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Performance improvement of vapor-injection heat pump system by employing PVT collector/evaporator for residential heating in cold climate region

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Abstract

The adoption of vapor injection (VI) cycle could overcome the defects of the conventional one-stage air source heat pump (ASHP) system under extremely low ambient temperature conditions. Nevertheless, the ASHP system with VI cycle still could not operate efficiently due to the low evaporating temperature in the fin-tube evaporator. For such a situation, the PVT (Photovoltaic/Thermal) collector/evaporator could reach higher evaporating temperature attributed to its physical structure, thereby improving the system COP. This study proposes the detailed mathematical model of the vapor-injection heat pump system by incorporating PVT collector/evaporator and verifies the effectiveness of the proposed system. Parametric studies have been then conducted. The proposed system can operate off-grid and its COP can reach 4.0 at the ambient temperature of -10 °C and the solar irradiation of 500 W/m². The practical electrical efficiency of PV panels achieves at 15.1%, whereas the thermal efficiency of the system is 44.8%. A hybrid control method including three modes has been also proposed based on the results for improving the system performance. The levelized cost of heat (LCOH) of this system is 0.054 \$/kWh, which is 51.5% lower than that of electric heating system (0.111 \$/kWh).

Keywords: vapor injection; PVT; direct expansion; solar assisted heat pump; comparative study; economic analysis

1 Introduction

The development of society and increasing urbanization need constant energy consumption, and the overexploitation of primary energy has raised concerns about the energy and environmental crises [1]. The building energy consumption accounts for about 40% of the world's total energy consumption [2]. Furthermore, the residential heating consumption takes a large portion of buildings in the cold climate region and the heating supply is mainly from the fossil fuels [3]. In this regard, the heat pump technology was proposed to realize efficient-operation and low-energy consumption residential heating supply in the building sector [4-6].

The air source heat pump (ASHP) systems have been recently developed and employed for efficient heating [7, 8]. Redón et al. [9] indicated two major problems of the ASHP working in low ambient temperature conditions, efficiency diminution, and the high compressor discharge temperatures. The adoption of vapor injection (VI) cycle could assist the ASHP to operate normally

under low-temperature conditions. For instance, Wang et al. [10] developed the vapor injection technique based on flash tank for two-stage ASHP operation under cold climate, and presented a novel control strategy to seek the optimized intermediate pressure for better performance. Additionally, Ko et al. [11] investigated the influence of volume ratio of a two-stage rotary compressor on ASHP experimentally, and it is found that the ASHP with VI cycle improves the heating power of 48% and the COP (coefficient of performance) of 36% in the comparison with the ASHP at the ambient temperature of -15 °C and water temperature of 60 °C. Nevertheless, the low evaporating temperature in the fin-tube evaporator of the ASHP system with VI cycle still limits the efficient operation of the system. Therefore, it is important to find a way that can increase the evaporating temperature for higher system efficiency. On the other hand, solar thermal could be a widely available low-temperature heat source for heat pumps due to its characteristics [12, 13]. Solar assisted heat pump (SAHP) has great potential in building energy-saving [14-17]. The direct expansion solar assisted heat pump (DX SAHP) is one type of the SAHP and it has become state of the arts research due to its stability in heat production [18].

The DX SAHP systems can be mainly divided into two categories: direct expansion evaporator (DXE) based SAHP and photovoltaic/thermal (PVT) based SAHP [19, 20]. The direct expansion evaporator could harvest the thermal energy from both solar and air when the ambient temperature is higher than the evaporating temperature. The merits of incorporating PVT modules into a SAHP are that it could produce thermal energy and electricity simultaneously, and the PV cells could obtain higher electrical efficiency due to the lower operating temperature [21, 22]. That is because the waste heat of the PV panels would be absorbed by the solar collector. Therefore, the PVT based DX SAHP has become a research hotspot due to its high comprehensive efficiency, and numerous works of this topic have been reported recently. Tsai [23] adhered multi-port copper tubes to the rear side of the PV panels as a thermal absorber to form a PVT module. Moreover, the PVT module was used as a solar collector/evaporator to produce heat for domestic hot water supply. The maximum PV efficiency is 12.43% while the overall efficiency is 86.23%. For the green building applications, Shao et al. [24] proposed a DX roof-PVT SAHP system for water heating. Their experimental results demonstrate that under winter conditions (ambient temperature ranges from 0°C to 10 °C), the electrical, thermal, and overall efficiencies are 11.67%, 60.17%, and 78.84%, respectively, and the system COP could reach 3.7. Numerous experiments have been conducted to evaluate the actual operation performance, for instance, Zhou et al. [25] conducted an experiment of PVT HP system based on four PVT modules during summer, the average solar irradiation is 657 W/m². They found that the average COP could reach 6.16 while the heating power is 4.7 kW. Mohanraj et al. [22, 26] designed and manufactured a novel PVT evaporator with circular and triangular tube configurations, and experimentally investigated its thermal and electrical performance. Moreover, they proposed an artificial neural network model to evaluate and forecast the system operation status. The variance analysis reveals that the solar irradiation and ambient temperature are the most influencing parameters which would significantly affect the performance of PVT evaporator.

