EVALUATION OF THERMAL PERFORMANCE OF AIR SOURCE HEAT PUMP HEATING SYSTEM BASED ON ELECTRICITY EQUIVALENT

by

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Thermal performance assessment and optimization for energy conversion and utilization systems are of high significance in building energy efficiency. Generally speaking, the evaluation of actual thermodynamic system performance is mainly based on the First law of thermodynamics, with emphasis on the quantity of energy consumption, while ignoring the energy quality. Thus, it results in a one-sided evaluation in the analysis of system energy saving. In this paper, the electricity equivalent is used to analyse and evaluate the air source heat pump heating system under the different working conditions. Moreover, the dynamic thermal performances of three typical space heating devices (radiator, fan coil, and radiant floor) are investigated and compared by weighing both energy quantity and quality. The results show that the COP of radiator, fan coil, and radiant floor are 3.00, 3.82, and 4.76, and coefficient of energy quality are 48.32%, 51.43%, and 50.57%, respectively. However, the robustness of its thermal performance is lower than that of radiator and fan coil. This work can provide reference for energy systems assessment and guidance for practical design of building heating systems.

Key words*: electricity equivalent, coefficient of energy quality, energy quality, air source heat pump, energy-saving evaluation*

Introduction

In 2018, buildings consumed 1 billion tons of standard coal in China, accounting for 22% of the total national energy consumption. Therein, energy consumption in public civil buildings is 520 million tons of standard coal, urban housing, rural housing, and public buildings accounted for 42%, 14%, and 44%, respectively [1]. The carbon emissions associated with fossil energy consumption from building operations are 2.1 billion tons of CO_2, of which 26% are from heating in northern China [1]. Research shows that developing appropriate energy supply and heating technology to reduce the energy consumption of heating in winter has become an urgent problem to be solved in northern China [2].

Since Watt, Carnot, Clausius, and others laid the foundation of the Second law of thermodynamics, scientists and researchers have never abandoned their efforts in thermody-

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namic cycles [3]. Zhang *et al*. [4, 5] propose a new concept, the heat adaptor, and obtained the best process and best device arrangement using entropy generation analysis. Sun *et al*. [6] analysed double-pressure organic Rankine cycle driven by low temperature heat source. It is found that the exergy efficiency has a peak value with the change of turbine high-level inlet pressure and turbine low-level inlet pressure. Karapekmez *et al*. [7] develop a new solar and geothermal based integrated system and defined the energetic and exergetic efficiencies and exergy destruction rates for the whole system and its parts. The results show that the highest overall energy and exergy efficiencies are calculated to be 78.37% and 58.40% in the storing period, respectively.

On the other hand, compared to Carnot cycle, the Carnot inverse cycle, also plays an important role in energy saving. Ding *et al*. [8] tested the heat pump/heat pipe composite system and the experimental results showed that the system can operate efficiently and steadily when the outdoor temperature is –20~5 °C, and meet the winter heating demand in cold areas. Wang *et al*. [9] analysed regenerative evaporative cooler and obtained better thermal performance based on the First and Second law of thermodynamics. Vučković *et al*. [10] analysis of individual components' exergetic efficiency and the overall system's exergetic efficiency reveals that the selection of the heat pump should be adjusted to end users, reducing the impact of the compressor's exergy destruction coefficient.

Nonetheless, clean heating in northern cities faces several big challenges [11]. Improving energy efficiency of terminal heating equipment is one of the key approaches to control total heating energy consumption. In recent years, as a clean energy technology, air source heat pump (ASHP) heating system has been widely used in the *coal to electricity* heating system renovation project in northern China [12, 13]. Estimating and predicting energy demand and consumption may contribute to improve energy performance and saving [14]. Hence, it is necessary to analyse the energy saving potential and efficiency of ASHP before and after they are applied in the actual buildings.

