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ANALYSIS OF AIR-TO-WATER HEAT PUMP IN COLD CLIMATE: COMPARISON BETWEEN EXPERIMENT AND SIMULATION

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Abstract. Heat pump systems are promising technologies for current and future buildings and this research presents the performance of air source heat pump (ASHP) system. The system was monitored, analysed and simulated using TRNSYS software. The experimental data were used to calibrate the simulation model of ASHP. The specific climate conditions are evaluated in the model. It was noticed for the heating mode that the coefficient of performance (COP) varied from 1.98 to 3.05 as the outdoor temperature changed from –7.0 ºC to +5.0 ºC, respectively. TRNSYS simulations were also performed to predict seasonal performance factor of the ASHP for Vilnius city. It was identified that seasonal performance prediction could be approximately 15% lower if frost formation effects are not included to air-water heat pump simulation model.

Keywords: air source heat pump, coefficient of performance, experiment, simulation, seasonal performance factor, TRNSYS.

Introduction

Renewable energy technologies, combined with energy efficiency measures, give a viable solution to countering the effects of global warming. Generally, heat pumps offer one of the most energy-efficient ways to provide heating and cooling in many applications, as they can use renewable heat sources in our surroundings (Omer 2008). Ground source and air source heat pumps (ASHPs) are widely used for energy supply in buildings. In contrast to ground source heat pumps, ASHP systems are compact in design, relatively inexpensive, and usually provide consistent performance under mild winter conditions. The major concern of ASHPs is that their heating capacity diminishes as the outdoor air temperature drops below 0 ºC (Safa *et al.* 2015).

Evaporator frosting and subsequent need for defrosting at low ambient temperatures at ASHPs causes reduced energy efficiency and heating shut down (Zhang *et al.* 2012). Different solutions to delay frosting and improve defrosting efficiency are used. However, the reverse cycle defrosting is currently the most widely used defrosting method (Dong *et al.* 2012). The energy consumption due to the defrosting should be taken into account in the evaluation of the ASHP performance but the calculation methods proposed by European standards ignore this effect (Vocale *et al.* 2014).

From the literature available, there are reported many studies on ASHP systems and a defrosting process in them. Qu *et al.* (2012) developed a semi-empirical mathematical model to analyze the effects of downwards flowing of the melted frost along the outdoor coil surface of an ASHP. This model allowed to study frost melting at upper and lower level circuits. Dong *et al.* (2012) carried out an experimental study on defrosting energy supplies and consumptions during a reverse cycle defrost operation. Their experiments showed that the heat supply from indoor air contributed to 71.8% of the total heat supplied for defrosting and 59.4% of the supplied energy was used from melting frost. However, taking heat away from indoor air can adversely affect indoor thermal comfort level. Wang *et al.* (2013) presented the cross hot-gas bypass defrosting method. This method could overcome the main disadvantage of the reverse-cycle defrosting. Jiang *et al.* (2013) proposed a novel defrosting control method that provided a possible alternative to initiate a defrosting operation when needed. Vocale *et al.* (2014) investigated the effect of the outdoor air temperature and relative humidity on the performance of an ASHP when the reverse-cycle defrosting is considered. Their results confirmed that the choice of the relative humidity value is very crucial in order to properly evaluate the frost accumulation rate on the cold surface of the external coil and consequently the effect of defrost on the heat pump performance.

Despite the large number of studies on the performance of ASHPs, defrosting necessarily causes the periodic interruption of outdoor heating and degradation in winter heating efficiency. Therefore, the simulation models should reflect as possible the actual processes in such systems. The main aim of this study is to make the TRNSYS model suitable for ASHP system analysis. It should offer possibilities to be modified to accommodate different simulation needs. Experiments are carried out to validate the model under different conditions. Experimental data is gathered in Laboratory of Building Energy and Microclimate systems (BEMS) in Vilnius Gediminas Technical University. By comparing the measured and the simulated parameters, we conclude that this model can accurately simulate the operation of the ASHP. Finally, the developed model is used to investigate the thermal performance of the ASHP system.