However, the performance (including evaporating temperature, condensing temperature, heating power, compressor power, etc.) of the DX SAHP system would be influenced by the environmental conditions significantly including solar irradiation, ambient temperature, etc. Especially, it also has operating restrictions under extreme cold weather conditions due to the same reason of ASHP, the evaporating temperature would become very low to absorb enough heat. Therefore, the vapor injection (VI) cycle has also been adopted to enhance the stability of the heat

pump system and heating power under cold climate conditions [27-29]. For instance, Chen et al. [30] developed a novel DX solar assisted vapor injection heat pump cycle for hot water application. The new cycle maximum improvement of the heating COP could reach 18.3% and the 64.4% of heating power in the comparison of conventional cycle in specific circumstance. In order to optimize the system performance, Zheng et al. [31] added a bypass circuit to avert liquid slugging in the compressor and this by pass mechanism can efficiently enhance the thermal performance of the novel vapor injection cycle system. The system COP can increase by 9.1% and 7.3% in specific conditions, for R290/R600a (50/50) and R32/R1234yf (40/60), respectively. In winter conditions, Lu et al. [32] presented a PVT based DX SAHP employing vapor injection cycle and conducted field tests. The average system COP is 3.27 when the ambient temperature is $-1.1\text{ }^{\circ}\text{C}$ and solar radiation intensity is 164 W/m^2 during the experiments.

There currently a few researches on the vapor-injection heat pump system employing PVT collector/evaporator for residential heating, but it is not clear about the details of their system performance in cold climate region. In this paper, a detailed mathematical model of the vapor-injection heat pump system employing PVT collector/evaporator is developed and verified by experimental results. Secondly, the influence of environmental parameters and operating parameters on system performance has been further studied. Finally, the performance of the proposed system, the ASHP system with VI cycle and the existing PVT SAHP system with VI cycle, has been compared to reveal the merits of the proposed system. The novelty of this study is to paratactic arrange the conventional finned-tube evaporator and PVT module, and propose a hybrid control method. The objective of this paper is to improve the system performance of the vapor-injection heat pump by utilizing PVT collector/evaporator under severe cold conditions.

2 System description

Fig. 1 shows the schematic diagram of the vapor-injection heat pump system employing PVT collector/evaporator (PVT SAHP system with VI cycle) for residential heating. The proposed system consists of evaporator (PVT collector/evaporator), fin-tube evaporator, two-stage compressor, condenser, expansion valve I/II, economizer, circulating pump, buffer water tank, water tank, underfloor heating, MPPT solar energy controller, accumulator, and inverter. The economizer is adopted to form a vapor injection (VI) cycle. The working principles are as follows: after the condensation process, the refrigerant at state 5 would be separated into two parts, the mainstream refrigerant would flow through the economizer and be cooled to state 6 by the bypass refrigerant (refrigerant in the vapor injection cycle). Then, the sub-cooled refrigerant at state 6 would be expanded to state 7 by expansion valve II. Thus, the refrigerant at state 7 could absorb heat from the evaporator and transform to state 1. The mainstream refrigerant at state 2 which has been compressed through first stage compression would be mixed with the bypass refrigerant at state 9 (already heated by economizer). The mixed refrigerant at state 3 would be further compressed to state 4 through second stage compression. Afterward, the refrigerant at state 4 could release heat through the condenser and transform to state 5.

