Quasi-dynamic model of the energy efficiency for an air-to-water heat pump

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Abstract. The energy efficiency of air-to-water heat pump operating in an actual heating installation depends on many factors. In order to create a reliable model of the unit, it is necessary to include as many of them as possible. Unfortunately, the most common data provided by heat pump manufacturers are based on tests performed in accordance with the EN 14511 standard [1]. These tests are performed in steady-state conditions and do not provide reliable information on the impact of dynamic effects on the energy efficiency of the device. The solution may be the tests in quasi-dynamic conditions. The article presents the possibility of creating the characteristics of an air-to-water heat pumps based on operational data. The accuracy of the created model has been compared with the characteristics resulting from measurements in steady state conditions. It has been confirmed that dynamic test data, after proper selection, will allow to determine the characteristics of repeatable parameters and this can be an alternative to tests performed in fixed conditions.

1 Introduction

The heating power and the coefficient of performance (*COP*) of air-to-water heat pumps are influenced by many factors. The most important of these are the parameters of the lower and upper sources, primarily the ambient temperature (T_a) and the supply temperature of the heating system (T_{in}). Correct determination of the actual *COP* of the heat pump in various operating conditions is crucial to determine its seasonal coefficient of performance (*SCOP*). Typically, the basis for the calculation of the instantaneous *COP* value and the heating power (Q_{HP}) is data provided by the heat pump manufacturer, which is based on tests performed in accordance with the EN 14511 [1] standard. At the moment, this is the most accurate and common source of information for the purpose of air-to-water heat pumps work simulations. The tests proposed in the standard [1] take into account changes of the heating system inlet temperature (upper source), changes in the ambient temperature (lower source) and the process of defrosting the heat pump's evaporator. Values of the declared coefficient of performance (COP_d) resulting from tests in accordance with the [1] standard also include the electrical energy supplied to the compressor, fans, circulation

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pump (if installed in the heat pump) and automation during the active cycle of the device. The tests are conducted at several measuring points. Most often, the data set includes three upper-source temperatures: 35°C, 45°C, 55°C and five to six lower source temperatures, e.g. -15°C, -10°C, -7°C, 2°C, 7°C, 20°C. Based on these points, the COP_d and the device heating power (Q_{HP}) calculation models should be developed. The model proposed by T. Afjei is a commonly used method for this purpose [2, 3]. A significant problem, however, is the extrapolation beyond the range of measurement points. The model assumes that outside this range the characteristic will be linear. The results obtained may differ significantly from the actual ones, which was observed in the works [3, 4]. It should be emphasized, that the transformation from single points measurements into a one-function characteristic is an important advantage of this solution. In the original method, it is a polynomial of the second degree with two variables. Regardless of the *COP* and the heating power model used, the most important issue is the quality of the input data. In general, the tests performed in accordance with EN 14511 [1] are not entirely appropriate for this purpose. The main problems concern: insufficient number of measuring points (e.g. no data for inlet temperature below 35°C), constant temperature difference on the condenser (5 K), no evaluation of the frequency inverter influence on the COP_d and heating power of the device, as well as the lack of evaluation of the partial load effect and the way of the device's power adjustment to its efficiency. The COP_d value determined on the basis of these tests concerns only the situation when the heat pump works at full heating load in the active cycle. In more precise simulations, corrections to take into account the effect of these dynamic conditions on the efficiency of the heat pump are taken into account [5–8].

The article presents the possibility of creating the characteristics of an air-to-water heat pump based on operational data. As mentioned above, tests in steady state conditions do not provide reliable information about the impact of dynamic effects on the energy efficiency of these devices. The solution may be the tests in quasi-dynamic conditions. In order to check this possibility, preliminary works described in this article are aimed at preparing the energy efficiency model of the device based on the collected measurement data. The accuracy of such model was compared with the characteristics resulting from measurements in steady state conditions.

2 Description of the measurements

The measurement data of the heating installation's operation was used to calibrate the efficiency model of the heat pump. In this case, 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 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_d), measured according to EN 14511 standard, are shown in Table 1.