System description and experimental conditions

Air-to-water heat pump system is used in BEMS Laboratory in Department of Building Energetics of Vilnius Gediminas Technical University. This laboratory (113 m^2) is equipped with series of renewable energy technologies and equipment those use renewable energy. When the ASHP operates in heating mode during the cold period, firstly it heats the storage tank. The mixture of water and ethylene glycol of storage tank is heated by the ASHP close circuit using a heat exchanger. The temperatures are controlled by temperature controller.

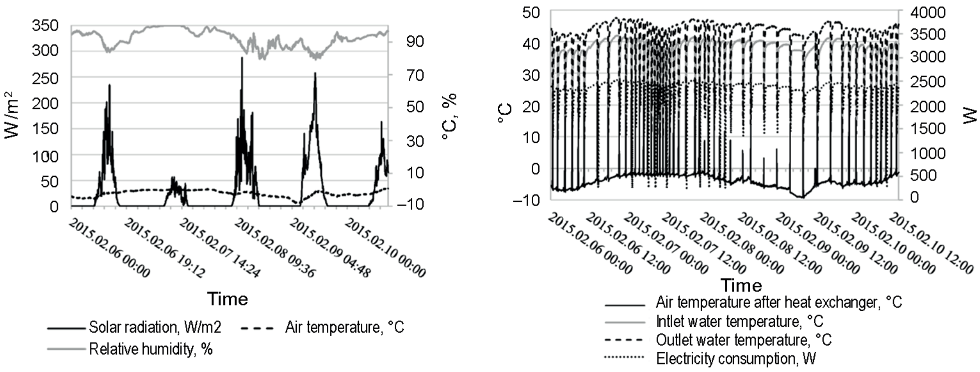
A variable-speed low temperature ASHP is selected. The ASHP was manufactured by Aermec ANK020 HP, and has a rated COP of 2.57 in heating mode (heat capacity of 7.40 kW and electrical power of 2.88 kW). Acoording to the manufacturers‘s specifications, the rated COP is determined at fixed outdoor temperature (7/6 ºC) and water temperature (47/55 ºC).

Table 1. Accuracy of various sensors used in the experiments

Sensor name	Measuring range	Accuracy of the measured values
Temperature sensor	–40 to 100 °C	Class B: $\Delta t = \pm(0.3 + 0.005 \cdot (t))$ (Temperature accuracy 2008)
Flow meter	25 to 5000 l/h	Class 3: $\Delta V = \pm\,(3 + 0.05 \cdot V_p/V)$, % (EN 1434-1: 2006)
Alternating current sensor	2 to 20 A	±4.5% of full scale

The systems and their models were tested in the BEMS laboratory at local weather conditions. For the ASHP system performance monitoring; temperature, flow rate, and power consumption sensors were installed. The range of accuracy of various sensors and measurements used for experiments is given in Table 1. To ensure accurate data was being collected, preliminary test data was made. While the data collection was taking place, parallel TRNSYS model of the ASHP system was developed and later validated using the actual performance data. The heating data collection commenced on February 06 and ended on February 10 (2015). During test period, the ambient temperature changed from –8 ºC to 1 ºC. Figure 1a shows climatic conditions when the ASHP operates. As can be seen from Figure 1a, values of outdoor air temperature, solar radiation and relative humidity were not stable. The measured performance parameters (air temperature after heat exchanger, inlet and outlet water temperatures and electricity consumption) of ASHP are plotted in Figure 1b. Calculations of electricity consumption were performed using data of the current in the three phases.

Where V_p is the permanent flow rate (it is the highest flow rate at which the heat meter shall function continuously without the maximum permissible errors being exceeded). For the calculations, it is used value of 2.5 m^3/h.



