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The impact of the work under partial load on the energy efficiency of an air-to-water heat pump

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Abstract. The energy efficiency of air-to-water heat pumps used for central heating depends significantly on the correctness of the device choice. Under given climate conditions, the temperature of the bivalent point defines the amount of energy supplied from an additional source, usually an electric heater, in periods when the power of the heat pump is too low. Commonly overlooked fact is that excess capacity is also unfavourable and has significant, negative impact on the energy efficiency of such a system. In the article, basing on the measurement data of the real installation of the air-to-water heat pump, a significant decrease in the energy efficiency of the device during periods of low energy demand was shown. The average monthly value of SCOP in the warm months was similar or even lower than in the winter months. The reason of this is the operation of the device under partial load. Although the values of COP were very high in short periods of operation, frequent switching on and off of the heat pump reduced its energy efficiency. Also, the length of operation in stand-by mode, i.e. the length of the periods between successive heating cycles, had negative influence on energy efficiency.

1 Introduction

Air-to-water heat pumps can be successfully used as a single source of heat in a building, even in the Polish climate conditions. Good sizing of the unit and conscious choice of the bivalent point is crucial to obtain good seasonal performances [1]. The problem of undersizing of air-to-water heat pumps is often analyzed. However, equally important but overlooked in the discussion is the problem of oversizing the unit and its operation in part-load conditions. Consequences of the work under partial load are dependent on the regulation system of the device used and the type of heating system. Many devices adjust the power generated by cyclic switching on and off (on/off controlled systems). This type of regulation can cause unstable work of a heat pump in dynamic conditions. This is particularly visible in case of a large disproportion between the thermal load of the building and the maximum capacity of the heat pump. In autumn and spring, the heat demand of the building is usually small. At the same time, the capacity of the air-to-water heat pump is high. The greater the disproportion, the shorter the compressor cycles and the higher the number of on-off cycles. In consequence, a significant degradation of the

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energy efficiency compared to manufacturer data based on EN 14511 [2] standard is observed [3]. According to this standard, the tests are carried out in steady state conditions at full capacity, hence many aspects of the actual work in dynamic conditions are not considered. In order to reduce the efficiency losses caused by the short cycling, the capacity control based on changing the rotational speed of the compressor by a frequency inverter is commonly used. This kind of regulation affects the achieved COP of the heat pump and is limited to the maximum power reduction (usually approx. 30% of the nominal power). It also reduces the range of the on/off regulation, but does not eliminate it completely. Below the minimum power (maximally reduced), this type of regulation will also occur in devices equipped with a frequency inverter [4]. In this article, the analysis of the air-to-water heat pump operation under partial load has been described. Additionally, the difference between the heat pump's efficiency level declared by the manufacturer and the actual measured values has been shown. Finally, the methods of calculation of the device's efficiency under partial load have been discussed, together with their accuracy comparing to the actual measured values.

2 Description of research and analysis

The analyzes presented in this paper were based on measurement data of the air-to-water heat pump, supplying heat to a low temperature installation. The heat is distributed to the offices of approx. 300 m^2 situated on the 1st floor and to the air handling unit room in which the heat source – an air-to-water heat pump with a compressor (split type) of the nominal power of 10.6 kW (A2/W35°C) – is installed. The heat pump performance data – the heating capacity (Q_{HP}) and the declared coefficient of performance ($COP_{decl.}$) measured according to EN 14511 standard are shown in Table 1. The unit works in the monoenergetic mode. In the periods of insufficient heating capacity or the ambient temperature below –15°C, the built-in electric heater delivers auxiliary energy. The installation does not have any heat buffer and is not used for domestic hot water. For further analysis, it is also important that the unit has a frequency inverter installed. It allows the capacity control of the heat pump up to the value of 5.0 kW.

Table 1. Declared heating capacity (Q_{HP}) and $COP_{decl.}$ of the unit, according to EN 14511.

