conditions are adopted.

Note that in this case, although the room temperature is set at the controller, t_{room} is calculated and is also variable. Variables and unknown values in the equation (2) are t_R, t_{room}, ϕ^*_{HP} and ϕ^*_{AH}. Since we have the mathematical model it is possible to describe the calculation procedure as follows:

1. Based on the known outside temperature (t_{out}), the water supply temperature (t_S) for the given slope is obtained as it was the case in standard calculation approach.
2. Equation (2) is decoupled and unknown variables are iteratively calculated by forcing equality of the terms 2, 3 and 4.
3. If the obtained room temperature (t_{room}) is within prescribed interval $t_{room} = t^{SET}_{room} \pm 0{,}5$°C then proceed to step 4, else go back to step 2, change the slope of the heating curve and proceed further.
4. Based on known t_{room} calculate ϕ_{out} for all set of outside temperatures.
5. Since the return temperature (t_R) is known calculate corrected available heat power from the heat pump by solving following equation

$$\phi^*_{HP} = \dot{m}c\{max[min(t_S, t^{max}_S) - t_R, 0]\} \tag{3}$$

6. Calculate corrected addition heat power (ϕ^*_{AH}) from equation (2) since all parameters are known.

Please note that the constraint imposed by equation (3) limits the water supply temperature on maximum value (t^{max}_S) and the real available delivered heating power from the HP will be $\dot{m}c(t^{max}_S - t_R)$. Also, if the return temperature is higher than the maximum value of the supply temperature, the HP will not start at all and zero heating power will be transferred to the circulating water ($\phi^*_{HP} = 0$).

Proposed standard bin methodology also introduces the part load operation correction, which is given as:

$$COP_{PL} = COP_{DL}\frac{CR}{C_C \cdot CR + (1 - C_C)} \tag{4}$$

where COP_{PL} is the part-load coefficient of performance of the HP which takes into account that available heat power is larger than the heat demand and HP works in the ON/OFF mode, COP_{DL} is the declared (available) heat power from the heat pump, CR is the capacity factor and C_C is the degradation factor proposed by the standard. In all calculations the degradation factor in this paper is taken to be $C_C = 0{,}9$.

We also introduce the total *COP* which takes into account the additional heater – in our case assumed to be electric heater with efficiency equal to unity. Total COP_{TOT} is calculated as:

$$COP_{TOT} = \frac{\phi^*_{HP} + \phi^*_{AH}}{\frac{\phi^*_{HP}}{COP_{PL}} + \phi^*_{AH}} \tag{5}$$

In this case we can obtain the overall COP_{TOT} of our heating system. At the other side this parameter shows that if more energy comes from the additional heater, the lower the COP_{TOT} would be. This is exactly what we want to demonstrate in this case. Consequently, the seasonal $SCOP_{net}$ is calculated as:

$$SCOP_{net} = \frac{\sum_{j=1}^{k} h_j[\phi^*_{HP}(t_j)]}{\sum_{j=1}^{n} h_j\left[\frac{\phi^*_{HP}(t_j)}{COP_{PL}(t_j)}\right]} \tag{6}$$

where *j* goes from 1 to *k*, where *k* represents the number of bins, h_j is the number of hours within bin *j*, $\phi^*_{HP}(t_j)$ is the available heat pump heating power as a function of t_j and $COP_{PL}(t_j)$ is the part load ratio, again dependent on t_j. Note that COP_{PL} is also given by the producers of heat pumps in the same format as the available heating power. The procedure for finding the COP_{PL} is exactly the same as it is demonstrated for the available power in Figure 2.

4. Results and discussion

For the validation of proposed approach and application of the developed model, an existing single-family house from the TABULA tool in Bosnia and Herzegovina is chosen (http://webtool.building-typology.eu/#bm). Building stock is represented with the baseline conditions and two improvements, regular scenario and ambitious scenario. Both these are used in this paper. Only difference is the fact that within this paper the heat pump replaced the existing old gas boiler.