At present, most of the research direction lies in the optimization of heating links [15]. The thermal performance evaluation of heat pump mainly concentrates on its COP, which is regarded as energy-saving heating equipment because its COP could be higher than 1 [16-18]. However, COP-based evaluation of heat pump only considers the efficiency of energy quantity, regardless of the difference of energy quality. Because heat pump consumes high-quality electrical energy to obtain low-quality heat energy [19]. Electricity equivalent was proposed by integrating energy quantity and quality, which can be evaluated more comprehensively by combining the efficiency of the first law of thermodynamics [20].

In this paper, based on the electricity equivalent, the dynamic thermal performance of building heating systems is investigated in more comprehensive regards by considering system energy utilization efficiency of both energy quantity and quality. In addition, the coefficients of electricity equivalent are obtained and compared for three typical different heating equipment (radiator, fan coil, radiant floor). Moreover, the sensitivity analysis is conducted under different working conditions with key design temperatures variations. This work can provide guidance for practical design of building heating systems.

Instructions

The ASHP is a device that uses high quality energy (electricity) to transfer heat from cold reservoir (ambient air source) to heat reservoir (terminal equipment) [3]. The ASHP heating gradually replaced coal-fired heating in northern China under the promotion of electric heating reform project. Increased demand for heating in the south has further boosted the use of ASHP heating system.

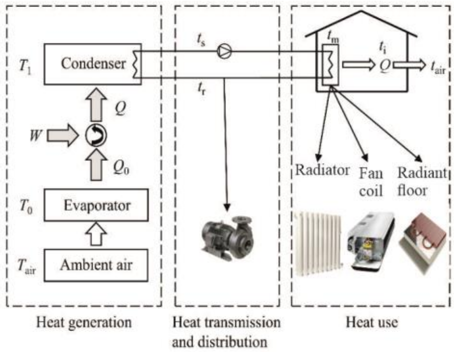

Figure 1. Schematic diagram of ASHP heating system

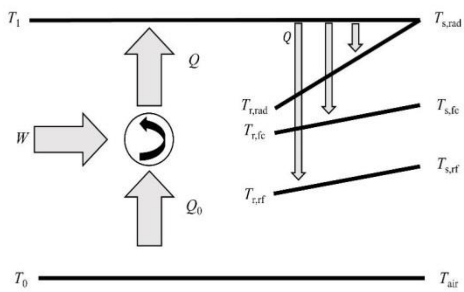

Figure 2. Schematic diagram of temperature of ASHP heating system

Figures 1 and 2 show ASHP heating system consists of three links and three terminal devices. On the left of the figure is the heat generation link, which containing cold reservoir, heat reservoir, heat pump and heat exchanger. The heat is transferred from the ambient air through the evaporator to the circulating working medium, which becomes high temperature and high pressure gas through the compressor. The heat is transferred to the return water in the condenser, and the condensing circulating working medium is returned to the evaporator after being throttled by the throttle valve, at this point a cycle completes. In this cycle, the work of the compressor is W, the heat released by the condenser is Q_0, and the heat absorbed by the condenser is Q. In the middle of the figure is the heat transmission and distribution link, which containing water pump and pipe network. The return water absorbs heat through the condenser and become supply water. The water temperature t_r rises to t_s. Then supply water is transited and distributed to the end users through the pump. The water temperature t_s reduces to t_r in the ends and become return water. On the right of the figure is the heat use link, which containing the user and the terminal device. The supply water transfers heat to the indoor environment through the terminal device to maintain indoor thermal comfort. Then the heat is transferred from the indoor to the outdoor environment. At present, the terminal device mainly adopt radiator, fan coil and radiant floor. There are differences in the heat transfer mode of the three ends. Radiator mainly transfers by heat radiation and heat convection to the indoor environment, fan coil mainly transfers by heat convection, and radiant floor mainly transfers by heat radiation. There is also a difference in the temperature of supply and return water, from high to low, respectively, for the radiator, fan coil, and radiant floor. Where Q_0 [kW] and Q [kW] represent heat pump absorb heat from ambient air source and release heat to heat reservoir, respectively, W [kW] means energy consumption of heat pump, t_s, t_r, t_m, t_i, and t_{air} [°C] mean supply water temperature, return water temperature, average temperature of terminal device, indoor air temperature, and ambient air temperature, respectively, T_{air}, T_0, and T_1 [K] mean ambient air temperature, evaporating temperature, and condensing temperature, respectively, $T_{s,rad}$, $T_{s,fc}$, and $T_{s,rf}$ [K] mean supply water temperature of radiator, fan coil, and radiant floor, respectively, and $T_{r,rad}$, $T_{r,fc}$, and $T_{r,rf}$ [K] are return water temperature of radiator, fan coil, and radiant floor, respectively.