	T_a, °C	-15	-7	2	7	10	12	20
Q_{HP} (kW)	T_{in} = 35°C	5.80	8.47	10.60	14.60	14.80	15.82	17.90
	T_{in} = 45°C	5.20	7.50	10.00	13.10	14.10	14.70	16.80
$COP_{decl.}$	T_{in} = 35°C	1.89	2.62	3.25	4.29	4.40	4.63	5.29
	T_{in} = 45°C	1.50	2.10	2.80	3.10	3.40	3.50	4.10

For the purpose of this paper, analysis of the system behavior over the heating season (September 2013 – May 2014) has been made. Energy consumption (Q_H) and the monthly average ambient temperature ($T_{a,avg}$) is shown in Fig. 1. The monthly average inlet temperature (t_{in}) shows an insignificant change in the range of 30.1–31.5°C. The flow, the inlet (t_{in}) and return temperatures (t_{out}), the ambient temperature (t_a) and the electric power consumption (Q_{el}) are measured. The data were recorded every 1 minute. The data collected allowed to analyze the monthly energy efficiency of the installation and the factors affecting this parameter. It was examined how the cycle length of the compressor and the energy loss caused by the cycling on/off impacts the COP comparing to the one declared by the manufacturer under these conditions ($COP_{decl.}$). In addition, measurements of the electricity consumption between the compressor work periods allowed to determine the energy consumption of the heat pump in stand-by mode. One of the observation was the constant work of the circulation pump.

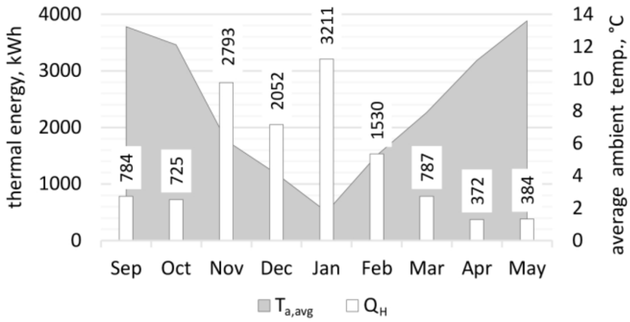

Fig. 1. The thermal energy for the building and ambient temperature in the analyzed months.

3 Data analysis

3.1 Analysis of the heat pump's monthly energy efficiency

The measurements allowed to determine the monthly values of the energy efficiency of the air-to-water heat pump. Two monthly coefficients of performance have been calculated: $SCOP_{month}$ (1) and $SCOP_{cycle}$ (2).

$$SCOP_{month} = \sum Q_{H} / \sum Q_{el,month} \quad (1)$$

$$SCOP_{cycle} = \sum Q_{H} / \sum Q_{el,cycle} \quad (2)$$

To determine the energy efficiency of the device without the losses resulting from the compressor off-mode energy consumption, the value of $SCOP_{cycle}$ has been calculated. It only includes the electrical energy consumed during the work of the compressor $Q_{el,cycle}$. The $SCOP_{month}$ includes all the electricity supplied to the heat pump during the month $Q_{el,month}$. Thus, the difference between the $SCOP_{month}$ and the $SCOP_{cycle}$ is the energy used in the stand-by mode.

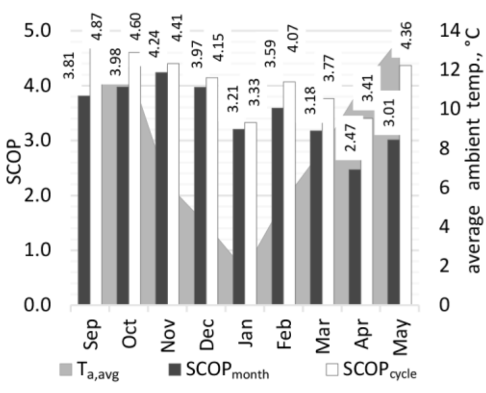

Fig. 2. $SCOP_{month}$ and $SCOP_{cycle}$ in the analyzed months.

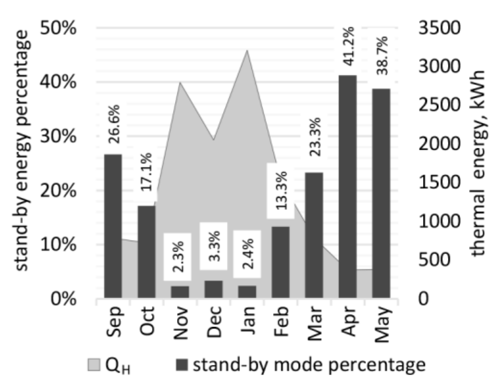

Fig. 3. The share of the standby energy in the total electrical energy delivered to the heat pump.