Fig. 3. Residential building BA.N.SFH.06.Gen downloaded from Tabula WebTool

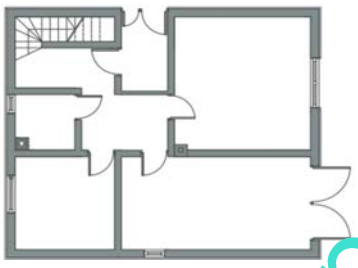

Fig. 4. The geometry of observed residential building

Figure 3 present the chosen-single family unit with its floor plan (Figure 4) for which the calculations are done. Scenario 1 given in the table assumes the insulation of outside walls with 10 cm with material λ=0,04 W/mK and installation of new windows with U_{win}=1,4 W/m^{2}K. Scenarios 2 and 3 are related to ambitious building refurbishment with the insulation of outside walls with 20 cm with material λ=0,04 W/mK and installation of new windows with U_{win}=1,1 W/m^{2}K. As a result of interventions at the building envelope the heat demand for scenario 1 decreased to 7,2 kW and 6,5 kW for scenarios 2 and 3. It is assumed that the existing heating system is the radiator system with old gas boiler, working in the temperature regime 90/70/20.

STATE/ SCENARIO	Heating system type	Design temp. regime (°C)	Real temp. regime (°C)	Heat demand (kW)	Oversize factor f_r (%)	Overflow factor f_f (%)
BASELINE	Radiators	90/70/20	82/64/20	10,0	20%	10%
SCENARIO 1	Radiators	–	68/55/20	7,2	20%	10%
SCENARIO 2	Radiators	–	64/52/20	6,5	20%	10%
SCENARIO 3	Underflow h.	45/40/20	44/40/20	6,5	7%	0%

Table 1. Reference data for baseline conditions and three scenarios elaborated in the paper.

Table 1 shows the basic data used in the calculations. Despite the fact that heat pump would not be applied for the high temperature levels (baseline condition), the calculations are also done for this case. The intention is to underline the difference between the standard and improved approach proposed in this paper. Design temperature regime is given for the existing state – baseline and for the underflow heating, since the first is related to existing system and latter assumes that a new underfloor heating system is designed. Oversize factor f_r is for all cases with radiators given based on the nominal or the reference conditions. The same is valid for overflow factor f_f. Also, note that for the Scenario 3, the new underflow heating system, perfect hydronic balancing is assumed, which is done because of the small nominal temperature differences.

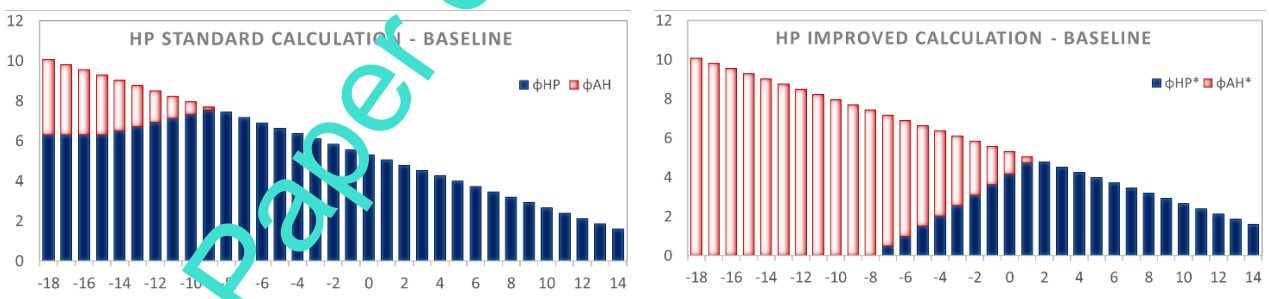

Fig. 5. Heating power from heat pump and additional heater versus outside temperature for baseline case using standard bin method (left) and improved bin method (right).

Figure 5 shows the redistribution of heat power and additional heater heating power for outside temperatures in the range −18°C to 14°C for standard bin calculation approach (left) and improved calculation approach (right). The difference is obvious and can be summarized in two major points. One is that in case of standard approach the designed maximum capacity of the additional heater is 3,8 kW, while in the improved calculation methodology no heating power is available from the heat pump for temperatures less than −7°C. This also means that the additional heater should be sized according to the designed heat load (10 kW). Second difference is that the bivalent temperature is for the standard calculation method −8°C, while in the improved method is at 2°C.