Figures 1 and 2 show the following mathematical relationships.

– Heat generation:

$$Q = Q_0 + W \tag{1}$$

$$T_{\mathrm{air}} \geq T_0 \tag{2}$$

$$T_1 \geq T_{\mathrm{s}} \tag{3}$$

– Heat transmission and distribution:

$$Q = \rho c\left(t_{\mathrm{s}} - t_{\mathrm{r}}\right) \tag{4}$$

– Heat use:

$$Q = \rho c\left(t_{\mathrm{m}} - t_{\mathrm{i}}\right) \tag{5}$$

where ρ [kgs^{-1}] and c [Jkg^{-1}°C^{-1}] are mass flow rate and specific heat, respectively.

Coefficient of performance, coefficient of energy quality, coefficient of electricity equivalent, and their sensitivity

Coefficient of performance

$$\varepsilon = \mu \frac{T_1}{T_1 - T_0} \tag{6}$$

where μ, T_0, and T_1 are the thermodynamic perfect degree, evaporating temperature, and condensing temperature, respectively.

Thermodynamic perfect degree represents the close degree between the heating cycle and the reverse Carnot cycle under the same temperature limit and depends on the process level of the heat pump manufacturing. In order to know the influence degree of T_0 and T_1.

Coefficient of energy quality

The Second law of thermodynamics is a law that clarifies the direction, conditions, and limits of various processes related to thermal phenomena. It reveals the grade nature of energy and describes the law of conversion between work and heat and the irreversibility of the system. From the perspective of heat and power conversion, electric energy is regarded as the highest quality energy. It is more reasonable to convert all forms of energy into electric energy in a unified way, so as to make statistics, analysis and evaluation more convenient for combining energy quantity and power capacity. Electricity equivalent is to convert all forms of energy into equivalent electricity according to the maximum possible conversion capacity of various forms of energy, and then according to the electricity to statistics, accounting for the amount of energy. Therefore, any form of energy Q can be converted to equivalent electrical Q_{e} in a given environment:

$$Q_{\mathrm{e}} = \lambda Q \tag{7}$$

where Q [W] and Q_{e} [W] mean any form of energy and equivalent electricity energy, respectively, and λ mean coefficient of energy quality (COEQ).

Electric energy is the highest quality energy, which can be completely converted into work. Its COEQ is 1 and other forms of energy's less than 1. According to the Second law of thermodynamics, the relationship among COEQ and water supply temperature, T_{s}, water return temperature, T_{r}, and the evaporating temperature, T_0:

$$\delta Q_{\mathrm{e}} = \left(1 - \frac{T_0}{T}\right)\delta Q = \left(1 - \frac{T_0}{T}\right) mc\mathrm{d}T \tag{8}$$

$$Q_{\mathrm{e}} = \int_{T_{\mathrm{r}}}^{T_{\mathrm{s}}} \left(1 - \frac{T_0}{T}\right) mc\mathrm{d}T = \left(1 - \frac{T_0}{T_{\mathrm{s}} - T_{\mathrm{r}}} \ln \frac{T_{\mathrm{s}}}{T_{\mathrm{r}}}\right) Q \tag{9}$$

$$\lambda = 1 - \frac{T_0}{T_{\mathrm{s}} - T_{\mathrm{r}}} \ln \frac{T_{\mathrm{s}}}{T_{\mathrm{r}}} \tag{10}$$

Table 1 shows COEQ of some common primary and secondary energy sources.