This system could operate in three modes including a) one-stage mode with solar assisted (expansion valve I closed; expansion valve II opened; magnetic valve I opened; magnetic valve II closed), b) two-stage mode with solar assisted (expansion valve I opened; expansion valve II opened; magnetic valve I opened; magnetic valve II closed), and c) parallel ASHP mode with two stage cycles (expansion valve I opened; expansion valve II opened; magnetic valve I closed;

magnetic valve II opened). A hybrid control method could be established based on these three modes for better system performance. The compressor could switch two-stage mode to one-stage mode through closing the expansion valve I, and the system could operate in parallel ASHP mode through closing the magnetic valve I. The parallel fin-tube evaporator is employed to form an ASHP cycle and extract heat from the ambient air during the night or cloudy days for heat applications. The condensation heat from the heat pump could be used for domestic hot water supply or be used for space heating through the underfloor heating system. On the other hand, the PVT module generates both electrical power and thermal energy simultaneously. The electricity produced by the PV panels could drive the compressor and circulating pump to realize zero-energy consumption, and the surplus electrical power could output to the grid for economic benefits.

Fig. 1. Schematic diagram of the vapor-injection heat pump system employing PVT collector/evaporator for residential heating.

Fig. 2 demonstrates the pressure-enthalpy diagrams of the conventional vapor compression cycle (Blue line), VI cycle employing fin-tube evaporator (Green line), and VI cycle employing PVT collector/evaporator (Redline). Different from the traditional vapor compression cycle, the VI cycle has two compression stages in the two-stage compressor. The adoption of economizer could expand the low temperature operating range of the thermodynamic cycle. The evaporating temperature (state 7) of the conventional vapor compression cycle is higher than the evaporating temperature (state 7') of the VI cycle employing fin-tube evaporator, and a low evaporating temperature allows the system to absorb thermal energy from low-temperature ambient air. However, the VI cycle employing fin-tube evaporator could not operate with high efficiency due to the low evaporating temperature (state 7'). In this circumstance, the VI cycle employing PVT collector/evaporator could reach a higher evaporating temperature (7'') attributed to its physical structure, and then leads to a higher system COP.

In the vapor injection cycle, the refrigerant in state 2'/2'' which is compressed from state 1'/1'' would be mixed with the refrigerant from the economizer (state 9'/9''). Then, the refrigerant vapor in state 3'/3'' would be processed to the discharged refrigerant (state 4'/4'') through second compression stage. The P_e , P_i , P_c refer to the evaporating pressure, vapor injection pressure, and

condensing pressure, respectively. The Q_{th} refers to the heat absorb rate in the evaporating process, and the Q_{cond} refers to the heating power of the system.

Fig. 2. Pressure-enthalpy diagrams of the conventional vapor compression cycle, VI cycle employing fin-tube evaporator, and VI cycle employing PVT collector/evaporator.

3 Mathematical model

This section presents the mathematical model of the vapor-injection heat pump system employing PVT collector/evaporator. Following assumptions are made for simulation: (a) The two-stage compressor and the expansion valves are regarded as steady-state components, and the internal volumes of compressor, valves, magnetic valves, three-way valves are neglected. (b) Refrigerant in the system is considered homogeneous. (c) Oil effects are ignored. Additionally, the mathematical model was solved in MATLAB.

3.1 PVT collector/evaporator

Fig. 3(a) shows the front side of the PVT module (PVT collector/evaporator), which is fully exposed to the solar radiation and transfers it into electrical power. In the meantime, a fraction of the radiation would be assimilated by the PV module and convert to heat. Thus, the PV cells temperature would increase and lead to a diminution of the PV cells' electrical efficiency. However, the waste heat could be extracted by the working fluid in the roll-bond evaporator (roll-bond panel) as useful heat for heat applications. Fig. 3(b) presents the physical structure of the roll-bond evaporator with the fluid channel which is encapsulated in the PVT module. Fig. 3(c) demonstrates the multi-layer structure of the PVT module and the heat loss model. The PVT module consists of seven layers: PV-glazing cover, PV cells, EVA grease, electrical insulation, EVA grease, roll-bond panel, and thermal insulation. As shown in Fig. 3(c), the upper surface (attached to the PV module through EVA grease) of the roll-bond panel is flat to ensure good thermal conductance, but the lower surface of the roll-bond panel (Fig. 3(b)) is raised due to the fluid channel.

