Simulation analysis of the influence of compressor speed on a solar-assisted heat pump hot water system

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Abstract. Inverter compressors have great potential in improving the overall performance of the solar-assisted heat pump (SAHP) system. However, in the actual process, due to the uncontrollability of environmental factors, the relationship between compressor speed, environmental factors and system performance is complicated, so that it is difficult to obtain intuitive data through operating experiments. Therefore, based on the SAHP system, a quasi-steady-state mathematical model is established using the Engineering Equation Solver (EES). And use this model to study the relationship between compressor speed and heat pump system performance. The simulation results show that when the ambient temperature is $20^\circ C$, if the heat collection efficiency is not lower than 0.45, when the solar radiation intensity is 400, 600 and 800 W/m$^2$, the best compressor speed values are 1600, 1800 and 2400 r/ min, the corresponding system coefficient of performance (COP) will reach the maximum value at this time, which are 7.7, 8.6 and 8.4 respectively. In addition, compared with higher solar radiation, at lower solar radiation intensity, increasing the compressor speed can significantly increase the heating capacity of the system.

1. Introduction
In the past few decades, with the increasingly serious energy and environmental problems, many countries have begun to devote themselves to the development and utilization of clean and renewable energy. Solar energy, as a kind of renewable energy, is considered to be an ideal and suitable new energy for development because of its safety and cleanliness, and it has attracted worldwide attention [1, 2]. The solar-assisted heat pump (SAHP) system benefits from excellent energy-saving effects, has been widely used, and still has great development potential [3]. At present, in pursuit of higher efficiency and better economy, many technologies using solar energy have been continuously developed, and various aspects of the SAHP system have been continuously studied.

Duarte et al. studied the influence of different working fluids on the performance of the direct expansion SAHPHW system [4]. When the solar radiation is 300–700W/m$^2$ and the ambient temperature is 10–35$^\circ C$, the COP of the system with R290 as the refrigerant is the best. Mohamed et al. studied the SAHP system under cold climate conditions [5]. They designed a unique multifunctional direct expansion SAHP system, combined with construction waste heat utilization technology, so the system can maintain efficient and stable operation in the cold winter. Rabelo et al. studied the relationship between the size of the collector area in the SAHPHW system, the refrigerant and the equipment investment recovery time [6]. Case studies conducted in different cities show that the recovery time of equipment investment is 1.77-3.24 years. Deng et al. established a mathematical
model to study the performance of air source and solar dual heat source heat pumps [7]. The dual-source heat pump has better overall performance than traditional solar heat pumps.

It is generally known that in the direct expansion SAHP system, the solar radiation energy is absorbed by the refrigerant in the solar collector, and the heat absorbed in the refrigerant is released to the water in the water tank during the condensation process. And, the compressor is inseparable to promote this cycle. Therefore, in the actual operation process, matching the compressor speed with the working conditions is beneficial to improve the performance of the SAHP system. However, in the actual process, because environmental factors are not easy to accurately control, it is difficult to obtain clear and intuitive data through experiments. For this reason, this study uses EES tools to establish a quasi-steady state mathematical model of the SAHP system, which is used to analyze the influence of frequency on system performance. The main purpose of this paper is to study the relationship between compressor frequency and system performance under different environmental conditions, which will help to further design and optimize the system. At the same time, the research will provide theoretical support and empirical reference for the next step of formulating the real-time control strategy of compressor speed.

2. System description
The direct expansion SAHP system is mainly composed of solar collector/evaporator, compressor, condensate tank, electronic expansion valve. The system schematic diagram is shown as in Figure 1. Among them, the electronic expansion valve is used to control the superheat of refrigerant at the input of the evaporator in a reasonable range, and feedback adjustment is used to control the opening of the expansion valve itself. In addition, in the water tank, a condensing coil is used to enhance the heat exchange of the refrigerant in the condensing process.

![Figure 1. Schematic diagram of direct expansion SAHP system.](image)

3. Mathematical model
The system model mainly includes solar collector/evaporator, compressor and condensate tank. These three models are related by the parameters under given conditions and the enthalpy difference between the refrigerants at various state points. A mathematical model was established to test the effect of compressor speed on the heat pump system based on the following assumptions:

1. The system is in a quasi-steady state within the simulation set time interval;
2. The pressure drop of the refrigerant in the heat collection and condensation process is ignored;
3. The process of refrigerant flowing through the expansion valve is regarded as an isenthalpic process;
4. The condensate tank model is assumed to be a non-layered model;
5. The effect of superheat in the evaporator on heat transfer is ignored.

