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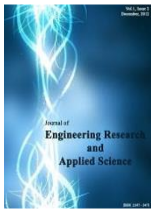


# Analysis of Optimum Thermal Performance of Air Source Heat Pump for Building Heating

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## Abstract

Driven by Turkey's dual-carbon goals and the national strategy of clean heating, the application of air source heat pump heating is growing rapidly. This paper analyzes the influence of the equilibrium point temperature selection of low ambient temperature and conventional air-source heat pump heating products in different regions, different energy-saving grades and different building types on the system performance. The results show that there is an obvious difference in the equilibrium point temperature corresponding to the optimal performance of air source heat pump heating in different regions. The equilibrium point temperature of different building types in the same region is different. In the same application situation, the equilibrium point of the low ambient temperature heat pump is generally lower than or equal to the conventional heat pump, the coefficient of performance of the heating season is higher, and has better application performance.

**Keywords:** *Air source heat pump; Building heating; Heating season performance factor*

## 1. Introduction

In order to improve the atmospheric environment in northern Turkey, the government has put forward a clean heating strategy, combined with the dual-carbon goals of "carbon peaking and carbon neutrality", and adopted electrified terminals to transform the seriously polluted bulk coal heating mode [1-3]. Air source heat pump (ASHP) has the advantages of easy availability of low heat source and flexible installation [4-8]. Driven by demand, it continues to improve the applicability of low temperature and has become the dominant way of "electricity instead of coal". The current market scale is about to 7 billion Turkish Liras (TL).

For the ASHP, the heating capacity and performance are positively correlated with the ambient temperature, that is, the higher the ambient temperature, the greater the heating capacity of the air source heat pump, the better the performance; for the user side, the indoor heat load is negatively correlated with the ambient temperature [8]. When the indoor design temperature is the same, the higher the ambient temperature is, the lower the indoor heat load is, and the lower the outdoor temperature is, the higher the indoor heat load is [4-10].

The heating capacity of air source heat pump and building heat load have the opposite trend with the ambient temperature. When they are equal, the corresponding ambient temperature is the equilibrium point temperature [5]. When the outdoor temperature is higher than the equilibrium point temperature, the heat pump unit runs under partial load, resulting in a decrease in the coefficient of performance; when the outdoor temperature is lower than the equilibrium point temperature, it is necessary for the auxiliary heat source to bear part of the heat load, which can't give full play to the energy-saving role of the air source heat pump. Therefore, a reasonable equilibrium point temperature can improve the coefficient of performance of the unit and give full play to the advantages of air source heat pump [5].

There have been many researches on the equilibrium point temperature of air source heat pump at home and abroad. Jiang's team [1-5] took the lead in putting forward the concepts of the optimal energy efficiency equilibrium point and the optimal energy equilibrium point, and took specific cities as examples to carry out relevant calculations. The research direction abroad is mostly focused on the method of determining the capacity of auxiliary heat source, including the difference of equilibrium point temperature when using different heating methods such as electric heating or gas heating as auxiliary heat source, which is also based on the temperature theory of heat pump equilibrium point [5].

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The above research uses more steady-state calculation methods in the calculation of indoor heat load. With the development of computer technology, much simulation software has been developed to simulate indoor heating and cooling load, such as TRNSYS and so on. TRNSYS has the advantages of convenient use and flexible calculation, so this study uses TRNSYS software to simulate the cooling and heating load of different energy-saving buildings in different regions, and then uses the simulated indoor hourly heat load to calculate the optimal equilibrium point temperature of low ambient temperature air source heat pump (LATASHP) and conventional air source heat pump (CASHP) [5].

## 2. Calculation Methods

The determination of the equilibrium point temperature depends on the choice of the final optimization objective [8-10]. At present, there are two kinds of optimization objectives: the economic goal and the system performance goal. Using the annual cost value or the whole life cycle cost as the objective function to find the best economic equilibrium point temperature, its purpose is to ensure that the sum of the initial investment and operation cost of the heating system is the lowest [1]. Using the seasonal coefficient of performance of the system heating HSPF or the seasonal coefficient of performance of the air source heat pump unit SCOP as the objective function to find the best performance equilibrium point temperature, the goal is to optimize the overall heating performance. The primary energy utilization coefficient of the air source heat pump and the primary energy utilization coefficient of the boiler are also used as the objective function, that is, when the heating energy utilization coefficient of the heat pump is equal to that of the auxiliary boiler, the primary energy efficiency of the heating system is the highest [5]. This paper mainly studies the balance point whose ultimate optimization goal is the system performance goal. Figure 1. Schematic of a dual source heat pump system (air + solar) and Figure 2 also shows information flow diagram on TRNSYS [8, 9].