The results of this calculation with the average ambient temperature are summarized in Fig. 2. The SCOP value depends on the ambient temperature. However, this is not the only

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factor influencing its value. The decrease in SCOP of warmest months (March, April, May) is in fact clearly visible. In April, the value of $SCOP_{month}$ was only 2.47, even though the average ambient temperature was 11.1°C. For comparison, in January ($T_{a,avg}$ = 1.8°C) the value was 3.21. This reflects the significant impact of factors other than the lower and upper source temperatures. Estimating the SCOP, it is necessary to take into account that the heat pump does not operate only in heating mode (active mode). Especially when the ambient temperature is high, there is also significant share of the stand-by mode operation (which is the time when the compressor is not working). The electrical energy is consumed by the automatic control system, the circulation pump and by the fan (during starts and stops). In Fig. 2 it is clearly visible that the differences between the $SCOP_{month}$ and the $SCOP_{cycle}$ are higher in the warmer months. This is consistent with the calculated share of energy consumed during the operation in this mode compared to the total electrical energy consumption, as shown in Fig. 3.

Despite of the small power consumption in the stand-by mode (in the presented case, according to the manufacturer's data approx. 25 W, without taking into account the circulation pump), as a consequence of low heating demand of the building, the share of the energy consumed in this mode in the warmer months is very large, e.g. in April and May about 40%. This means that most of the time the heat pump was operating without the compressor switched on, consuming electrical energy without generating heat energy. The situation is quite different in the colder months (November – January), where the share of the energy consumed in stand-by mode ranges from 2.3% to 3.3% and only slightly affects the $SCOP_{month}$ in relation to the $SCOP_{cycle}$ (for example 3.21 to 3.33 for January). The electrical energy consumed in this mode lowers the average monthly value of $SCOP_{month}$, however it has no effect on the energy efficiency of the device when it is operating in the heating mode. Still, the achieved $SCOP_{cycle}$ is in many months lower than the estimates based on the relatively high ambient temperature and the working curves from the manufacturer (the COP_{decl} given in Table 1). For the monthly data this has been shown in Fig. 4. The value of this decrease can by expressed as the ratio of $SCOP_{month}/SCOP_{cycle}$. For the analyzed system it has been shown against the compressor run time in Fig. 5.

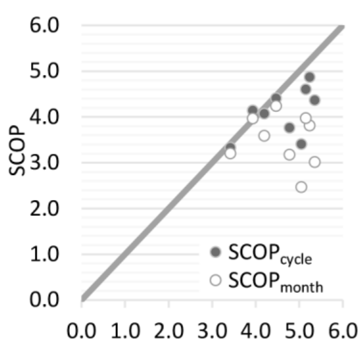

Fig. 4. $SCOP_{cycle}$ and $SCOP_{month}$ versus $SCOP_{decl}$.

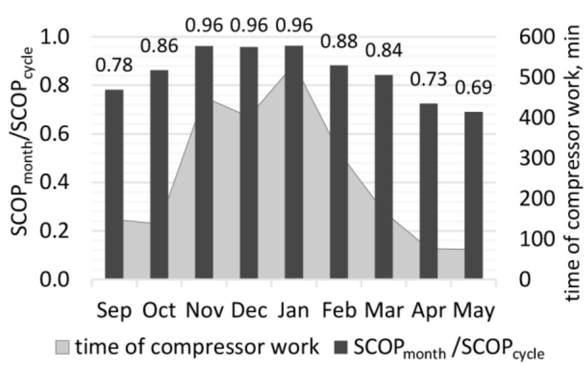

Fig. 5. The share of $SCOP_{month}$ in $SCOP_{cycle}$ in the analyzed months.

In the months of low ambient temperature, the heating demand of the building is higher, the cycles of active compressor work are usually longer and the intervals between them shorter. This results in a low share of operation time spent in the stand-by mode and a high average cycle length. The opposite situation occurs in the warm months, where short cycling and a significant time between the compressor operation lowers the average monthly value of SCOP. The impact of the average cycle length on the ratio of an actual

$SCOP_{cycle}$ to the $SCOP_{decl.}$ declared by the manufacturer is shown in Fig. 6. In particular, for very short cycles (below 10 min) a significant decrease in the energy efficiency is observed, reaching over 30%. Fig. 7 shows how does the ratio of $SCOP_{month}$ to $SCOP_{cycle}$ changes depending on the stand-by operation time as a percentage of the total time. The change is significant, e.g. for 60% heat load (which implies 40% of time in the stand-by mode) it equals about 25%. Similar results were observed in other studies [4, 5].