3.1. Solar collector/evaporator
The heat absorbed by the refrigerant in the solar collector \(Q_u\) can be calculated by the Hottel-Whillier-Bliss correlation:
where $Q_u$ is the solar collector area, $F'$ is the collector efficiency factor, $\alpha$ is the absorption rate of collector plate, $T_r$ is the refrigerant temperature.

The collector efficiency factor ($F'$) is given by:

$$F' = \frac{\frac{1}{a}}{\frac{1}{a} + \frac{1}{b} + \frac{1}{c} + \frac{1}{d}}$$

where $W$ is the pipe spacing, $D_o$ is the outer diameter of the tube, $D_i$ is the inner diameter of the tube, $F$ is the fin efficiency, $C_p$ is the bond conductance between pipes and absorber plate, $h_r$ is the heat transfer coefficient of the refrigerant in the tube.

The fin efficiency ($F$) is given by:

$$F = \frac{\tanh[\sqrt{\frac{W}{\lambda_f}d_f(W-D_o)/2}]}{\sqrt{W/\lambda_f(d_f(W-D_o)/2)}}$$

where $\lambda_f$ is the thermal conductivity of the fin, $d_f$ is the thickness of the fin.

Ignoring the heat loss on the back and side of the collector plate, the total heat loss of the bare-plate solar collector can be written as two parts:

$$U_t = h_{wi} + h_{ra}$$

The coefficient of convective heat loss caused by wind blowing across the surface of the collector ($h_{wi}$) can be estimated as:

$$h_{wi} = 2.8 + 3v$$

where $v$ is the ambient wind speed.

The heat loss caused by thermal radiation ($h_{ra}$) can be calculated as:

$$h_{ra} = \epsilon(\sigma(T_p^2 + T_a^2))(T_p + T_a)$$

where $\epsilon$ is the emissivity of collector plate, $\sigma$ is the Stefan-Boltzmann constant, $T_p$ is the collector plate temperature, $T_a$ is the ambient temperature.

The Heat transfer coefficient of refrigerant in tube ($h_r$) can be given by Kandlikar correlation [8]:

$$\frac{h_r}{h_l} = C_0(C_0)^{C_2}(25F_{r_l})^{C_5} + C_3B_0C_4F_{r_l}$$

where $h_l$ is the heat-transfer coefficient assuming all mass to be flowing as liquid, $C_0$ is the convective number, $F_{r_l}$ is the Froude number with all flow as liquid, $B_0$ is the boiling number, $F_{r_l}$ is the fluid-dependent parameter, for R134a, the value is 1.63, $C_1$, $C_2$, $C_3$, $C_4$, $C_5$ are constants.

The convective number ($C_0$) is given by:

$$C_0 = \left(\frac{1-x}{x}\right)^{0.8}(\frac{\rho_g}{\rho_l})^{0.5}$$

where $x$ is the refrigerant dryness, $\rho_g$ is the gas phase density, $\rho_l$ is the liquid phase density.

The heat-transfer coefficient assuming all mass to be flowing as liquid ($h_l$) can be calculated as:

$$h_l = 0.023Re_l^{0.8}Pr_l^{0.4}\frac{\lambda_l}{\rho_l}$$

where $Re_l$, $Pr_l$ and $\lambda_l$ are the liquid phase Reynolds number, Prandtl number and thermal conductivity of the refrigerant respectively.

### 3.2. Compressor

In this paper, the mathematical model of the inverter compressor is established using the lumped parameter method.

The mass flow of refrigerant flowing through the compressor ($m_r$) can be calculated as:

$$m_r = \frac{N\phi V}{60\nu_e}$$

where $N$ is the compressor speed; $V$ is the theoretical displacement of the compressor; $\nu_e$ is the inspiratory specific volume; $\phi$ is the volumetric efficiency.

The volumetric efficiency ($\phi$) is given by [9]:

$$\phi = 0.959 - 0.006422\frac{P_c}{P_e}$$
where \( p_e, p_c \) are evaporation pressure and condensation pressure respectively.

The input power of compressor (\( W_{\text{com}} \)) can be calculated as:

\[
W_{\text{com}} = m_r \frac{p_e p_c \kappa}{\eta_{\text{com}} (\kappa - 1)} \left( \frac{p_c}{p_e} \right)^{\frac{\kappa - 1}{\kappa}} - 1
\]  
(12)

where \( \kappa \) is the polytropic index of compression process, \( \eta_{\text{com}} \) is the total efficiency of the compressor.