(a) (b)

Fig. 1. Recorded weather conditions during measurements (a); performance parameters of the ASHP (b)

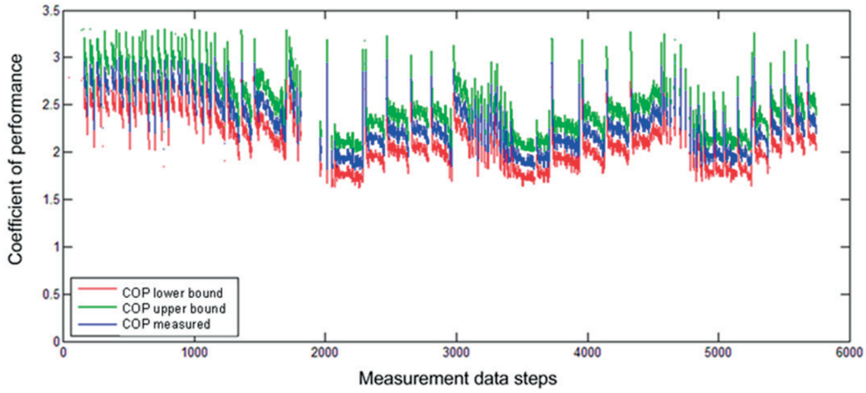


Fig. 2. Uncertainty rages of the calculated COP

Every measurement is always subject to an uncertainly which arises from the layout and method of measurement, the measuring equipment and taking the reading. For a value calculated from measurements of several individual values the uncertainly of the resultant value is determined by applying the law of propagation of uncertainties of the individual values measured. It is assumed that the uncertainties are independent of one another and that each follows a normal Gaussian distribution. Thus, for instance, two randomly selected values might both have an uncertainty in the same direction. However, the uncertainty for each parameter will lie within the limits stated in the previous sections. If the operating data fluctuate during the measurement period, the effect of this on measured results shall be taken into consideration (EN 12599:2000).

The coefficient of performance (COP) of the heat pump is evaluated by:

$$\mathrm{COP} = \frac{Q}{P_e}, \tag{1}$$

where: Q – heat output by heat pump, kWh; P_e – consumed electricity, kWh.

This formula could be rewritten as follows:

$$\mathrm{COP} = \frac{\dot{V} \cdot \tilde{n} \cdot c_p \cdot (t_{out} - t_{in})}{\sqrt{3} \cdot I \cdot U \cdot \cos(\varphi) \cdot 3.6}, \tag{2}$$

where: $\dot{V}$ – volumetric flow rate, m^3/h; ρ – fluid density, kg/m^3; c_p – specific heat capacity of fluid, kJ/(kg·K); t_{out} – fluid temperature at the outlet, ºC; t_{in} – fluid temperature at the inlet, ºC; I – electrical current, A; U – voltage, V; cos(φ) – power factor.

Mains voltage is taken into account as a constant value equal to 230 V. Current is calculated as the average of the current in each phase. The power factor is assumed to be 0.82. The heat transfer fluid density and specific heat are taken as constants in this calculation.

The measurement reliability of COP is in the confidence interval of 95% and it can be expressed as the function:

$$\Delta\mathrm{COP} = f(\dot{V}, t, I). \tag{3}$$

The calculated COP variation for experiments is shown in Figure 2.

Using the uncertainty methodology, the maximum uncertainty of the COP is ±9.50% and the minimum – ±8.55% when the ASHP operates under prescribed weather conditions (see Fig. 1). This indicates that measurements were carried out with reasonable uncertainty (ΔCOP does not exceed 10 %).

TRNSYS model and its calibration

In this section, a TRNSYS model is presented to predict the performance of ASHP. TRNSYS software is dynamic simulation software, first developed by the University of Wisconsin-Madison Solar Energy laboratory (Klein 2004). In TRNSYS, a model is developed within the Simulation Studio environment a graphical user modeling interface. This software has a modular structure that was designed to solve complex energy systems problems by breaking the problem down into a series of smaller components known as „Types“ – predefined components and algorithms which model the behavior of common systems. The model is executed by TRNEXE, an algebraic and differential equation solver iteratively computes the system state at each time step – at this case 1 min as necessary to reflect control time constant. Because the model is stored in text format, it can be parameterized by a scripting languages such as Matlab (Matlab 2012).The TRNSYS simulation model of ASHP operation is depicted in Figure 3.