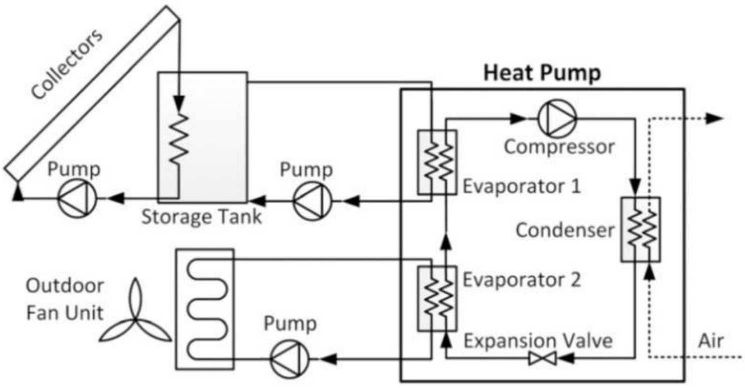


Figure 1. Schematic of a dual source heat pump system (air + solar) [6].

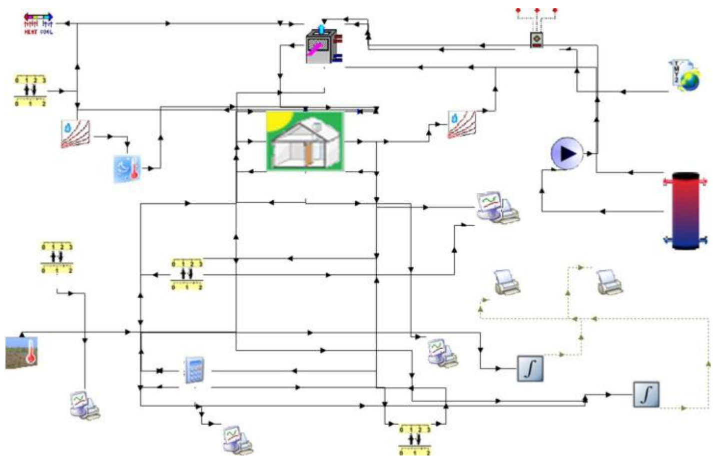

Figure 2. Information flow diagram on TRNSYS [9].

**2.1. Optimal Performance Equilibrium Point Temperature of Heating System**

In the formula [5]: $Q_{1i}$ is the indoor heat load when the outdoor air temperature is I, kW; $W_i$ is the power consumption of the heat pump when the temperature is I, kW; $Q_{fi}$ is the heat consumption of the auxiliary heat source when the temperature is I, kW; $T_l$ is outdoor calculation temperature for design conditions for use region, ℃; $T_{min}$ is the higher value of the minimum outdoor temperature that can be operated for the heat pump unit and the outdoor calculated temperature under the design condition, ℃; $T_{max}$ is the highest outdoor temperature for heating operation of a heat pump unit, ℃; $T_b$ is the preset equilibrium temperature of the heating system, ℃ ; $Q_{hi}$ and $Q_{hj}$ are the heat generated by the heat pump unit at a certain outdoor temperature during the heating period, kW ; $COP_h$ is the coefficient of heating performance of heat pump unit ; $PLF$ is the COP correction factor of partial load heat pump unit ; $P_f$ is the energy consumption conversion coefficient of auxiliary electric heating equipment, and the electric heating equipment generally takes 1;