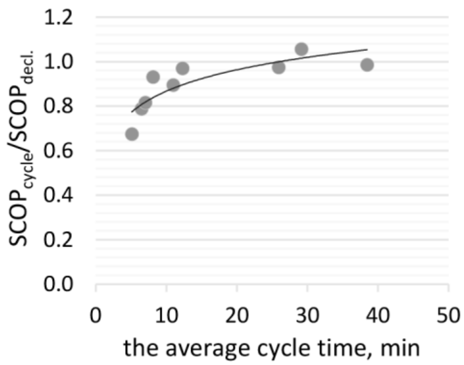

Fig. 6. The share of $SCOP_{cycle}$ in $SCOP_{decl}$ versus the average cycle time.

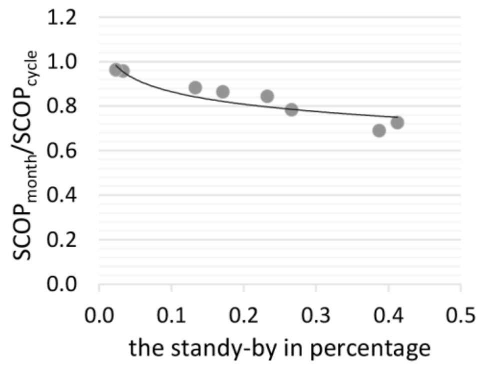

Fig. 7. The share of $SCOP_{month}$ in $SCOP_{cycle}$ versus the stand-by time as a percentage of total time.

3.2 Analysis of the heat pump operation

Fig. 8 shows the profile of the average hourly heat demand of the building and the heat pump's capacity during the entire month (January). Due to the various periods of ambient temperature conditions in this month, a wide range of the heat pump's operating parameters was observed. Most of the time, the capacity of the heat pump corresponds to the heat demand of the building. This is when the ambient temperature is below 5°C. However, there are also periods of very low heat demand of the building, outside of the regulation range of the heat pump (ambient temperatures above 5°C).

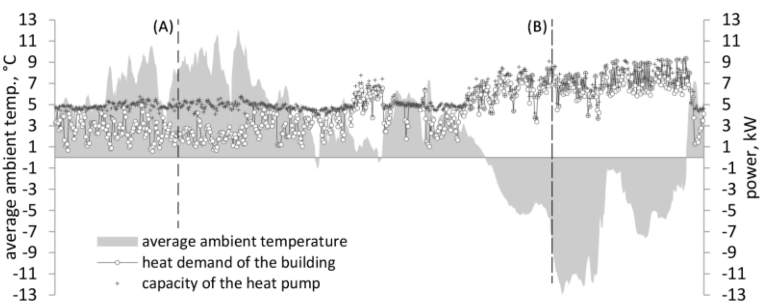

Fig. 8. The profile of the hourly heat demand of the building and the heat pump's capacity.

The difference in operation of the device in these extreme conditions is shown in Fig. 9, illustrating a period of 5 hours of constant work of the device. Picture (A) shows the operation of the heat pump in part-load conditions, with the capacity regulated by the frequency inverter and the on/off cycling. One can notice frequent compressor starts and the work with the minimum power for a short time. Part (B) illustrates the operation of the device with a high heating demand of the building. Compared to (A), longer cycles are observed, interrupted periodically due to the freezing of the heat exchanger. The heat pump performs the process of defrosting by reversing the refrigerant cycle, using heat accumulated in the heating system. The calculations demonstrating the effect of these factors on the COP of the device are shown for both periods in Section 3.3.

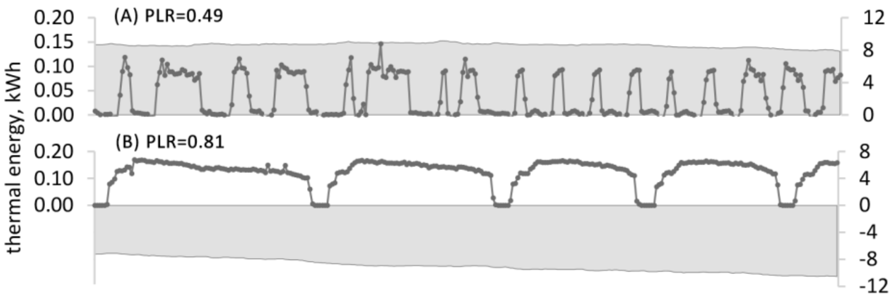

Fig. 9. The operation of the heat pump in two different part-load conditions.