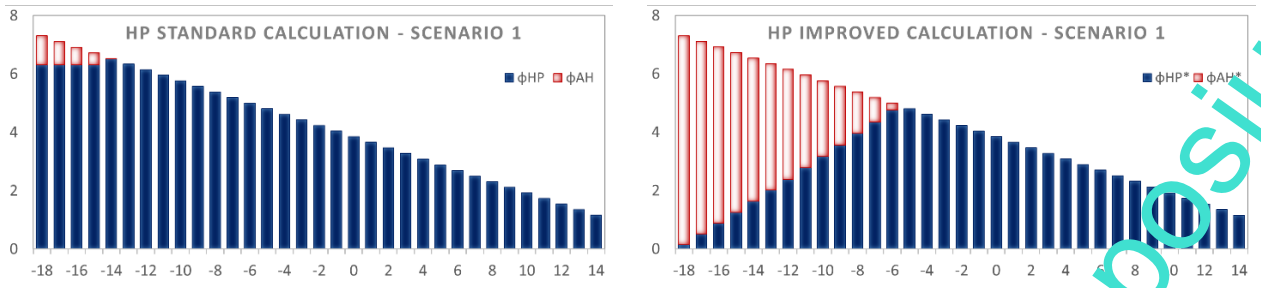

Fig. 6. Heating power from heat pump and additional heater versus outside temperature for Scenario 1 using standard bin method (left) and improved bin method (right).

In Figure 6 Scenario 1 data are given, in which is visible that differences between standard and improved exist. Additional heater for the standard method is 1 kW, while for improved method is 7,15 kW which is almost the design heat load for refurbished building. Also, for standard method the bivalent temperature is at −14°C, while in improved method is at −5°C. In terms of the energy consumption which will be elaborated later it should be said that it depends upon the duration of temperature bins.

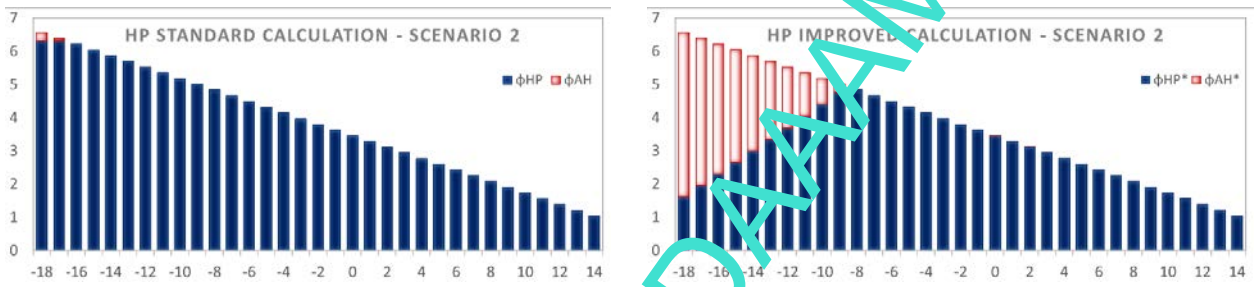

Fig. 7. Heating power from heat pump and additional heater versus outside temperature for Scenario 2 using standard bin method (left) and improved bin method (right).

Figure 7 is related to Scenario 2 and it is visible that according to standard method additional heater is not necessary while for the improved method its size is almost 5 kW, which is more than 75% of the design heat load. For standard method no additional heater is necessary since its calculated power is less than 1 kW and the bivalent point in this case is −16°C, while in improved method is at −8°C.

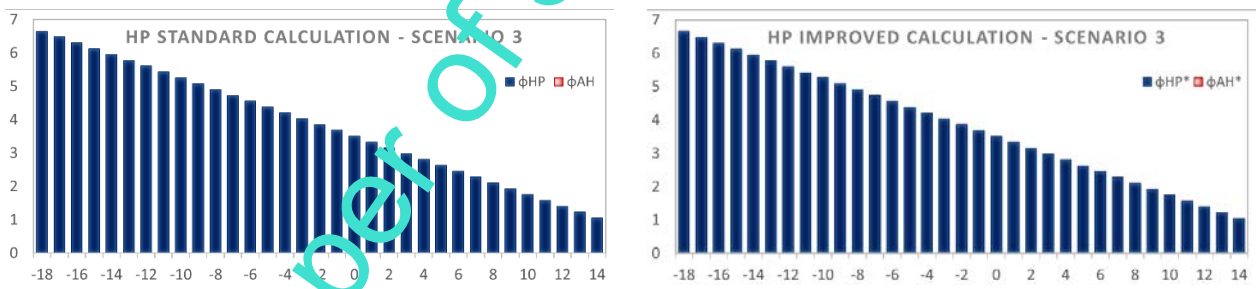

Fig. 8. Heating power from heat pump and additional heater versus outside temperature for Scenario 3 using standard bin method (left) and improved bin method (right).