Table 1. The COEQ of common primary and secondary energy sources

Types of energy	Reference temperature, T_1	Ambient air temperature, T_{air}	COEQ	Quantity of heat, Q	Equivalent electricity, Q_{e}
Electric power	–	–	100%	1.000 kWh	1.000 kWh/ kWh
Natural gas	1500	0	65.9%	10.825 kWh	7.133 kWh/m^3
Crude oil	1500	0	65.9%	11.628 kWh	7.663 kWh/kg
Gasoline	1500	0	65.9%	11.977 kWh	7.893 kWh/kg
Diesel oil	1500	0	65.9%	11.860 kWh	7.816 kWh/kg
Standard coal	700	0	50.4%	8.140 kWh	4.103 kWh/kg
Hot water (95℃/70℃)	–	0	23.2%	1.000 kWh	0.232 kWh/kWh
Hot water (50℃/40℃)	–	0	14.1%	1.000 kWh	0.141 kWh/kWh
Saturated vapor (0.4 MPa)	–	0	34.5%	1.000 kWh	0.345 kWh/kWh
Saturated vapor (0.3 MPa)	–	0	32.9%	1.000 kWh	0.329 kWh/kWh
Air-conditioned chilled water (7℃/12℃)	–	30	7.3%	1.000 kWh	0.073 kWh/kWh

Coefficient of electricity equivalent

$$\eta = \varepsilon\lambda \tag{11}$$

Sensitivity

$$\varepsilon_{\mathrm{sen}} = \frac{\varepsilon_1 - \varepsilon_0}{\varepsilon_0} \tag{12}$$

$$\lambda_{\mathrm{sen}} = \frac{\lambda_1 - \lambda_0}{\lambda_0} \tag{13}$$

$$\eta_{\mathrm{sen}} = \frac{\eta_1 - \eta_0}{\eta_0} \tag{14}$$

where $\varepsilon_{\mathrm{sen}}$, $\lambda_{\mathrm{sen},}$ and η_{sen} are the sensitivity of the COP, COEQ, and coefficient of electricity equivalent (COEE), respectively, ε_1, λ_1, and η_1 are the COP, COEQ, and COEE after change, respectively, and ε_0, λ_0, and η_0 are the COP, COEQ, and COEE before change, respectively.

The follow section gives an illustrate example and sensitivity analysis.

Illustrate example

known conditions

In eq. (1), COP mainly influenced by evaporating temperature, T_0, and condensing temperature, T_1. In eq. (3) COEQ influenced by water supply temperature, T_{s}, water return temperature, T_r, and the evaporating temperature, T_0. In eq. (4), COEE influenced by these four temperatures. In the ideal countercurrent heat exchanger system, the condensing temperature cannot be lower than the water supply temperature, $T_1 \geq T_s$, and the ambient air temperature cannot be lower than the evaporating temperature, $T_{air} \geq T_0$. Under the same load rate, the thermal perfection of the heat pump unit is basically unchanged. In order to simplify analysis, assuming that:

- $T_1 = T_s$.
- $T_{air} = T_0$.
- The rated working condition of the ASHP is T_{air} = 273.15 K.
- The COP of ASHP with the radiator as the terminal is 3.
- The thermodynamic perfect degree of ASHP is constant when the ambient air temperature, T_{air}, and the water supply temperature, T_s, change.
- The rated thermodynamic perfect degree of ASHP is a constant value, $\mu \equiv 0.54$.