Table 1. Declared heating capacity (Q_{HP}) and COP_d of the unit according to EN 14511 [1].

	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_d	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

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 4.4 kW.

The basic results of the heating system measurements in the months from IX to V are summarized in Figures 1 and 2. Figure 1 shows the heating demand of the building (Q_B) and the electricity consumption of the heat pump system (Q_{EL}) in a monthly step.

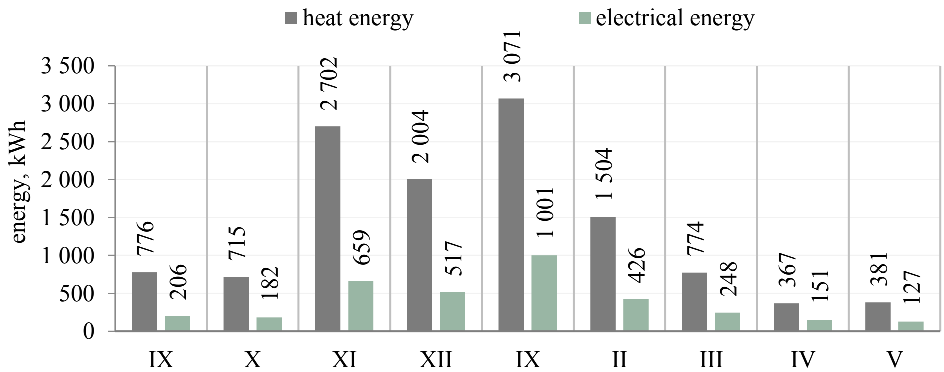

Fig. 1. Thermal energy delivered to the building and electricity consumption of the heat pump in the months from IX to V (measurement data).

Figure 2 contains the monthly *COP* (calculated on the basis of measurement data shown in Figure 1), COP_d and the average ambient temperature ($T_{a,avg}$) data. It is worth noting that the dependence of the monthly *COP* to the ambient temperature is not as it appears directly from the COP_d model under full load. In the transitional periods (e.g. IX, III) the decrease in the *COP* value relative to the declared value (COP_d) is the result of operating in the part load conditions. This fact was discussed in previous research regarding the impact of work under partial load on the energy efficiency of an air-to-water heat pump [7, 8].

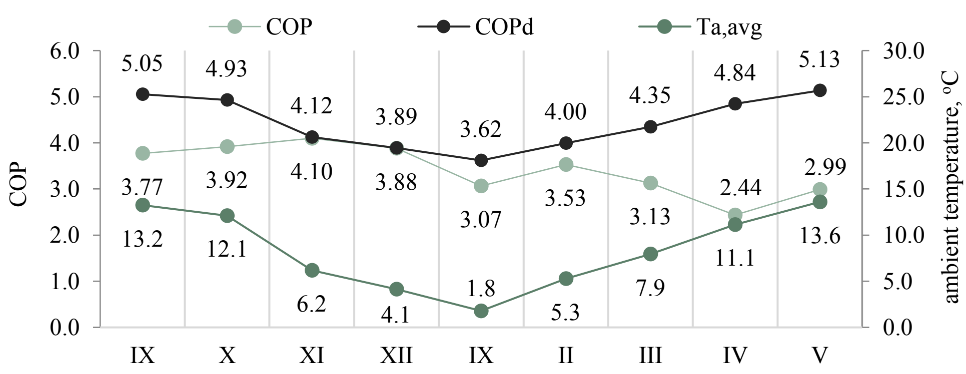

Fig. 2. Average monthly values of the average ambient temperature ($T_{a,avg}$), average monthly COP_d and *COP* (measurement data).