Figure 8 corresponds to the Scenario 3 which is the case with implementation of the new distribution system with underflow hydronic heating. It is visible that no additional heater is necessary and that in both cases – standard and improved method the heat pump covers the heat demand for any outside temperature. It should be said that both methods give the identical results.

The major differences between what we here call standard and improved calculation methodology for this special purpose can be outlined in the fact that the standard approach does not recognize the return temperature. Its control volume is shown in Figure 1 (left). At the opposite side, an improved method control volume is given in Figure 1 (right) and it takes into consideration the return temperature from the heating system and recognize that the heat pump will not start with return temperatures higher than 55°C.

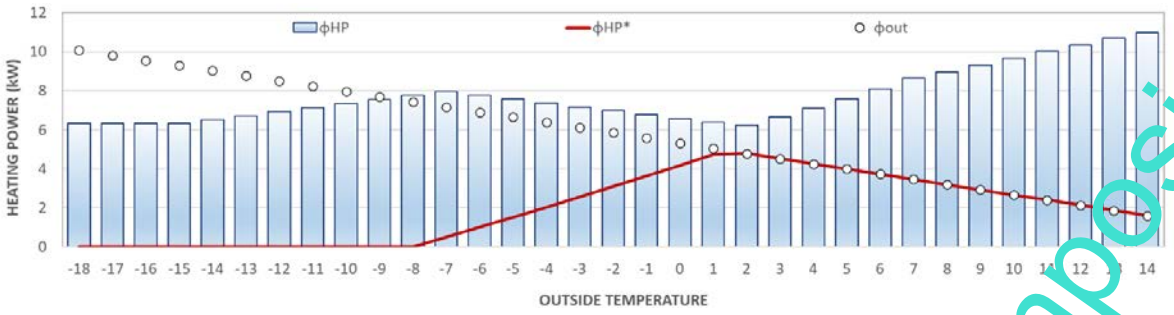

Fig. 9. Heat demand, available heat pump heating power for standard and improved methodology for baseline case.

Figure 9 illustrates the differences at the baseline case in which the standard and improved methodology discrepancy is obvious. As it can be seen the heat demand varies linearly with the temperature change. Available heating power from the producer's data sheet significantly differs from the available heating power introduced in improved methodology. It can be seen that the improved method shows that the heat pump will not deliver any heating power for temperatures equal or lower than 55°C. This is because of the high temperature levels in the system having in mind that the return temperature from the heating system is exactly 55°C for the outside temperature of −8°C. It can also be seen that the red line corresponds with the circles at Figure 9 for the outside temperatures of 2°C onwards, which presents the bivalent temperature for the heat pump. Note that this is consistent with the data presented in Figure 5 (right).