3.3 Operation under partly load in the current European standards

The problem of operation under partly load is discussed in the current European standards for the calculation of seasonal energy efficiency – EN 14825 [6] and EN 15316-4-2 [7]. These documents have different approaches to the problem.

The standard EN 14825 defines the COP_{pl} taking into account the impact of the partial load factor as a function of PLR and Cc, described by equation (3). The PLR is the ratio of the building‘s heat demand (Q_H) to the current capacity of the heat pump (Q_{HP}). The Cc factor takes into account the decrease of the energy efficiency of the device due to the cyclic operation (on/off control) and power consumption in the stand-by mode. These factors are not considered when testing the device according to EN 14511 and during the calculation of $COP_{decl.}$. The Cc value should be determined individually for each heat pump. Unfortunately, it is not measured by the manufacturers. This often results in accepting the value suggested by the standard – Cc=0.9.

$$COP_{pl} = COP_{decl.} \cdot PLR \,/\, [PLR \cdot Cc+(1\text{-}Cc)] \qquad (3)$$

The standard EN 15316, on the other hand, defines the COP_{pl} basing on the value of heat demand of a building (Q_H), $COP_{decl.}$ and the amount of electrical energy consumed in the stand-by mode ($Q_{el,stand\text{-}by}$), as described by equation (4). Power of the device in the stand-by mode is usually contained in the manufacturer’s data. However, it does not cover the power of the circulation pump, which often occurs in installations. In addition, any imprecise calculation of the working time in the stand-by mode may result in significant inaccuracies in estimating the COP_{pl}.

$$COP_{pl} = Q_H \,/\, (Q_H/COP_{decl.}+Q_{el,stand\text{-}by}) \qquad (4)$$

A detailed analysis of calculating COP_{pl} has been made basing on the measurements of the periods A and B illustrated in Fig. 9. Table 2 shows the measurement results from both periods. The PLR and COP_{pl} were calculated. On this basis, the actual values of the reduction factor PLF and the parameter Cc have been determined. For the tested heat pump, the Cc factor is defined as 0.82. Table 3 shows the actual values of working time (τ), compressor run time (τ_{on}), compressor off time (τ_{off}) and the electric energy consumption during the compressor work ($Q_{el,cycle}$), stand-by ($Q_{el,stan\text{-}by}$) and defrosting ($Q_{el,defr.}$).

Table 2. The measurement results for periods A and B (two different part-load conditions).

	T_a	T_{in}	$COP_{decl.}$	Q_H	Q_{el}	Q_H	Q_{HP}	PLR	COP_{pl}	PLF	Cc
	°C	°C	-	kWh	kWh	kW	kW	-	-	-	-
period (A)	8.6	30.9	4.80	11.22	2.77	2.24	4.58	0.49	4.05	0.84	0.82
period (B)	-9.0	31.6	2.60	37.19	14.91	6.69	8.29	0.81	2.49	0.96	0.82

Table 3. The electrical energy used in periods A and B (two different part-load conditions).

	$Q_{el,cycle}$	$Q_{el,stand\text{-}by}$	$Q_{el,defr.}$	τ	τ_{on}	τ_{off}	$Q_{el,stand\text{-}by}$
	kWh	kWh	kWh	min	min	min	W
period (A)	2.4	0.4	0.0	300	141	159	136
period (B)	14.3	0.0	1.04	300	249	51	-

The measured data were used to calculate the value of hypothetical COP_{pl} according to the standards [6, 7]. In the case of the standard EN 14825, basing on the actual PLR value and the assumed in the standard parameter Cc=0.9, the discrepancy between the calculated and the actual COP_{pl} was significant, as shown in Table 4. The inaccuracies were at the level of 2% and 7%, for period B and A respectively. In the case of the standard EN 15316, basing on the manufacturer's information about the stand-by energy and the actual operation time in this mode, the discrepancies between the calculated and the actual COP_{pl} were 4% and 13% respectively.

Table 4. The discrepancy between the calculated and the actual COP_{pl} for periods A and B.