Fig. 3. (a) Front side of the PVT module. (b) Physical structure of the lower surface of the roll-bond panel. (c) Heat loss model and cross-section view of the PVT module.

The electrical power generated by the PV module can be calculated by:

$$Q_e = A \cdot I \cdot \tau_c \cdot \alpha_{pv} \cdot \beta_{pv} \cdot \eta_e \quad (1)$$

where A is the area of PVT module which is determined as 2 m^2 due to the limitation of manufacture process; I is the insolation; τ_c is the transmittance of the PV-glazing cover; α_{pv} is the absorption ratio of the PV cells; β_{pv} is the packing factor of PV panel; $\eta_{e,ideal}$ is the ideal PV cells' electrical efficiency and it can be expressed as [33]:

$$\eta_{e,ideal} = \eta_{rc} \cdot \left[1 - \beta_p \cdot (T_p - T_{rc}) \right] \quad (2)$$

where η_{rc} is the standard photovoltaic efficiency value of the PV cells at 25°C ; β_p is the temperature coefficient ($1/^\circ\text{C}$) of the PV cells' efficiency [33]; T_p is the PV cells' temperature; T_{rc} is the reference standard temperature (25°C).

The thermal energy assimilated by the PVT panel can be expressed as:

$$Q_{abs} = A \cdot I \cdot \tau_c \cdot \left[\alpha_{pv} \cdot \beta_{pv} \cdot (1 - \eta_e) + \alpha_b \cdot (1 - \beta_{pv}) \right] \quad (3)$$

where α_b is the absorption ratio of the baseboard (electrical insulation); the total heat loss rate of PVT module can be expressed as:

$$Q_L = A \cdot U_L \cdot (T_p - T_a) \quad (4)$$

where T_a is the environment temperature; U_L is the total heat loss coefficient, and as shown in Fig. 3(c), the heat loss of the PVT module includes: (1) heat transfer from the PV cells to PV-glazing cover; (2) heat transfer from the PV-glazing cover to ambient. Thus, the total heat loss coefficient could be expressed as:

$$U_L = \left[1 / (h_{cd,p-c} + h_{rd,p-c}) + 1 / (h_{cv,c-a} + h_{rd,c-a}) \right]^{-1} \quad (5)$$

The calculation expression of each heat transfer coefficient could refer to the authors' previous work [34]. The heat transfer rate of the PVT panel equals the heat transfer rate of the assimilated thermal energy minus the total heat loss, calculated by:

$$Q_{th} = Q_{abs} - Q_L \quad (6)$$

The thermal energy absorbed by the PVT module equals to the heat assimilated through the refrigerant through its evaporating process:

$$Q_{eva} = Q_{th} \quad (7)$$

The characteristic parameters of the PVT module for further simulation are listed in Table. 1. In this study, the refrigerant R404A is selected as the working fluid. R404A has a low boiling point which is 226.9 K, and it could allow the refrigerant to work at low temperatures. Moreover, R404A presents the lowest critical point, thus, it is the most appropriate working fluid for low and medium temperature refrigeration applications. In addition, it has low toxicity and is not flammable [35]. The above-mentioned system is proposed to work in cold climate regions which means a low ambient temperature. In this regard, the R404A is more suitable for this system than R134a.

Table. 1. Characteristic parameters of the PVT collector/evaporator.