3 The model development

3.1 Model of the energy efficiency for an air-to-water heat pump

In this article, the model of T. Afjei [2] was used to describe the energy efficiency of the heat pump. The independent variables for COP model are the inlet temperature (T_{in}) and the ambient temperature (T_a). In the original method, the parameters A-F of the function described by equation (1) are determined by statistical analysis (by multiple regression) of measurement data obtained in steady state test conditions performed in accordance with the EN 14511 [1] standard.

$$COP_d = A + B \cdot T_a + C \cdot T_{in} + D \cdot T_a \cdot T_{in} + E \cdot {T_a}^2 + F \cdot {T_{in}}^2 \tag{1}$$

Additionally, in order to include the influence of dynamic effects on the change of the energy efficiency of the heat pump, the correction factor described in the article [7] was applied. Correction of the COP_d value allows taking into account the effect of the partial load of the heat pump (described by the *PLR* parameter) based on the part load factor (*PLF*) value. The reduced energy efficiency value is called COP_{pl} (*COP* in part load conditions) and is determined in accordance with the equation (2).

$$COP_{pl} = COP_d \cdot PLF \tag{2}$$

where the *COP* reduction factor (*PLF*) is calculated according to equation (3). The *PLR* value was calculated as the ratio of hourly average building heating demand (Q_B) to the hourly average heating power of the heat pump (Q_{HP}).

$$PLF = 1 + a \cdot ln\left(PLR + e^{(-1/a)}\right) \tag{3}$$

The energy efficiency model described by equation (1) was developed on the data presented in Table 1. The calculations performed for the heat pump tested proved that the T_a^2 and T_{in}^2 variables are not statistically significant for this function (the p value is higher than 0.05). The coefficient of determination (R^2) for the estimated function is 0.99. Parameter a in function (3) is also determined on the basis of statistical analysis. A detailed description of how to determine this value was presented by the authors in the article [7]. Its value determined in relation to the COP_d function developed for measurement data under steady state operating conditions is 0.28. The proportion of the dependent variable translated by the model R is 0.81. The final form of the model developed on the basis of test data under steady state working conditions according to the standard EN 14511 [1] is therefore (4).

$$COP_{pl} = (5.67 + 0.18 \cdot T_a - 0.067 \cdot T_{in} - 0.0023 \cdot T_a \cdot T_{in}) \cdot \\ \cdot \left(1 + 0.28 \cdot ln\left(PLR + e^{(-1/0.28)}\right)\right) \tag{4}$$

In the case of determining the energy efficiency characteristics of a heat pump operating in dynamic conditions, the same equations (1), (2) and (3) were used. However, the statistical analysis is based on the hourly measurement data of the work of the heating installation described in section 2. Function (1) applies to the operation of the heat pump under a significant heating load, therefore hourly average measurement data with a PLR value higher than 0.8 were selected to determine the A-F parameters. As in the case of previous calculations performed for working in steady state conditions, the T_a^2 and T_{in}^2 variables proved to be irrelevant to this function (the p value is higher than 0.05). The coefficient of determination (R^2) for the estimated function is 0.90. The correction of this characteristic related to work under partial load conditions was determined from equation (3). In this case, the key range of the inlet temperature (T_{in}) varied from 0°C to 10°C and the PLR from 0.17 to 0.80 were selected. The proportion of the dependent variable translated by the model R is 0.86. In this case, the a parameter is 0.31. The difference between this value and the previously determined for the measurements in steady state working conditions (0.28) results from different *PLF* values used for parameter estimation. The change in the PLF value is related to the different COP_d value determined from function (1). The final form of the model developed on the basis of measurement data of a heat pump operating in dynamic conditions is therefore (5).

$$COP_{pl} = (2.64 + 0.49 \cdot T_a + 0.031 \cdot T_{in} - 0.010 \cdot T_a \cdot T_{in}) \cdot \cdot \left(1 + 0.31 \cdot ln\left(PLR + +e^{(-1/0.31)}\right)\right) \quad (5)$$

3.2 Scope of simulations

The purpose of the calculations was to precisely determine the SCOP value and the electricity consumption of the heat pump system. For the comparison, the commonly used methods of taking into account the partial load were used: according to EN 14825 [5] (Simulation 1) and EN 15316 [6] (Simulation 2). The key parameters of these functions have been determined on the basis of measurement data. The results obtained from four simulations were compared with the results of measurements. The characteristics of each simulation are described below:

- Simulation 1: calculated for a heat pump model determined on steady state test data and for the reduction coefficient *Cc* = 0.8 (*Cc* value was determined based on measurements of heat pump operation [7]).
- Simulation 2: calculated for a heat pump model determined on steady state test data and for $Q_{el,s\text{-}by}$ = 91 W ($Q_{el,s\text{-}by}$ value was calculated based on measurements of heat pump operation [7]).
- Simulation 3: calculated for a heat pump model determined on steady state test data and for a reduction coefficient *a* = 0.28 (*a* value was determined based on measurements of heat pump operation [7]).
- Simulation 4: calculated for a heat pump model determined on measurement in dynamic conditions data and for a reduction coefficient *a* = 0.31 (*a* value was determined based on the operational characteristic of the heat pump).

In order to compare the results of the simulation (*s*) with the measurements (*m*), the errors of estimating the monthly values of *SCOP* ($\delta SCOP^m$) and hourly *COP* values (δCOP^h) were calculated according to the equations (6) and (7).

$$\delta_{SCOP}{}^{m} = \frac{SCOP_m - SCOP_s}{SCOP_m} \quad (6)$$

$$\delta_{COP}{}^{h} = \frac{\sum_1^i \left|\frac{COP_m - COP_s}{COP_m}\right|}{i} \quad (7)$$

3.3 Results and discussion

The results of simulations and measurements for individual months and the entire year are summarized in Table 2. Error values are presented in Figure 3 ($\delta SCOP^m$) and Figure 4 (δCOP^h).

Comparing the results of the monthly values of *SCOP* for simulations 1, 2 and 3 (that is for the COP_d model determined on steady state test data) shows that simulation 3 allowed the most precise description of the heat pump's operation. With two methods proposed in the standards, the method based on the *Cc* reduction coefficient (simulation 1) allows to achieve more precise results. In addition, it should be noted that all models describing the impact of working in part load conditions have been calibrated with measurement data. Using the manufacturer test data instead of operating data can bring significant overstatement, especially during transient periods. It is worth noting that the *Cc* coefficient given by manufacturers oscillate in the range of 0.96–0.99. Authors of the publications [7, 8, 10] suggest that such an approach to the overall impact of operating in part load conditions is inappropriate and leads to errors in simulation results. For this reason, research has been carried out and the model described in equation (3) has been developed. The model allows to obtain the best fit among the respondents to the operational data. In addition, a more accurate characteristic of a heat pump operating under full load was developed by replacing the test data in steady state conditions by dynamic, measurement data. This allowed for further improvement of the results. Errors in monthly *SCOP* values, obtained with this modelling method do not exceed 4.5%. The average errors of hourly COP^h values in particular months are shown in Figure 4. The average errors for simulations 1 and 2 are respectively 13.5% and 12.3%. When analyzing the results for individual months, it should be noted that the highest errors relate to the transitional period (e.g. III, IV). Simulation 3 is characterized by slightly lower errors of hourly COP^h values, with the annual average of 11%. Simulation 4 generates the lowest errors, the annual average is 8.7%.

Table 2. Simulation results and measurements for the heat pump system.

Month	Measurement			Simulation 1		Simulation 2		Simulation 3		Simulation 4	
	Q_B	*SCOP*	Q_{EL}	*SCOP*	Q_{EL}	*SCOP*	Q_{EL}	*SCOP*	Q_{EL}	*SCOP*	Q_{EL}
IX	772	3.77	205	3.40	227	3.90	198	3.48	222	3.93	196
X	715	3.92	182	3.69	194	4.02	178	3.69	194	4.07	176
XI	2 699	4.10	658	4.00	675	4.05	666	3.98	677	4.20	642
XII	2 004	3.88	516	3.82	525	3.92	511	3.80	528	3.97	505
I	3 071	3.07	1 002	3.32	926	3.38	909	3.30	931	3.02	1 018
II	1 504	3.53	426	3.54	425	3.84	391	3.48	432	3.69	408
III	772	3.13	247	2.92	265	3.50	221	2.89	267	3.09	250
IV	366	2.48	148	2.19	167	2.93	125	2.31	158	2.54	144
V	381	3.00	127	2.60	146	3.30	115	2.75	139	3.05	125
year	12284	3.50	3 511	3.46	3 550	3.71	3 315	3.46	3547	3.62	3 391