$$\mathrm{HSPF} = \frac{\sum_{i:T1}^{i:Tmax} Q1i}{\sum_{i:Tmin}^{i:Tmax} Wi + \sum_{i:T1}^{i:Tb} PfQfi} = \frac{\sum_{i:T1}^{i:Tmax} Q1i}{\sum_{i:Tmin}^{i:Tb}\left(\frac{Qhi}{COPh}\right) + \sum_{i:Tb}^{i:Tmax}\left(\frac{Qhj}{COPh}\cdot\frac{1}{PLF}\right) + \sum_{i:T1}^{i:Tb} PfQfi} \tag{1}$$

$$PLF = \frac{PLR}{[1 - C_d\,(1\text{-}PLR)]} \tag{2}$$

In the formula [5]; PLR is the partial load rate, which is defined as the ratio of actual heating capacity to the rated capacity of the equipment; and $C_d$ is the attenuation coefficient of heating performance of heat pump under partial load, which is generally 0.9;

$$Q_{h0} = [(Q'0/K_1K_2)] \tag{3}$$

In the formula [5]; Q′0 is the heat output of the heat pump unit at the preset equilibrium point temperature, kW; $K_1$ is the correction coefficient of heat pump unit at preset equilibrium point temperature; $K_2$ is the defrosting loss coefficient of heat pump unit at preset equilibrium point temperature [5];

$$Q_h = K_1(T)K_2(T)Q_{h0} \tag{4}$$

In the formula [5]; $Q_h$ is the heat produced by the heat pump unit at a certain outdoor temperature during the heating period; $K_1(T)$ is the ambient temperature correction coefficient of a heat pump unit during the heating period; $K_2(T)$ is the defrosting loss coefficient of the heat pump unit at a certain ambient temperature during the heating period;

$$Q_{fi} = Q_{1i} - Q_{hi} \tag{5}$$

When the heat load of the building, the air source heat pump unit and the auxiliary heat source are determined, HSPF is only a function of the equilibrium point temperature, so by changing the preset equilibrium point temperature, the curve of HSPF change can be obtained, and the equilibrium point temperature corresponding to the maximum value of HSPF is the optimal performance equilibrium point temperature.

**2.2. Load Simulation**

Building energy efficiency in Turkey is based on the building energy consumption in 200-2001. It is a stage to improve energy efficiency by 40% on the basis of the previous stage in each step, that is, to save energy by 40%, 60%, 75% and 85% respectively compared with the benchmark building. On the basis of 75% energy saving rate, buildings that can save 75-85% more energy are near zero energy consumption buildings. Using TRNSYS to simulate the load of different types and different energy-saving buildings, the relevant parameters are based on the building energy efficiency standard system, and the residential building area is 130 $m^2$, the office building area is 1000 $m^2$, and the commercial building area is 5000 $m^2$.

The higher the energy saving level is, the lower the indoor heat load is at the same ambient temperature. Because of the interference of different factors, the relationship between indoor heat load and ambient temperature is not completely linear, and the fitted linear formula can be used to replace it in practical application. However, hourly heat load is used to calculate the energy consumption of air source heat pump in this study. For near-zero energy consumption buildings, the indoor heat load is basically less than 50 W/m$^2$, and the performance of the building envelope is excellent, so the ambient temperature is only one of the factors affecting the indoor heat load, and the internal heat source also has a great influence on the indoor heat load, so the linear characteristic between the indoor heat load and the outdoor air temperature is not obvious.

### 2.3. Performance of Air Source Heat Pump

Through market investigation, the heating products of air source heat pump in the market are classified, and the representative ordinary air source heat pump products and jet enthalpy quasi-two-stage compressed LATASHP products are selected for performance analysis. The heating system is fixed as air source heat pump + fan coil end, and the water supply temperature is 45 ℃. The heating capacity and performance of the product are fitted by regression. As shown in Table 1 and 2, according to the heating capacity of the two products and the regression fitting formula of COP, we can know the change of the heat production of the air source heat pump with the outdoor temperature.