The rated water supply and return temperatures of different terminal ends are shown in tab. 2.

Table 2. Rated water supply and return temperatures at different ends

Ends	t_s / t_r
Radiator	60°C/45°C
Fan coil	45°C/40°C
Radiant floor	35°C/30°C

Result of ideal condition

Based on the previous constraints, the influences of T_s, T_r, and T_{air} on COP, COEQ, and COEE are respectively calculated. The results are shown in figs. 3-5.

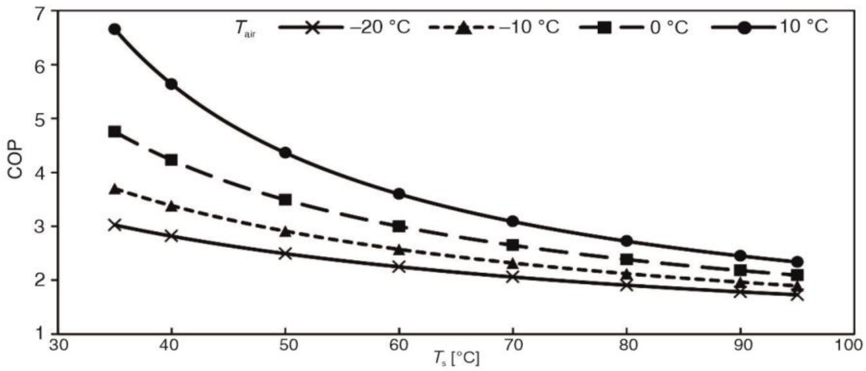

Figure 3. The effect of COP on T_s and T_{air}

It can be seen that the COP decreases when T_s increases, and at the same T_s, COP increases with increasing T_{air}, fig. 3. The situation in figs. 4 and 5 is more complex compared to fig. 3. As fig. 4 shows, at the same T_{air} and T_r, with the increasing T_s, the COEQ increases. At the same T_s and T_r with the increasing T_{air}, the COEQ decreases, and at the same T_s and T_{air}, with the increasing T_r, the COEQ increases. In fig. 5, the COEE decreases when T_s increases. At the same T_s and T_r with the increasing T_{air}, the COEQ decreases, and at the same T_s and T_{air} with the increasing T_r, the COEQ increases.