Model structure could be divided into three main blocks – heat consumer, accumulation and distribution and heat source. While the task for this study was to increase the prediction accuracy of air source heat pump model – only last model block was calibrated according to measured data. Other two blocks were used to examine seasonal behavior of the model and predict possible deviations due to

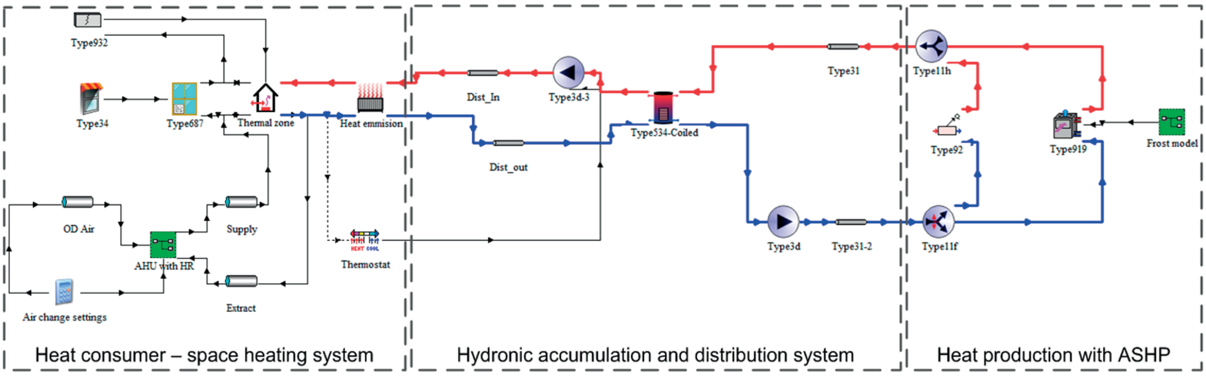


Fig. 3. TRNSYS model of ASHP system

standards allowed assumption to neglect evaporator frosting and defrosting process.

For heat pump performance modeling Type 919 from TESS libraries, which represents operation based on performance map, is taken from manufacturer documentation (Airmec 2008). Supplied data contains catalog data of heating capacity and consumed power for the given inlet air and water temperatures. Calibrating this Type with inlet and outlet temperatures of water and air streams, mismatch between measured and simulated results could easily go below 5%.

To reach higher accuracy level of model and to have possibility to investigate different defrosting strategies impact on frosting duration, model based on experimental data was used. Frosting model expresses time from one to other defrosting period based on amount of condensation rates on evaporator and inlet temperature data. Such data driven model was created by using MATLAB Curve fitting tool from measurement data. Additional boundaries due to thermodynamical and psychrometric limits for the three dimensional surface were added. Filing these bounds with experimental data, polynomial function was obtained and used to calculate time till the critical frost amount, at which occurrence defrosting procedure must be executed. It was only possible way to simulate this operation aspect because exact algorithm for this sequence of normal operation and defrosting is unknown due to commercial restrictions of heat pump manufacturer.

As at this case, an Airmec heat pump is using reverse defrosting method, TRNSYS type 919 is switched off for defrosting period. According to measurements average defrost duration lasts approximately 11 minutes. During this period heat is taken out form the storage tank to reflect energy consumed due to defrosting. This additional operation mode has high influence on seasonal performance factor (SPF). Model used at this research is combined from tabulated data based on heat pump component and frost formation model written in MATLAB code. Due to simplification reasons, COP degradation during frost formation on evaporator is not taken into account. Only duration till total frosting is calculated (see Fig. 4) and defrosting procedure with reverse cycle is simulated.

Function determining the state of normal operation or defrosting mode is explained in scheme shown in Figure 5.