The total efficiency of the compressor (\( \eta_{\text{com}} \)) is given by:

\[
\eta_{\text{com}} = 0.874 - 0.0135 \frac{p_c}{p_e}
\]  
(13)

Besides, the input work of compressor can be written as enthalpy difference between inlet and outlet of compressor:

\[
\eta_{\text{com}} W_{\text{com}} = m_r (h_2 - h_1)
\]  
(14)

### 3.3. Condensate tank

The heat transfer capacity of refrigerant in the condenser (\( Q_c \)) can be expressed as the enthalpy difference of refrigerant at the inlet and outlet:

\[
Q_c = m_r (h_2 - h_3)
\]  
(15)

Besides, according to Newton's cooling formula, the total heat transfer in the condenser can also be written as follows:

\[
Q_c = U_c A_c (T_r - T_w)
\]  
(16)

where \( U_c \) is the total heat transfer coefficient in the condensate tank, \( A_c \) is the heat exchange area of the condensing coil, \( T_r \) is the temperature of the refrigerant in the condenser coil, \( T_w \) is the temperature of the water in the tank.

The total heat transfer coefficient in the condensate tank (\( Q_c \)) can be estimated as:

\[
U_c = \frac{\Delta T}{h_c + \frac{A_c}{h_w}}
\]  
(17)

The heat transfer coefficient of refrigerant side of single-phase flow can be given by Dittus-Boelter correlation:

\[
h = 0.023 \Re^{0.8} \Pr^{0.3} \frac{\lambda}{D_c l}
\]  
(18)

The heat transfer coefficient of refrigerant side of two-phase flow can be given by Shah correlation [10]:

\[
h_2 = h_{c,l} \left[ (1 - x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{(p/p_{cr})^{0.38}} \right]
\]  
(19)

where \( h_{c,l} \) is the heat-transfer coefficient assuming all mass to be flowing as liquid, which can be calculated by Eq. 18; \( x \) is the refrigerant dryness, \( p \) is actual pressure; \( p_{cr} \) is critical pressure.

The water side heat transfer coefficient (\( h_w \)) is given by [11]:

\[
h_w = 0.5 \frac{\lambda_w}{D_{c,o}} \left( \frac{\beta_w D_{c,o}^3 \rho_w^2 C_{p,w} \Delta_t}{\mu_w \lambda_w} \right)^{0.25}
\]  
(20)

where \( \lambda_w, \beta_w, \rho_w, C_{p,w} \) and \( \mu_w \) are the thermal conductivity, thermal expansion coefficient, density, specific heat capacity and dynamic viscosity of water, \( D_{c,o} \) is the outer diameter of condensing coil, \( \Delta_t \) is the heat transfer temperature difference.

Assuming that the heat leakage loss of the water tank is 5%, the heat gained by the water in the tank (\( Q_w \)) can be expressed as:

\[
Q_w = 0.95 Q_{\text{con}}
\]  
(21)

And the heat gained by the water in the tank can also be written as:

\[
Q_w = M_w C_{p,w} \frac{dt_w}{d\tau}
\]  
(22)

where \( M_w \) is the total mass of water in the tank, \( dt_w \) is the increased value of water temperature, \( d\tau \) is the time interval.
3.4. Performance evaluation index

The heat collection efficiency of the collector can be expressed as:

\[ \eta = \frac{q_u}{A_{st}} \]  

(23)

The coefficient of performance of the SAHP system can be expressed as:

\[ COP = \frac{q_w}{W_{com}} \]  

(24)

4. Results and discussion

The system parameters used in the simulation are shown in Table 1. Besides, the range of environmental factors in the simulation is as follows: the range of environmental temperature is 5-25°C, and the range of solar radiation intensity is 400-800W/m². The operating adjustment range of compressor speed is 1600-2400r/min. The simulation setting time step is 180s, and the iteration stop temperature is 50°C.

Table 1. Parameters used in the simulation.

| Parameter                                      | Value           |
|------------------------------------------------|-----------------|
| Refrigerant                                    | R134a           |
| Displacement volume of compressor              | 10.2 cm³/rev    |
| Collection area                                | 1.8m²           |
| Bond conductance                               | 16 W/(m·K)      |
| Emissivity of collector plate                  | 0.92            |
| Absorptivity of collector plate                | 0.9             |
| Thermal conductivity of collector plate         | 210 W/(m·K)     |
| Thickness of collector plate                   | 15mm            |
| Heat pipe spacing                              | 36mm            |
| Outer diameter of heat pipe                    | 8mm             |
| Ambient temperature                            | 20 °C           |
| Thickness of heat pipe                         | 0.5mm           |
| Water tank size                                | 200 L           |
| Thickness of condensing coil                   | 1mm             |
| Outer diameter of condensing coil              | 12mm            |
| Initial water temperature                      | 25 °C           |
| Wind speed                                     | 3 m/s           |
| Superheating                                   | 3 °C            |