Parameters	Nomenclature	Value	Unit
Thickness of PV-glazing cover	δ_c	1	mm
Emissivity of PV-glazing cover	ε_c	0.84	[-]
Transmissivity of PV-glazing cover	τ_c	0.9	[-]
Thickness of PV cells	δ_{pv}	0.3	mm
Emissivity of PV cells	ε_{pv}	0.96	[-]
Absorptance of PV cells	a_{pv}	0.85	[-]
Thermal conductivity of PV cells	κ_{pv}	203	W/m·°C
Standard electrical efficiency of PV cells	η_{rc}	18	%
Temperature coefficient of PV cells	β_p	-0.45	%/°C
Absorptance of PV baseboard	a_b	0.8	[-]
Thickness of EVA grease	δ_{EVA}	0.5	mm
Thermal conductivity of EVA grease	κ_{EVA}	0.311	W/m·°C
Thickness of electrical insulation	δ_{ei}	0.5	mm
Thermal conductivity of electrical insulation	κ_{ei}	0.15	W/m·°C
Electrical insulation material	[-]	Tedlar	[-]
Packing factor	β_{pv}	0.9	[-]
Length of PVT collector/evaporator	L	2.0	m
Width of PVT collector/evaporator	W	1.0	m
Area of the PVT collector/evaporator	A	2.0	m ²
Number of the PVT modules	n	10	[-]
Total area of the PVT module	A_{tot}	20.0	m ²
Thermal conductivity of roll-bond panel	κ_{rb}	151	W/m·°C
Thickness of roll-bond panel	δ_{rb}	1	mm

Width of the fluid channel	W_{rb}	10	mm
Height of the fluid channel	H_{rb}	2.8	mm
Refrigerant type	ref	R404A	[-]

3.2 Compressor

The vapor-injection scroll compressor is adopted in this study. For a two-stage compressor, the refrigerant mass flow rate $m_{ref,1}$ of the first compression stage could be expressed as [36, 37]:

$$m_{ref,1} = \lambda_v \cdot V_{th} / v_1 \quad (8)$$

where the λ_v is the compressor volumetric efficiency, V_{th} is the theoretical displacement volume of the compressor in the first stage [38], v_1 is the specific volume of the refrigerant in the suction period in the first stage.

The compressor power in the first stage can be calculated by:

$$P_{com,1} = m_{ref,1} \cdot (h_2 - h_1) / \eta_{ele,1} \quad (9)$$

where h_2 and h_1 are the enthalpy of discharged and suction refrigerant in the first compression stage, respectively; $\eta_{ele,1}$ is the electrical efficiency of the compressor in the first stage which could be expressed as:

$$\eta_{ele,1} = -0.17938 + 0.87501 \frac{p_2}{p_1} - 0.30014 \left(\frac{p_2}{p_1} \right)^2 + 0.04135 \left(\frac{p_2}{p_1} \right)^3 - 0.00206 \left(\frac{p_2}{p_1} \right)^4 \quad (10)$$

The compression stage is treated as isentropic, the temperatures and pressures of the suction and discharge stages satisfy this equation:

$$T_2 / T_1 = (p_i / p_e)^{(\chi-1)/\chi} \quad (11)$$

where χ is the polytropic index of refrigerant [11]; p_i (equal to p_2) and p_e (equal to p_1) are the pressures of the discharged and suction refrigerant in the first compression stage, respectively.

The refrigerant' mass flow rate in the vapor injection cycle can be calculated by:

$$m_{ref,2} = \beta \cdot m_{ref,1} \quad (12)$$

where β is the mass flow rate ratio of injected vapor; the injected vapor pressure could be calculated as:

$$P_i = \sqrt{P_e \cdot P_c} \quad (13)$$

where P_e and P_c are the evaporating pressure and condensing pressure, respectively.

The mixing process in the compressor is assumed as isochoric without pressure drop, then the mixing process could be calculated by:

$$(m_{ref,1} + m_{ref,2}) \cdot h_3 = m_{ref,1} \cdot h_2 + m_{ref,2} \cdot h_9 \quad (14)$$

The compressor power of the second stage can be expressed as:

$$P_{com,2} = (m_{ref,1} + m_{ref,2}) \cdot (h_4 - h_3) / \eta_{ele,2} \quad (15)$$

where h_4 and h_3 are the enthalpy of discharged and suction refrigerant in the second compression

stage, respectively; $\eta_{ele,2}$ is the electrical efficiency of the compressor in the second stage which could be expressed as:

$$\eta_{ele,2} = -0.17938 + 0.87501 \frac{p_4}{p_3} - 0.30014 \left(\frac{p_4}{p_3} \right)^2 + 0.04135 \left(\frac{p_4}{p_3} \right)^3 - 0.00206 \left(\frac{p_4}{p_3} \right)^4 \quad (16)$$

The compression stage is treated as isentropic, the temperatures and pressures of the suction and discharge stages satisfy this equation:

$$T_4 / T_3 = (p_c / p_i)^{(\chi-1)/\chi} \quad (17)$$

where p_c (equal to p_4) and p_i (equal to p_3) are the pressures of discharged and suction refrigerant in the second compression stage, respectively.