	PLR	Cc	COP	Error
	-	-	-	-
EN 14825, A	0.49	0.9	4.34	7%
EN 14825, B	0.81	0.9	2.54	2%

	$Q_{el,stand\text{-}by}$	$Q_{el,stand\text{-}by}$	COP	Error
	W	kWh	-	-
EN 15316, A	25	0.066	4.67	13%
EN 15316, B	25	0.021	2.60	4%

Fig. 10 summarizes the value of PLF calculated basing on the Cc according to the standard EN 14825 and the approximate value of this parameter according to EN 15316 including the PLF for the actual C_C based on the measurements. This is noticeable that the actual value of this parameter is much lower than the one assumed in the calculations. The problem of estimating the degradation coefficient is also analyzed in the work [3] where the importance of this issue was emphasized.

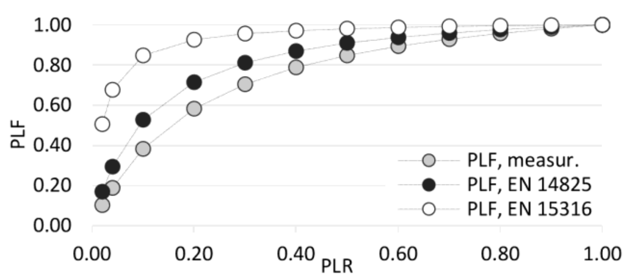

Fig. 10. The PLF factor: measured and according to the standards EN 14825, EN 15316.

Fig. 11 shows how the estimated values of COP_{pl} fit the actual COP of the unit in January 2014. Analyzing the measurements themselves, the decrease of the unit's energy efficiency due to the on/off control is noticeable. It occurs when the heat demand is outside of the range of the frequency inverter operation, i.e. in ambient temperatures above 5°C. Furthermore, the problem of correct estimation of the COP_{pl} is clearly visible in these methods. The values of COP_{pl} calculated according to the standards (C_C=0.9 and the stand-by mode power 25 W) are shown on the left side of Fig. 10. Better precision is obtained when calculating according to the standard EN 14825. The standard EN 15316-4-2 does not represent the actual work of the tested heat pump. Better accuracy is achieved when the actual measured data is used in the calculations for both standards: Cc=0.82 and the stand-by power consumption of 136 W (including the circulation pump). Although the accuracy is improved, still none of the methods reflects correctly the trend of COP_{pl} of the unit.

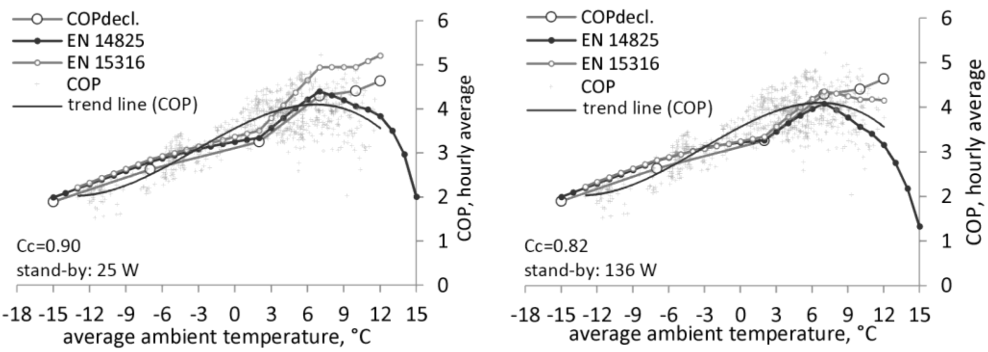

Fig. 11. The estimated values of COP_{pl} and the actual COP in January 2014.

4 Conclusions

Basing on the analysis of the SCOP in multiple following months it was found that there is a significant difference between the operational energy efficiency of the heat pump and the efficiency declared by the manufacturer. The analysis of the hourly data allowed for the calculation of the actual value of Cc, which describes the level of reduction of COP_{pl} in relation to the declared COP_{decl}. For the analyzed device, the value of Cc is 0.82 and it is lower than the standards EN 14825 and EN 15316 suggest. The COP calculation methods described in these standards ignore the impact of the defrosting process and the circulation pump's energy consumption when the compressor is turned off. Probably both methods assumed that the influence of these parameters is included in the COP obtained during the tests according to EN 14511. In fact, these tests are performed in the steady-state conditions and do not properly reflect these issues. It is necessary to either change the test procedure of the COP or the calculation procedure of the COP_{pl}.

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