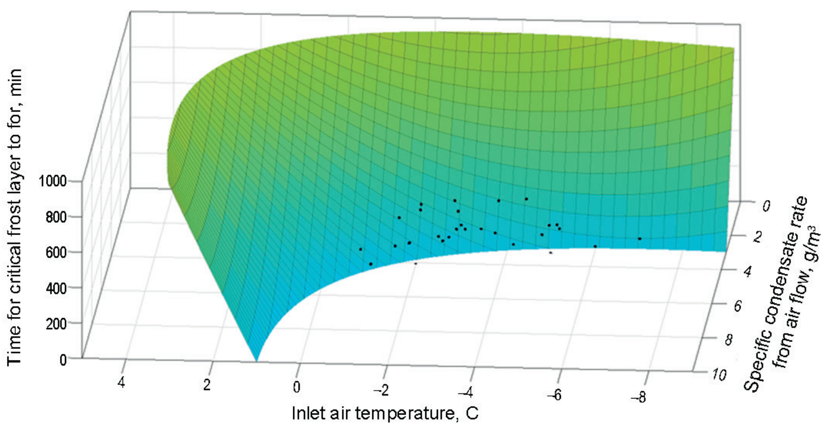


Fig. 4. Polinomal surface used to define time till defrosting according to ambient temperature and absolute humidity

During defrosting operation mode, Type 919 is stopped and water flow from the storage tank is cooled down at rate needed to defrost partialy frozen evaporator. The water flow is cooled at rate equal to energy amount needed to melt mass of frost accumulated on the evaporator. In order to take into account that this amount could be higher in non ideal conditions, cooling capacity is corrected by an additional factor. Such configuration helps to evaluate short term efects which influence the degradation of seasonal performance factor (SPF) of ASHPs conected to space or domestic hot water heating systems.

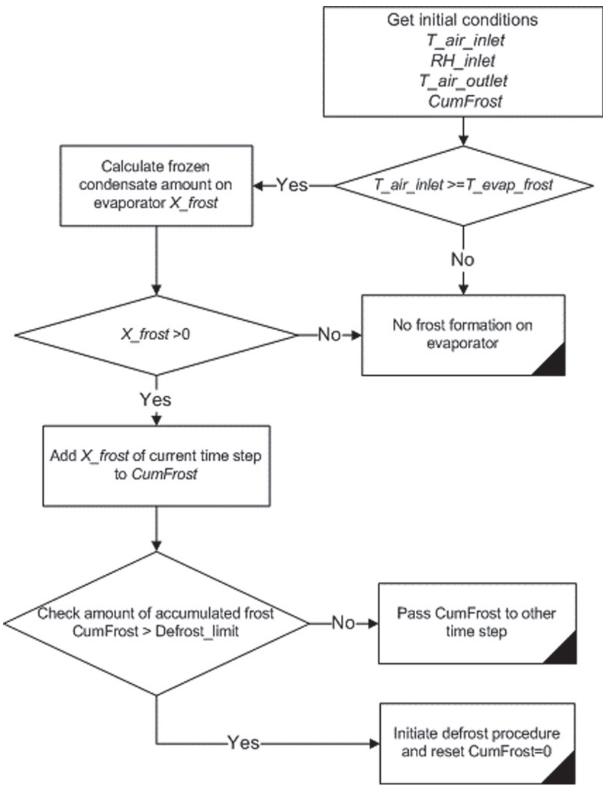


Fig. 5. ASHP frost formation and defrosting mode initiation algorythm

Calibration of model elements is important step during preparation for complex simulations routines used for system design, accurate performance prediction and parametric studies. Many authors perform model calibration in following way: (1) select mathematical models suitable for particular elements for required time scale; (2) identify tuning parameters in mathematical models; (3) compare simulation and measurement data and express the differences with error function; (4) find parameters ensuring best model fitting measured data (outputs).

When experimental data is available as input and output, setting parameters of models could be done manually in trial-and-error way or by employing optimization tools. Typically mismatch between measured and simulated data is expressed by error function. The goal of calibration procedure is to find model parameters that ensure minimum possible value of error function. At cases when multiple parameters should be fitted, there is risk to choose incorrect parameters combination while mismatch is minimal. This could be handled by applying bounding values for each parameter limited by physical means.

Most processes where capacity is neglected by averaging could be simplified to quasi-steady state approach without taking operational history into account. Seeking to improve simulation accuracy and capture dynamic in systems, higher time resolution (smaller time step) is applied.