Table 1. Fitting Formula of Heat production and COP of Air Source Heat pump

| LATASHP | Fatting formula | $R^2$ |
|---|---|---|
| Heating capacity | $Y = 4.9756x^2 + 398.96x + 15246$ | $R^2 = 0.9669$ |
| COP | $Y = 0.0008x^2 + 0.0668x + 2.8682$ | $R^2 = 0.9689$ |

Table 2. Fitting Formula of Heat production and COP of Air Source Heat pump

| CASHP | Fatting formula | $R^2$ |
|---|---|---|
| Heating capacity | $Y = 3.6756x^2 + 428.96x + 13246$ | $R^2 = 0.9769$ |
| COP | $Y = 0.0004x^2 + 0.0568x + 2.7682$ | $R^2 = 0.9659$ |

## 3. Results and Discussion

The calculation results are analyzed from four directions: different climatic regions, different building types, different energy saving grades and different types of heat pumps. Select typical cities in different climate zones: Trabzon, Erzurum, Ankara. Trabzon is located in a mild climate region, Erzurum is located in a very cold region, and Ankara is located in a hot summer and cold winter region. The buildings of the three are residential buildings, and the heat pump type is LATASHP. The current building energy efficiency in Trabzon and Erzurum is 75%. The current building energy efficiency rating is 50% [10-20].

In addition, we can also see that the HSPF of the heat pump is greatly affected by the ambient temperature, and the HSPF of the typical cities in the three climate zones has obvious stratification. Region with hot summer and cold winter has great advantages in using heat pumps. The HSPF at the optimal performance equilibrium point in severe cold region is only 2.2, while that in region with hot summer and cold winters can reach 3.2. Therefore, improving the performance of heat pump at low temperature is one of the key research directions in the future [17-20].

Select different types of buildings in Beijing: residential buildings, office buildings, commercial buildings. The type of heat pump is LATASHP. The building energy saving rate is 65%. Different types of buildings mainly affect the indoor heating load. There are few indoor heat sources in residential buildings, and their winter heating load is mainly affected by outdoor temperature, while office buildings and commercial buildings have more indoor heat sources, and their winter heating load is affected in many ways. Fig.3 shows the heating load of different types of buildings.

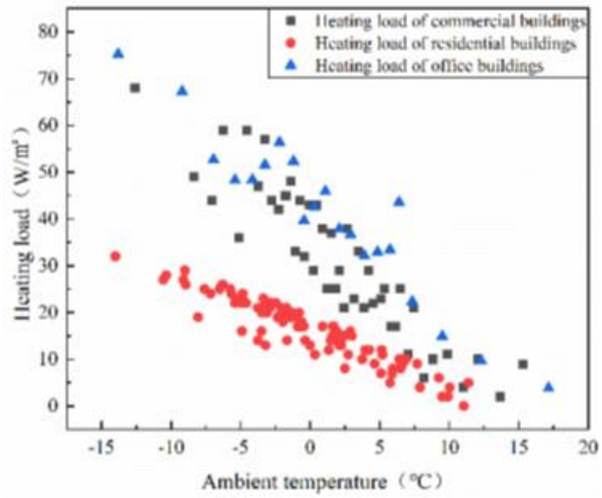

Fig. 3. Heating load of different types of buildings.

Due to the work and rest of the staff, residential buildings are heated almost all day, while office buildings and commercial buildings need heating only half the time. Therefore, it can be seen from the picture that the heat load point of residential buildings is more than that of the other two buildings. However, the air exchange times of residential buildings are small and the thermal insulation performance is good, so the heating load per square meter of residential buildings is lower than that of office buildings and commercial buildings. Fig.4 shows how the HSPF of different building types varies with the performance balance point. The best performance balance point of office building and commercial building is -6℃, and that of residential building is -8℃. Because the load of office building and commercial building changes slowly in the low temperature section, and the influence of partial load is limited, so the optimal equilibrium point of office building and commercial building is higher than that of residential building. Similarly, the change of heating load of different types of buildings can be seen from the changes of HSPF. When the equilibrium point changes from -9℃ to -6℃, the HSPF change of office buildings and commercial buildings is not obvious, while that of residential buildings has obvious inflection point, indicating that the linear relationship between heat load and temperature of residential buildings is stronger [10-20].

Select residential buildings with different levels of energy conservation in Beijing: 30%, 50%, 65%, 75% and near zero energy consumption; the type of heat pump is LATASHP. For heating season, buildings with different energy saving levels mean that their indoor heat load is affected by ambient temperature.