3.3 Condenser

The condensation heat could be calculated by:

$$Q_{cond} = (m_{ref,1} + m_{ref,2}) \cdot (h_4 - h_5) \quad (18)$$

where h_4 and h_5 are the inlet and outlet refrigerant enthalpy in the condenser, respectively.

The condensation heat is equal to the heating power of the SAHP systems for heating applications including residential heating, space heating or domestic hot water.

3.4 Expansion valve

The expansion process is treated as adiabatic and isenthalpic when the refrigerant flows through the expansion valve I and II [36]. Therefore, the enthalpy of inlet refrigerant equals to the enthalpy of the outlet refrigerant for the VI cycle which is shown in Fig. 2(a):

$$h_8 = h_5 \quad (19)$$

$$h_7 = h_6 \quad (20)$$

where h_5 and h_8 are the inlet refrigerant enthalpy and outlet refrigerant enthalpy through the expansion valve I, respectively; h_6 and h_7 are the inlet refrigerant enthalpy and outlet refrigerant enthalpy through the expansion valve II, respectively.

3.5 Economizer

The system has an economizer to subcool the refrigerant in the mainstream. In the meantime, the bypass refrigerant in the vapor injection cycle would absorb thermal energy from the refrigerant in the mainstream. The energy conservation equation of this process could be calculated by:

$$m_{ref,1} \cdot (h_5 - h_6) = m_{ref,2} \cdot (h_9 - h_8) \quad (21)$$

where h_5 and h_6 are the inlet refrigerant enthalpy and outlet refrigerant enthalpy through the economizer of the mainstream, respectively; h_8 and h_9 are the inlet refrigerant enthalpy and outlet refrigerant enthalpy through the economizer of the vapor injection cycle, respectively.

3.6 Economic assessment

Levelized cost of heat (LCOH) is an economic evaluation index to compare the economic performance for the proposed PVT based SAHP system with VI cycle and other systems. The definition of LCOH is shown in Eq. 20 [39]:

$$LCOH = \frac{I_0 - S_0 + \sum_{t=1}^T \frac{C_t(1-TR) - DEP_t \cdot TR}{(1+r)^t} - \frac{RV}{(1+r)^T}}{\sum_{t=1}^T \frac{E_t}{(1+r)^t}} \quad (22)$$

where $LCOH$ is the levelized cost of heat (\$/kWh); I_0 is the initial investment for the investigated system (\$); S_0 is subsidies and incentives (\$); C_t is the operation and maintenance costs in year t (\$); TR is the tax rate (%); DEP_t is the asset depreciation in year t (\$); RV is the residual value of the system (\$); r is the discount rate (%); E_t is the saved final energy in year t (kWh). The detailed values of different economic indicators have shown in Table. 2.

Table. 2. Detailed values of different economic indicators.

Parameters	Nomenclature	Value	Unit
Initial investment for the investigated system	I_0	257.1	\$
Subsidies and incentives	S_0	0.0	\$
Operation and maintenance costs in year t	C_t	150.2	\$
Tax rate	TR	0.16	%
Asset depreciation in year t	DEP_t	0.06	\$
Residual value of the system	RV	0.1	\$
Discount rate	r	0.03	%
Saved final energy in year t	E_t	730	kWh
Durable year	T	15	year
Average daily working hours	t_{day}	3.5	hour

3.7 Performance indices

The definition of the system COP (Coefficient of Performance) is the ratio of the total heating power and compressor power and could be expressed as [40]:

$$COP = (m_{ref,1} + m_{ref,2}) \cdot (h_4 - h_5) / (P_{com,1} + P_{com,2}) \quad (23)$$