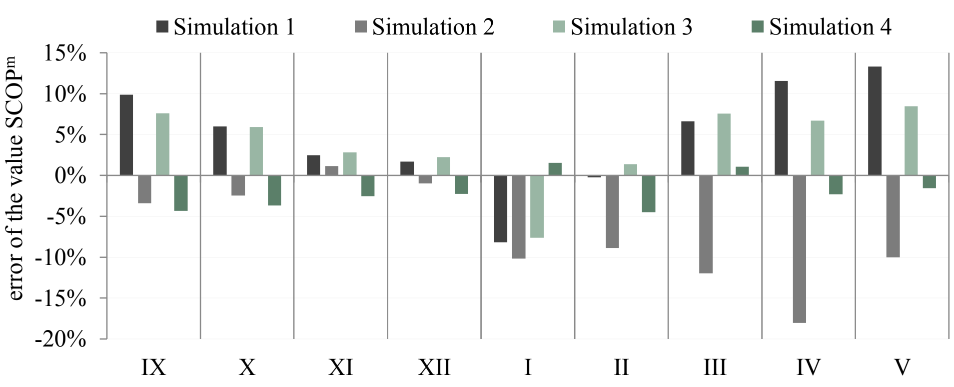

Fig. 3. Errors of monthly $SCOP^m$ values (simulation results in relation to measurement data).

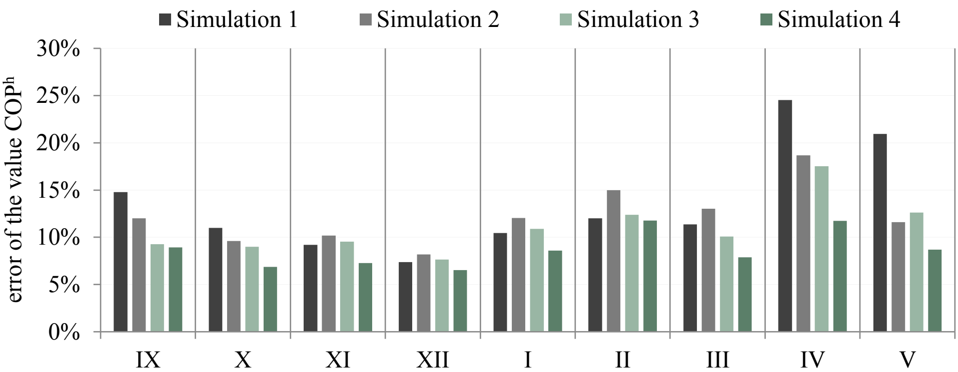

Fig. 4. Errors of hourly COP^h values in individual months (simulation results in relation to measurement data).

Figure 5 presents the hourly COP_h values for measurement data and simulations 1–4 for two selected months: January and March. Plots allow to observe the effects of model calibration. Simulations 1, 2 and 3 are based on static characteristic of COP_d and three different models of operating in part load conditions calibrated with measurement data. Correct results are visible for simulations 1 and 2. A detailed analysis of errors showed that simulation 2 has slightly better accuracy. Results of COP^h values based on simulation 3 are overestimated. Simulation 4, determined on the basis of operating characteristics of the *COP*, better describes the actual work of the air-to-water heat pump. The model takes into account the influence of changes in air humidity, a different defrost cycles frequency and the effect of inverter frequency change on the *COP*. It should be noted that this impact has not been demonstrated directly due to the insufficient scope of changes of these parameters during the operation of the device. Under the influence of these factors the parameters of variables T_{in} and T_a have been corrected.