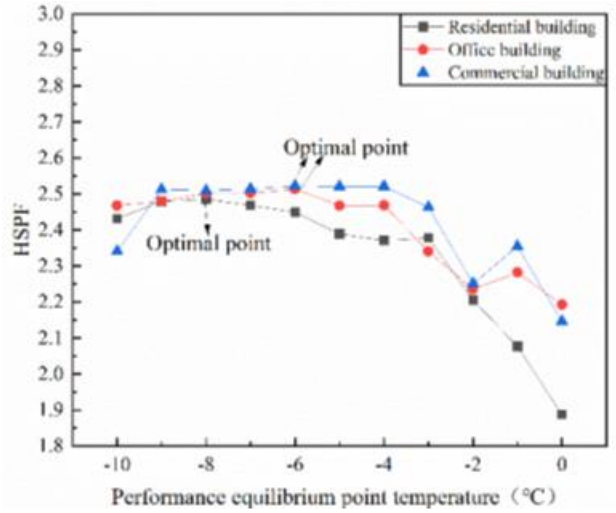

Fig. 4. The change of HSPF of different building types with the performance equilibrium point.

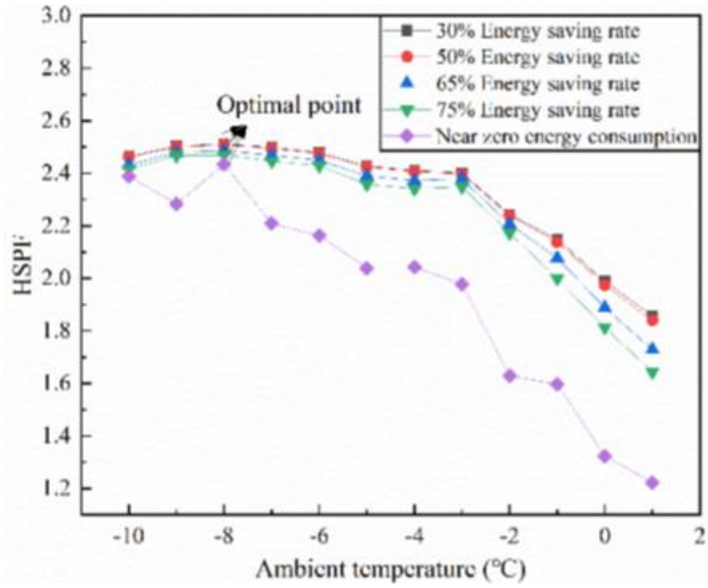

Fig. 5. The change of HSPF of different levels of energy saving with the performance equilibrium point.

The higher energy saving grade has better performance of enclosure structure, less affected by ambient temperature, and less linear. However, after the actual calculation, it is found that the temperature of the optimal performance equilibrium point is the same, which is -8 ℃. Based on the analysis of the load characteristics and other parameters in the calculation formula, it is found that this is because the linear degree is not reduced obviously, and the indoor heating load decreases proportionally with different energy saving levels, so it has little effect on the selection of the optimal equilibrium point temperature [10-20].

## 4. Conclusions

Through the optimization of the performance equilibrium point of the air source heat pump, the optimal equilibrium point temperature of typical cities in different climatic regions is determined. After analyzing the calculation results, the following conclusions are drawn:

- The HSPF of heat pump is greatly affected by ambient temperature, the HSPF of typical cities in different climatic regions have obvious stratification, the temperature of the optimal performance equilibrium point in hot summer and cold winter region is higher than that in cold region, and the optimal performance equilibrium point temperature in cold region is higher than that in severe cold region.
- Due to the slow change of load, the optimal performance equilibrium point of public buildings is higher than that of residential buildings in the same region.
- The optimal equilibrium points of different energy saving levels are close, but the corresponding seasonal coefficient of performance is different. The higher the energy saving grade, the better the performance.
- The optimal equilibrium point of the LATASHP is lower than that of the CASHP in the cold region, and the cold region is the focus of the next step of clean heating, so it is necessary to continue to develop a higher performance LATASHP.
- There are many parameters in the temperature optimization formula of the optimal performance equilibrium point, and the sensitivity analysis should be carried out later.

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