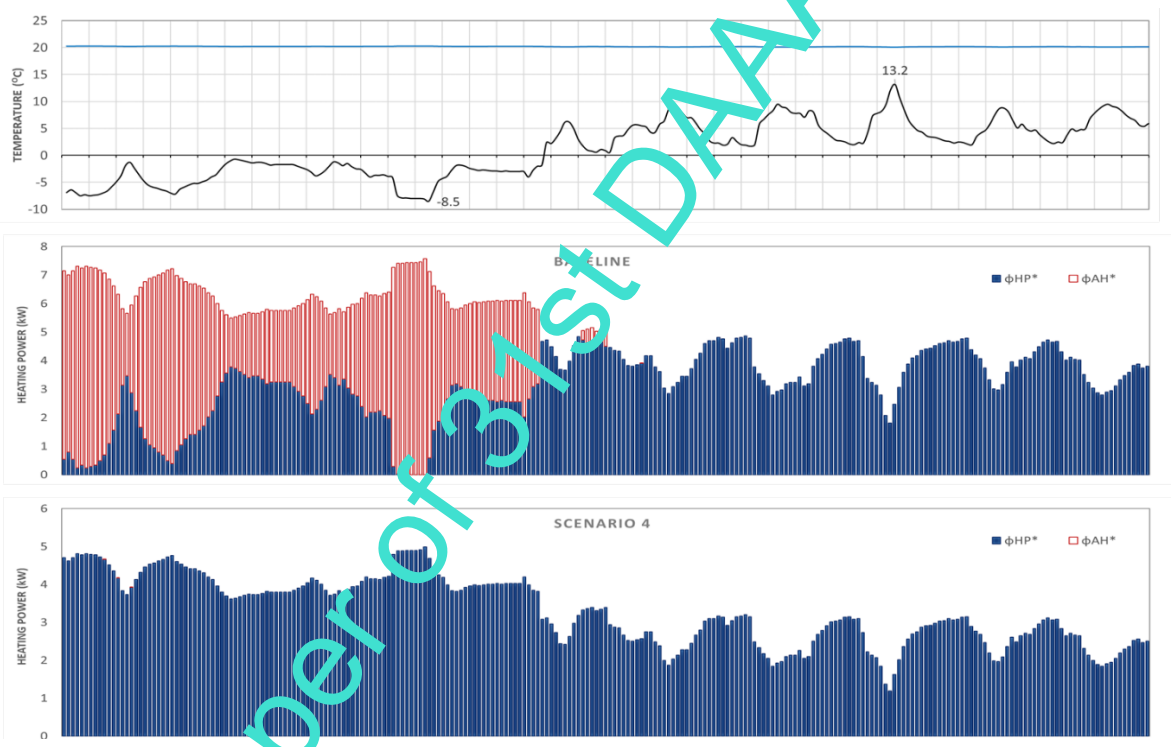

Fig. 10. Variation of outside temperature and maintained room temperature (top), heat demand covered by heat pump and additional heater for baseline case (middle) and heat demand covered by heat pump for Scenario 4 (bottom).

For the validation purposes a period of 10 days is taken for which the measured values of the outside temperatures were available. Variation of the outside temperature is presented in Figure 10 (top) together with maintained temperature in the building (blue line). It can be seen that the outside temperature variation in this period is significant, providing an average temperature of 1,1°C with extreme values of −8,5°C and 13,2°C. It is supposed that heating system works 24 hours a day without setback settings. Transient effects are not taken into account but the assumption of continuous heating minimizes a modelling error in this case

Figure 10 (middle) shows the behavior of the baseline system with the heat pump installed. It is visible that during the cold periods where the temperature is below −5°C almost all heating power is coming from the additional heat source. However, it can be also seen that during the periods of high outside temperature the whole heat demand is covered by the heat pump. Figure 10 (bottom) shows the best case or Scenario 4 with the underflow heating in which is clearly visible that the complete heat demand is covered by the heat pump irrespective of the variations of the outside temperature.

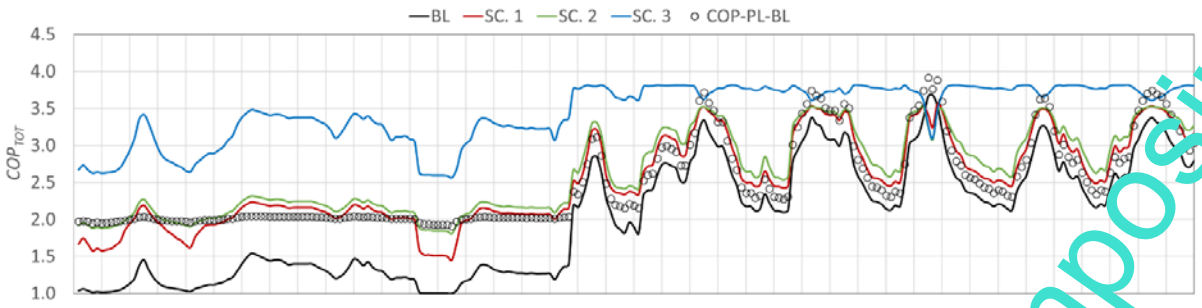

Fig. 11. Comparison of the total coefficient of performance introduced by equation (5) for baseline condition and three scenarios, including the part load factor for the baseline case.