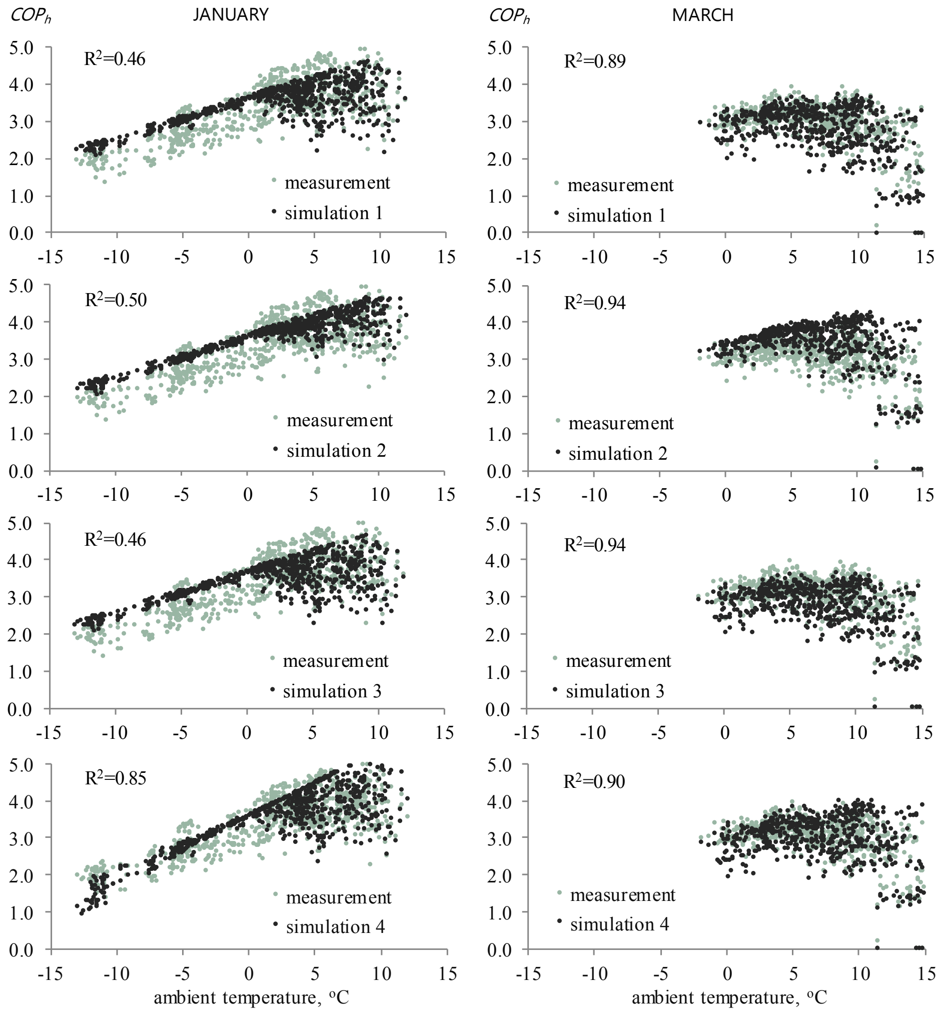

Fig. 5. The hourly COP^h values in January and March (simulation results in relation to measurement data).

4 Conclusion

The article discusses the possibility of creating an air-to-water heat pump energy efficiency model based on operational data. Tests in steady-state conditions do not provide much information about dynamic effects affecting the energy efficiency of these devices, because they are conducted in conditions that rarely exist in heating installation. Tests under quasi-dynamic conditions can be an alternative to these procedures. In this article, it was confirmed that dynamic test data, after proper selection, will allow to determine the parameters of the *COP* characteristics. Of course, the research for procedure of such a test is necessary. For example, in this study, the influence of air humidity and compressor rotation reduction is not directly determined. However, comparable device models were developed on the basis of dynamic measurement data and data from test in steady-state conditions. The simulation results carried out on both characteristics were compared. The agreement between the results of the dynamic model and the measurement results was better than for the data developed in static conditions.

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