The η_{th} is the PVT collector/evaporator's thermal efficiency, which could be expressed as:

$$\eta_{th} = Q_{th} / (A \cdot I) \quad (24)$$

The η_e is the PV module's electrical efficiency and it could be defined as:

$$\eta_e = Q_e / (A \cdot I) \quad (25)$$

The η_{ove} is the overall efficiency which can be defined as [41]:

$$\eta_{ove} = \eta_e / 0.38 + \eta_{th} \quad (26)$$

The TCR is the total compression ratio which could be defined as:

$$TCR = P_c / P_e \quad (27)$$

4 Experimental validation

The mathematical model of the vapor-injection heat pump system employing PVT collector/evaporator (PVT SAHP system with VI cycle) should be verified by experiments to ensure its reliability. As shown in Fig. 2, the thermodynamic cycles of the VI cycle employing fin-tube evaporator and the VI cycle employing PVT collector/evaporator are the same. Therefore, it is reliable to verify the mathematical model of the vapor injection cycle through the ASHP system with VI cycle. The only difference between the above-mentioned two systems is the type of evaporator. The mathematical model of PVT collector/evaporator has been verified in the authors' previous paper [42], and the mathematical model of the fin-tube evaporator is referred to Huang's paper [43]. Therefore, the experimental results in references [44] was adopted for validation of the mathematical model of the vapor injection cycle. The specific parameters in the experiments used for verification are listed in Table. 3.

Table. 3. Experimental parameters [44] of the air source heat pump system with VI cycle.

Parameters	Nomenclature	Value	Unit
Rated power of compressor	P_{com}	360~910	W
Type of compressor	[-]	SXT200LCJW	[-]
Power source specification of compressor	V_{com}	220	V
Water tank volume	V_{water}	200	L
Refrigerant charge	M_{ref}	1.4	kg
Rated hot water yield	v_{water}	35~80	L/h
Outlet water temperature	T_o	55	°C
Ambient temperature range	T_a	-20~10	°C
Refrigerant type	ref	R410A	[-]

Fig. 4 shows the comparison results and error analysis of the experimental and simulation COP and compressor power. The errors are mainly caused by the precision of the sensors and the propagation errors generated by the proxy variable combinations. The simulation COP is higher than the experimental COP due to the assumption of no refrigerant pressure loss in each component. In this circumstance, the simulation compressor power would lower than that of experimental results. Both the experimental and simulation COP increase when the ambient temperature rises. Higher ambient temperature would result in an increment of evaporating temperature and then benefits the system performance. The lowest experimental COP is 1.78 which is obtained when the ambient temperature is -20 °C. In the meantime, the simulation COP is 1.93 and the relative error is 8.16% while the relative error of compressor power is 3.79% (maximum error). The highest experimental COP (3.69) occurs at the ambient temperature of 10 °C, meanwhile, the simulated COP is 3.80 and the relative error is 3.05%. Higher ambient temperature would benefit the system performance significantly. The average relative error of the COP is 4.40% while the maximum relative error is 8.16%, and the average relative error of the compressor power is 1.56%. The results show that the simulation results are in well-match with the experimental results. Consequently, the mathematical model used for further performance analysis is considered reliable.

Fig. 4. Error analysis of the experimental and simulated (a) COP; (b) compressor power varying with ambient temperature.

5 Performance analysis

In this section, the performance analysis of the proposed system has been carried out under different conditions, including ambient temperature, solar radiation intensity, and mass flow rate ratio of injected vapor. Moreover, the system performance comparison of the proposed system and the ASHP system with VI cycle and the existing PVT SAHP system with VI cycle has been carried out to show the performance improvement of the proposed system. Finally, the economic analysis has been presented in the last sub-section to reveal its economic potential compared with other residential heating systems. The area of PVT module is 20 m² in this analysis. The working conditions for the simulation have been listed in Table. 4.

Table. 4. Working conditions for the simulation.

Parameters	Nomenclature	Value	Unit
Ambient temperature	T_a	-20~10	°C
Solar radiation intensity	I	200~800	W/m ²
Mass flow rate ratio of injected vapor	γ	0~0.15	[-]
Wind velocity	v	1.0	m/s
Total area of PVT module	A_{tot}	20.0	m ²

5.1 Ambient temperature

In this sub-section, the performance indices are carried out under the working conditions: ambient temperature ranges from -20 °C to 10 °C, solar irradiation is 500 W/m², wind speed is 1 m/s, and mass flow rate ratio of injected vapor is 10%. This system could operate in one-stage mode (close expansion valve I) and two-stage mode.