Figure 11 presents variation of the COP_{TOT} for the baseline case and three scenarios and the temperature variation given in Figure 10 (top). On the top of that for the illustration purposes the COP_{PL} variation is given in circles which is related to the baseline scenario and which is dependent only upon the outside temperature (standard method). It is clearly visible that the COP_{TOT} value is largest for the underfloor heating (Scenario 3) with its average value of 3,46. Difference between Scenarios 1 and 2 is not large (red and green line in Figure 11) and both values are significantly below values from Scenario 3, giving an average values of 2,52 and 2,64 respectively. Values of the COP_{PL} should be compared to baseline case and it can be seen that the differences are significant which is especially valid for the period with low outside temperatures.

Finally, we will show the seasonal performance factors for all treated cases which is more relevant than already shown data for the arbitrary chosen period as shown in Figures 10 and 11.

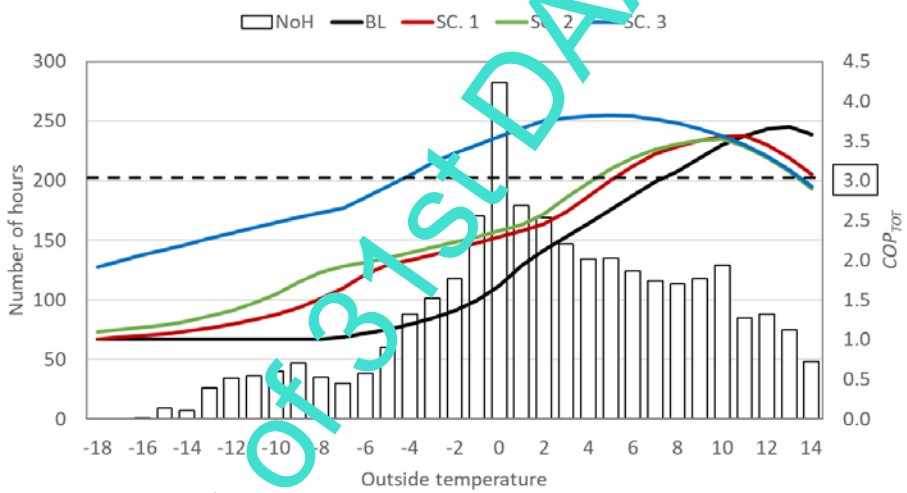

Fig. 12. Number of hours for every temperature bin and representation of $\boldsymbol{COP_{TOT}}$ for baseline case and three scenarios.

Figure 12 summarized data needed for the calculation of the net seasonal coefficient of performance ($SCOP_{net}$). Data for determination of duration of every outside temperature bin are taken for the city of Sarajevo (Bosnia and Herzegovina) in the heating season 2011/2012. Left hand side axes represents the hours and the right-hand side axes values of COP_{TOT} added to the graph for the analysis. At the graph in Figure 12 a dashed line representing the value of $COP_{TOT} = 3$ is added. It can be noticed that the Scenario 3 with underfloor heating maintains the value of COP_{TOT} over 3 for the outside temperature values larger than −4°C and keeps it until value of 13,5°C, which is anyhow the limit temperature at which the heating system is going to be off.

STATE/ SCENARIO	BP (°C)	BP* (°C)	AH (kW)	AH* (kW)	$SCOP_{net}$	HP ratio (%)
BASELINE	−8	2	3,8	10,0	2,38	66,2%
SCENARIO 1	−14	−5	1,0	7,2	2,40	92,3%
SCENARIO 2	−16	−8	–	5,0	2,42	96,8%
SCENARIO 3	–	–	–	–	3,27	100,0%

Table 2. Summarized results for all calculated cases.

At the same time the number of hours for this outside temperature range (−4°C to 13,5°C) represents the 87% of the total heating time and the same reasoning can be applied for other Scenarios in order to draw to conclusions. Table 2 summarizes all calculation values obtained for the seasonal values.

Table 2 shows values of the bivalent point of operation for standard and improved method (with * as superscript), design capacity of the necessary additional heaters and the seasonal net coefficient of performance. It is visible that the value of $SCOP_{net}$ is relatively high even for the baseline case and does not differ much in comparison with Scenarios 1 and 2. The reason is that the net seasonal coefficient of performance is calculated when the heat is delivered by the heat pump. However, the HP ratio given in the last column represents the share of the delivered heat from the heat pump in overall energy need. It is visible that even with Scenario 1 this value increases significantly. This was one of the reasons why the COP_{TOT} value has been introduced in this work. As already said, Scenario 3 is far the best and the heat pump in this case covers the total energy required by the system with no additional heat.