Obviously, for the SAHP system, the solar radiation intensity is the most critical factor that determines the heat capacity of the system. Figure 2 shows the changing trend of the COP and η of the SAHP system with the increase of the compressor speed under different solar radiation intensities. From the figure, for any solar radiation intensity, the system COP decreases with the increase of compressor speed, and η increases with the increase of compressor speed. The main reason for this is that as the speed of the compressor increases, the input power of the compressor will also increase, thereby reducing the system COP. At the same time, the decrease in evaporation temperature and the decrease in heat loss also increase the system η. In addition, in the actual operation process, it is often necessary to ensure a more ideal value of η, such as η> 0.45. It can be clearly seen from Figure 2 that when the solar radiation intensity is 400 W/m², the compressor speed under the maximum system COP value is 1600r/min, and the corresponding system COP is 7.7. When the solar radiation intensity is 600 W/m², the compressor speed corresponding to the maximum system COP value is 1800r/min, and the corresponding system COP is 8.6. When the solar radiation intensity is 800 W/m², the compressor speed under the maximum system COP value is 2400r/min, and the corresponding system COP is 8.4. This means that when solar radiation increases, in order to maintain the heat collection efficiency of the system at a larger value, a higher compressor speed is required, which will inevitably lead to a decrease in system COP.
Figure 3 shows the time required for the water in the water tank to be heated to 50°C under two different compressor speeds when the solar radiation is 400 and 600 W/m² respectively. When the compressor speed is 1600r/min, the time it takes for the water to be heated to 50°C is 408min and 365min respectively. When the compressor speed is 2000r/min, the time it takes for the water to be heated to 50°C is 344min and 309min respectively. It is not difficult to find that increasing the compressor speed can significantly shorten the time consumed by the system under low solar radiation intensity. The reason for this phenomenon is that, compared with higher solar radiation intensity, when the compressor speed increases, the system COP decreases more slowly at lower solar radiation intensity, while the system η grows faster. These results show that compared with higher solar radiation, at lower solar radiation intensity, increasing the compressor speed can more significantly improve the increasing scale of heating capacity.

Figure 2. The influence of compressor speed on COP and η of SAHP system under different solar radiation intensity.

Figure 3. The influence of rotating speed on heating time of SAHP system under different solar radiation intensity.

Figure 4. The influence of compressor speed on COP and η of SAHP system under different ambient temperatures.

Figure 4 shows the influence of compressor speed on system COP and η under different ambient temperatures. As shown in the figure, when the temperature is 5, 15 and 25°C, the compressor speed increases by 200r/min each time, and the system COP decreases by 0.28, 0.38 and 0.51 respectively. This is because when the solar radiation intensity is constant, the higher the ambient temperature means the smaller the heat loss of the collector, and increasing the speed by the same value will bring about the same compressor power change. Therefore, compared with a lower ambient temperature, the speed change at a higher ambient temperature has a greater impact on the system COP. It can also be seen from Figure 4 that when the system is at a lower ambient temperature of 5°C, a higher
compressor speed is required to obtain the ideal system heat collection efficiency. The reason for this phenomenon is that even if the ambient temperature does not change, the evaporation temperature will decrease at a higher compressor speed, thereby reducing the heat loss of the system and increasing the system $\eta$, and the decline of the system COP has slowed down. This means that under low ambient temperature conditions, the heating capacity of the system can be increased by choosing to increase the compressor speed appropriately, although this will reduce the COP of the system to a certain extent.

5. Conclusions
In this paper, a mathematical model is established by EES, which is used to study the influence of compressor speed on SAHP system performance under different conditions. According to the experimental results, the following conclusions can be concluded:

1) The increase of the compressor speed will reduce the system COP and increase the system $\eta$. For different solar radiation intensities, on the basis of ensuring a good system $\eta$, an optimal COP can be obtained by adjusting the compressor speed. And the higher the solar radiation means the higher the compressor speed when the best system COP is achieved.

2) Compared with higher solar radiation intensity, when the rotation speed increases, the system COP decreases more slowly at lower solar radiation intensity, while the system $\eta$ increases faster. This means that at lower solar radiation intensity, increasing the compressor speed can more significantly improve the heating capacity of the system. In addition, for working conditions with low ambient temperature, increasing the compressor speed will slightly reduce the system COP, but it can increase the system $\eta$ and system heating capacity. These can provide an effective empirical reference for formulating the real-time control strategy of compressor speed.

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