Calibration quality is assessed via mismatch function. This function expresses cumulative difference between measured and simulated data points. In order to avoid error compensation in summation due to positive and negative differences, comparison of single data point pair is squared before summation. Uncertainty of data used as simulation (data driven) model input are assumed to be constant and this allows to neglect this parameters in error function calculation. The mismatch is expressed using basic statistical expression Weighted Root Mean Square Error (WRMSE) as a modified Root Mean Square Error (RMSE) (Reddy 2010):

$$\mathrm{WRMSE} = \sqrt{\frac{\sum_{period}\left(\left(\frac{\Delta_{\max}}{\Delta_{curr}}\right)\left(X_{measured} - X_{simulated}\right)^2\right)}{\sum_{period}\left(\frac{\Delta_{\max}}{\Delta_{curr}}\right)\cdot\tau_{period}}}, \tag{4}$$

where: $X_{measured}$ – measured value; $X_{simulated}$ – simulated value; Δ_{curr} – deviation at current time step; $\Delta\max$ – maximum deviation; τ_{period} – time period, min.

While measurement has uncertainty, this aspect should be given into account during the calibration process. During the model calibration procedures, the parameters of TRNSYS types were adjusted to have better model fitting by minimizing the RMSE and WRMSE parameters.

Results and discussion

Simulation model with technical parameters from documentation or initial parameters have quite unacceptable discrepancy from measured data. As main indicators heat pump outlet temperature and heat pump COP were chosen. To create closest possible match, such model parameters were adjusted:

− Air temperature limit for evaporator frosting: 5 ºC;

− Total cumulated frost amount till defrosting: 205 [–];

After adjustment of performance data used to simulate ASHP operation in TRNSYS type 919, a good agreement

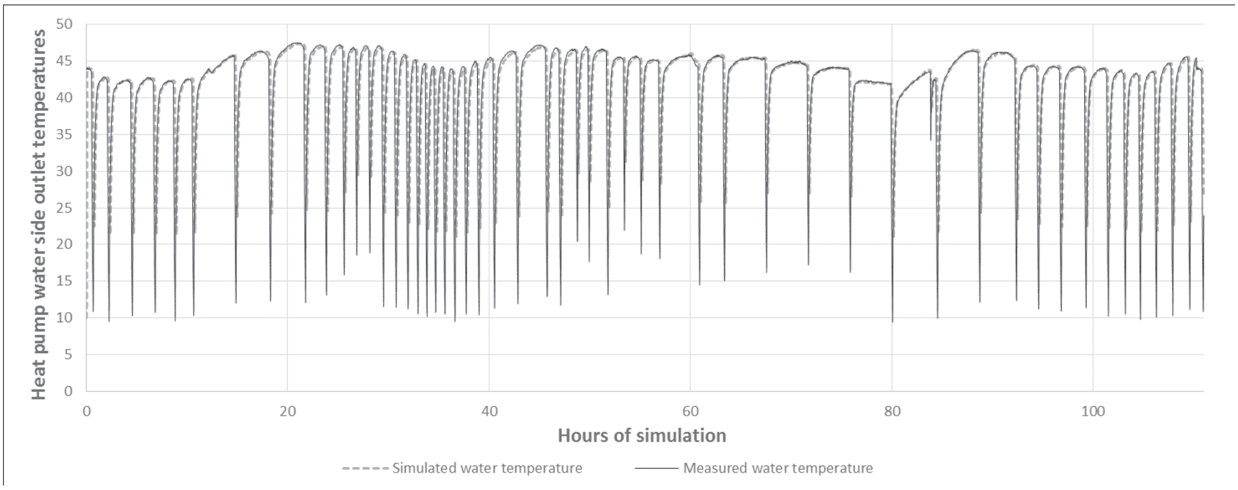


Fig. 6. Measured and simulated water temperatures after ASHP

between simulated and measured data was reached using measured inlet water temperatures as model input (see Fig. 6).

Due to frosting effects, heat pump operation could not be simulated accurately when these effects are neglected as it is done in EN 15316-4-2 standard based methods. While manufacturers gives performance data without frosting and there are no explained and standardized way of including frost formation and decrease of seasonal performance factor due to defrosting periods, we used empirical data driven frost formation model which helps to create better fit simulation model which more accurately reflects air source heat pump operation under frosting temperatures.

During the calibration procedure statistical parameter WRMSE was examined and parameters ensuring the lowest error values were selected. By combining measurement uncertainty with model fitting, overall prediction error could be expressed for absolute values.