5. Conclusions and further work

This paper provides the guidelines how to deal with combination – a refurbished building with decreased heat demand and installation of the heat pump, especially in cases where the radiator system exists. It has been demonstrated how the analysis has been undertaken step by step for each scenario. For this particular case one may conclude that the installation of heat pump with radiators is not energy efficient. However, if the building envelope allows interventions such that heat demand drops to let us say 50%, than the temperature level would be decreased and the heat pump installation would probably result in higher values of $SCOP_{net}$ and smaller capacity of the additional heater.

However, there is no general conclusion which can be used in all practical situations. In the current study we assumed the oversizing of the existing radiators of 20% and the flow rate of 10%. If these ratios are changed, all numbers will change as well. In general, a conclusion can be made that if the nominal return temperature of the radiator system is close to the maximum heat pump temperature limit then it can be expected that the additional heater will have to cover nominal heat load and the seasonal COP would be lower as well. In case where the supply temperature is only slightly above the maximum heat pump limit temperature the heat pump operation should be reasonable but the calculations should be performed. In that sense, this paper introduces the methodology and necessary mathematical model how to deal with this problem including the systematic applied in this text.

Further work will be focused on heating system where the hot water is prepared together with heating of the building including the operation of the storage tank and its role in the system. Also, the target of the future analysis will be to include the transient effects and intermittent heating which will be close to the realistic situations.

6. References

[1] Directive 2010/31/EU of the European Parliament and of the Council of 19 May 2010 on the energy performance of buildings (EPBD). Off. J. Eur. Union, (2010). 153, pp. 13–35.

[2] Action Plan for Energy Efficiency: Realising the Potential, Accessed on: 2020-03-13

[3] Directive 2012/27/EU of energy efficiency (EED) of the European Parliament and of the Council of 25 October 2012. Off. J. Eur. Union, (2012). 315, pp. 1–56.

[4] European Parliament. 2009. "Directive 2009/28/EC of The European Parliament and of the Council of 23 April 2009 on the promotion of the use of energy from renewable sources and amending and subsequently repealing Directives 2001/77/EC and 2003/30/EC). Off. J. Eur. Union (2009).

[5] Naldi, C., Dongellini, M. & Morini, G. L. (2015). Climate influence on seasonal performances of air-to-water heat pumps for heating, Energy procedia, Vol. 81, 2015, pp. 100-107., ISSN: 1876-6102, DOI: 10.1016/j.egypro.2015.12.364

[6] Dongellini, M., Naldi, C. & Morini, G. L. (2015). Seasonal performance evaluation of electric air-to-water heat pump systems, Applied thermal engineering, Vol. 90, 2015, pp. 1072-1081., ISSN: 1359-4311, DOI: 10.1016/j.applthermaleng.2015.03.026

[7] Naldi, C., Morini, G. L. & Zanchini, E. (2014). A method for the choice of the optimal balance-point temperature of air-to-water heat pumps for heating, Sustainable cities and society, Vol. 12, 2014, pp. 85-91., ISSN: 2210-6707, DOI: 10.1016/j.scs.2014.02.005

[8] Szreder, M. & Miara, M. (2020). Effect of heat capacity modulation of heat pump to meet variable hot water demand, Applied thermal engineering, Vol. 165, 2020, Article 114591, ISSN: 1359-4311, DOI: 10.1016/j.applthermaleng.2019.114591

[9] Blazevic, R., Teskeredzic, A. & Jugo, E. (2019). Optimal Heat Flux in Intermittently Heated Buildings, Proceedings of the 30th DAAAM International Symposium, 23-26 October 2019 Zadar, ISBN 978-3-902734-22-8, ISSN 1726-9679, B. Katalinic (Ed.), pp. 0701 – 0708, Published by DAAAM International, Vienna, Austria, DOI: 10.2507/30th.daaam. proceedings.096

[10] European Committee for Standardization, Standard EN 14825: 2013, Air Conditioners, Liquid Chilling Packages and Heat Pumps, with Electrically Driven Compressors, for Space Heating and Cooling e Testing and Rating at Part Load Conditions and Calculation of Seasonal Performance (2013).