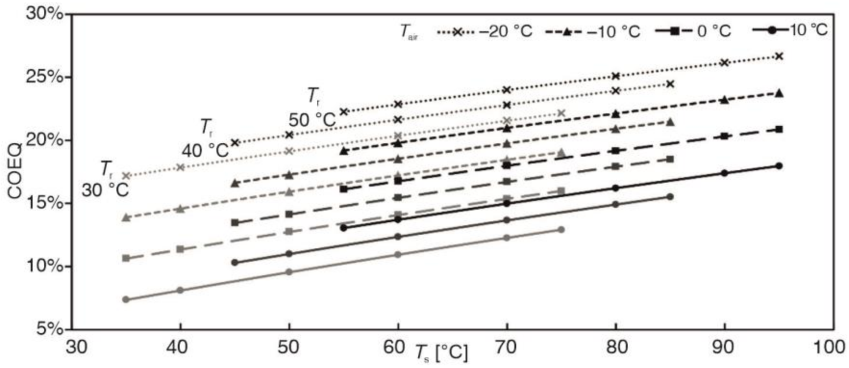

Figure 4. The effect of COEQ on T_s, T_r, and T_{air}

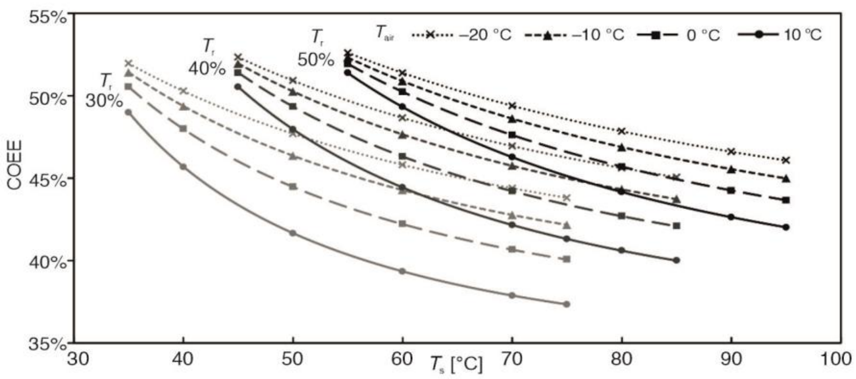

Figure 5. The effect of COEE on T_s, T_r, and T_{air}

In other word, in all three figures it can be seen that each individual curve is monotonic. However, fig. 5 differs from figs. 3 and 4 in that figs. 3 and 4 have no intersecting curves, while fig. 5 has a considerable number of curves that are intersecting. From this we can learn that evaluating ASHP from the First law of thermodynamics is oversimplified, while evaluating from the Second law of thermodynamics is relatively more complex. Therefore, previous studies show that ASHP should use the end of the heat source at low temperature, which is not entirely correct.

Result and discussion of actual condition

Based on the aforementioned assumption and the known conditions, increased T_s and decreased T_{air} lead to decreased COP, increased COEQ, and increased COEE. But for the actual equipments, different systems have different changes in COP, COEQ, and COEE. The tab. 3 shows the effects of different systems on COP, COEQ, and COEE when the T_{air} is constant and the T_s increases by 10 ℃. The tab. 4 shows the effects of different systems on COP, COEQ, and COEE when the T_s is constant and the T_{air} decreases by 10 ℃.

Table 3 shows that the rated COP of radiator, fan coil and are radiant floor are 3.00, 3.82, and 4.76, respectively. The variations are –0.35, –0.60, and –0.94, respectively. The sensitivities are –11.71%, –15.61%, and –19.70%, respectively, the rated COEQ of radiator, fan coil, and are radiant floor are 16.11%, 13.46%, and 10.63%, respectively. The variations are 3.07%, 3.17%, and 3.27%, respectively. The sensitivities are 7.72%, 9.92%, and 13.39%, respectively, the rated COEQ of radiator, fan coil, and are radiant floor are 48.32%, 51.43%, and 50.57%, respectively. The variations are –2.37%, –3.72%, and –4.52%, respectively. The sensitivities are –4.89%, –7.24%, and –8.95%, respectively.

Table 3. Ambient air temperature is constant and water supply temperature increases by 10 °C

t_s	Ends	COP	COEQ	COEE
Initial value	Radiator	3.00	16.11%	48.32%
	Fan coil	3.82	13.46%	51.43%
	Radiant floor	4.76	10.63%	50.57%
Variation	Radiator	–0.35	1.24%	–2.37%
	Fan coil	–0.60	1.34%	–3.72%
	Radiant floor	–0.94	1.42%	–4.52%
Sensitivity	Radiator	–11.71%	7.72%	–4.89%
	Fan coil	–15.61%	9.92%	–7.24%
	Radiant floor	–19.70%	13.39%	–8.95%

Table 4. Water supply temperature is constant and ambient air temperature decreases by 10 °C

t_r	Ends	COP	COEQ	COEE
Initial value	Radiator	3.00	16.11%	48.32%
	Fan coil	3.82	13.46%	51.43%
	Radiant floor	4.76	10.63%	50.57%
Variation	Radiator	–0.43	3.07%	0.99%
	Fan coil	–0.69	3.17%	0.55%
	Radiant floor	–1.06	3.27%	0.87%
Sensitivity	Radiator	–14.29%	19.07%	2.06%
	Fan coil	–18.18%	23.53%	1.07%
	Radiant floor	–22.22%	30.78%	1.71%

The rated COP, COEQ, and COEE are the same as tab. 3. Table 4 shows that the variations of COP are –0.43, –0.69, and –1.06 and the sensitivities are –14.29%, –18.18%, and –22.22%, respectively, the rated COEQ are 16.11%, 13.46%, and 10.63%, respectively. The variations of COEQ are 3.07%, 3.17%, and 3.27% and the sensitivities are 7.72%, 9.92%, and 13.39%, respectively. The variations of COEE are 0.99%, 0.55%, and 0.87% and the sensitivities are 2.06%, 1.07%, and 1.71%, respectively.