As a conclusion could be drawn that the COP value of ASHP could be predicted with accuracy of 0.41 using the developed TRNSYS model. The temperature of the outlet from heat pump has the 0.77 °C absolute error. When typically heat pump systems are designed to operate 5 degrees temperature difference, this model could predict energy consumption with 11% accuracy for the periods when frosting on evaporator appears. For periods when there is no frost formation, ASHP performance could be predicted with greater accuracy due to closer conditions to manufacturer declared data. While CEN standards allow uncertainty up to 9% for air/water heat pumps (Ertesvag 2011), additional 3% possible deviation due to polynomial frost model seems to be acceptable when included effects bring higher deviation to final result.

Seasonal performance

To predict effects on seasonal efficiency in cold climate where heating season average temperature for typical buildings is about 0 °C, the weather file of Vilnius was utilized in the simulation. The heating season was assumed to begin on Oct. 1 and end on May 21. Heat pump system was simulated to satisfy the heating demand which has specific envelope losses of 56.4 W/K and ventilation system with 90% heat recovery creates 11.9 W/K of specific losses. Infiltration effects were taken into account via Type 932 and by assuming that leakage area is ~0.31 m^2. Solar heat gains were included in to simulation with Type 687 which takes heat transfer coefficient and solar transmittance as a parameter. Overall energy consumption for space heating demand covered with air source heat pump was calculated 59 kWh/m^2.

For the ASHP, at the end of the heating season the total electricity consumption turned out to be 1737 kWh and the total heating output was 5927 kWh, leading to a SPF of 3.41 without taking frosting into account. If those effects were taken into consideration at simulation model, the annual efficiency degrades to 2.86 or approximately 15.9 % of SPF at simulated conditions when heating system supply temperature at design conditions is 40 °C.

Conclusions

In the present paper, the experimental investigations and simulations using TRNSYS software on the operation of the ASHP were carried out. The following conclusions are obtained:

1. It is very important to take into account frost formation and defrosting procedure when simulating annual performance of air source heat pumps. In climate condition like or similar to Lithuania this effect may cause 15.9%

deviation from prediction of models neglecting frost formation influence to performance. Increased accuracy of simulation predictions may help to assess and compare energy source alternatives more objectively.
2. Simulation model was calibrated according to experimental measurement which has COP uncertainty of from 8.5 to 9.5%. After calibration procedures, model can predict outlet temperatures with ±0.77K accuracy.
3. Suggested model takes into consideration more environmental parameters than many other models implemented in TRNSYS simulation software. That makes possible to predict performance of the ASHP system for climate specific conditions more accurately – without 15.9% under prediction of consumed electricity to produce same amount of heat.
4. It is possible to use this air source heat pump model with manufacturer data, which are suitable for each location worldwide and add local climate specific humidity and temperature combination influenced effects.

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ŠILUMOS SIURBLIO „ORAS – VANDUO“, VEIKIANČIO ŠALTAME KLIMATE, EKSPERIMENTINIŲ IR MODELIAVIMO REZULTATŲ PALYGINAMOJI ANALIZĖ

K. Januševičius, G. Streckienė

Santrauka

Straipsnyje pateikiami orinio šilumos siurblio sistemos tyrimo rezultatai: sistemos monitoringo duomenų ir TRNSYS aplinkoje sudaryto modelio rezultatų palyginamoji analizė. Imitacinis modelis buvo kalibruotas pagal eksperimentinius duomenis. Išorės oro temperatūrų intervale nuo –7.0 ºC iki +5.0 ºC šilumos siurblio efektyvumas (COP) svyravo tarp 1,98 ir 3,05. Imitaciniame modelyje, atsižvelgiant į užšalimo įtaką, šis efektas sezoninį efektyvumą sumažina 15 %, lyginant su atveju, kai garintuvo užšalimo ir atitirpinimo efektai nėra įvertinami.

Reikšminiai žodžiai: orinis šilumos siurblys, efektyvumo koeficientas, eksperimentinis tyrimas, imitacinis modeliavimas, sezoninio efektyvumo faktorius, TRNSYS.