In all, considering the First law of thermodynamics alone, the results show that the effect of T_s on COP is greater than T_{air}, and radiant floor is the best end. But considering the Second law of thermodynamics alone, the result is the opposite, fan coil is the best one. On the other hand, considering the sensitivity alone, radiator is the best one. Hence, when the T_{air} and T_s are stable, it is better to choose radiant floor as the end, otherwise choose radiators and fan coils.

Conclusions

The COP-based evaluation for ASHP and other building energy systems are widely used with emphasis on energy quantity from the First law of thermodynamics. However, it is one-sided that cannot depict the energy quality difference. Thus, in previous studies, equivalent electricity was defined by conversing different energy states to corresponding electricity, which is viewed as the energy typed of highest quality from the second law of thermodynamics.

In this paper, combined with energy quantity and quality. The COP, COEQ, and COEE of ASHP are calculated at ideal condition. In addition, the thermal performance is obtained and compared for three typical different heating equipments (radiator, fan coil, and radiant floor). Moreover, the sensitivity analysis is conducted under different conditions with design temperatures variations. It can be concluded that:

- In the same conditions, the COP and COEE of ASHP decreases with the increase of T_s and the decrease of T_{air}. The COEQ of ASHP increases with the increase of T_s and decreases with the decrease of T_{air}. Evaluating ASHP on COP is more simplified than that on COEQ.
- The results of an illustrative example show that the rated COP of radiator, fan coil, and radiant floor are 3.00, 3.82, and 4.76, rated COEQ are 16.11%, 13.46%, and 10.63%, and COEQ are 48.32%, 51.43%, and 50.57%, respectively. Under the same condition, the evaluation on COP is radiant floor > fan coil > radiator. The result on COEQ is the opposite. The evaluation on COEE is fan coil > radiant floor > radiator.
- The example also shows that the evaluation on robustness is radiator > fan coil > radiant floor.

All in all, in terms of considering energy quantity and quality evaluation, adopting radiant floor as the terminal equipment of ASHP heating system is a better choice with the highest comprehensive energy efficiency. On the other hand, the robustness of its thermal performance is lower than that of radiator and fan coil system, since it shows high sensitivity in temperatures, which imposes high operation demand on the system stability and reliability. In this paper, only simplified cases are considered with ideal assumptions. For example, the relationship between T_{air}, T_s, and μ has not considered, which presents new challenges for further detailed analysis. The present work shows the preliminary application of the COP, COEQ, and COEE to pre-estimate and compare thermal performance of ASHP heating systems. Although the specific results obtained for the studied case may not be applicable to all situations, the analysis approach used here is general. This work can provide an energy system assessment reference to guide the practical work on building heating systems.

Nomenclature

Q – heat energy capacity [W]
T – temperature [K]
t – temperature [°C]
W – work [W]

Greek symbols

ε – coefficient of performance
λ – coefficient of energy quality
η – coefficient of electricity equivalent
ρ – mass flow rate [kgs^{-1}]

Acronyms

ASHP – air source heat pump
COEQ – coefficient of energy quality
COEE – coefficient of electricity equivalent

Subscripts

air – ambient air
e – equivalent electricity
fc – fan coil
i – indoor
m – average
r – return
rad – radiator
rf – radiant floor
s – supply
sen – sensitivity
0 – evaporator
1 – condenser

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