Fig. 5(a) illustrates the influences of ambient temperature on the electrical, thermal, and overall efficiencies of the system in both one-stage and two-stage modes. It could be found that the two-stage mode is more efficient than the one-stage mode. The electrical efficiency of the PV module

decreases linearly from 15.51% to 14.07% (15.26% to 14.01% in one-stage mode) while the thermal efficiency increases from 40.18% to 49.72% when the ambient temperature increases in the two-stage mode. Higher ambient temperature would result in an increment of the PV cells' temperature and then lead to a lower electrical efficiency because of the PV cells' temperature effect. Moreover, the evaporating temperature would increase when ambient temperature increases, then the heat loss rate between the PVT panel and the environment would reduce and lead to higher thermal efficiency. The overall efficiency increases from 81% to 86.75%, and the peak value which is 86.75% occurs at the ambient temperature of 10 °C

As shown in Fig. 5(b), the system COP constantly rises from 3.76 to 5.20 when the ambient temperature varies from -20 °C to 10 °C of the two-stage mode. Higher ambient temperature would increase the PV cells' temperature as well as the evaporating temperature, and then leads to a higher system COP. However, the system COP and the evaporating temperature of the one-stage mode would higher than that of the two-stage mode when the ambient temperature exceeds 1.5 °C. For instance, the system COP of one-stage mode could reach 6.26 which is 20.4% higher than that of two-stage mode when the ambient temperature is 10 °C. In the meantime, the evaporating temperatures of two-stage mode and one-stage mode are 7.4 °C and 10.5 °C, respectively. That is because the adoption of the VI cycle would cause a diminution of the heat extract capacity of the evaporator from the environment under high ambient temperature conditions. In this regard, the system should switch two-stage mode to one-stage mode when the ambient temperature exceeds 1.5 °C for better system performance.

Fig. 5. Influence of ambient temperature on (a) electrical, thermal, and overall efficiencies; (b) system COP and evaporating temperature for one/two-stage modes.

As shown in Fig. 6(a), the evaporating pressure, condensing pressure, and vapor injection pressure increase constantly when the ambient temperature increases for both one-stage mode and two-stage mode, while the total compression ratio decreases from 5.11 to 4.32 of the two-stage mode (5.4 to 4.0 of the one-stage mode). The highest total compression ratio (5.11 of two-stage mode) occurs at the ambient temperature of -20 °C, meanwhile, the evaporating pressure is 0.41 Mpa, vapor injection pressure is 0.92 Mpa, and the condensing pressure is 2.07 Mpa. In this circumstance, the evaporating pressure and condensing pressure of one-stage mode are 0.47 Mpa and 2.38 Mpa, respectively. It could be found that the first compression ratio of the two-stage mode is much lower than the total compression ratio of the one-stage mode, which means the adoption of a two-stage

compressor would significantly decrease the compression ratio for a better operation state of the compressor.

Fig. 6(b) demonstrates the influence of ambient temperature on evaporating temperature, condensing temperature, evaporating-ambient temperature difference, and PV cells' temperature under the hybrid control method. The evaporating and condensing temperatures show the same variation trend as evaporating and condensing pressures in Fig. 6(a). The evaporating-ambient temperature difference decreases from 8.4 °C to -2.6 °C when the ambient temperature increases from -20 °C to 10 °C, while the PV cells' temperature increases from -3.1 °C to 20.1 °C. The lowest evaporating and condensing temperatures are -11.6 °C and 45.4 °C when the ambient temperature is -20 °C, respectively.

Fig. 6. Influence of ambient temperature on (a) evaporating pressure, condensing pressure, vapor injection pressure, and total compression ratio of one/two-stage modes; (b) evaporating temperature, condensing temperature, evaporating-ambient temperature difference, and PV cells' temperature.

The PVT module could produce thermal energy and electricity simultaneously, thus, the PVT based SAHP systems could realize zero-energy consumption. Fig. 7 illustrates the effect of ambient temperature on heating power, output power to grid, compressor power, and electrical power of the system under the hybrid control method. The heating power of the system increases from 4811 W to 5634 W when the ambient temperature rises from -20 °C to 10 °C. The electrical power produced by the PV cells could drive the compressor under different ambient temperatures at the solar irradiation of 500 W/m². Meanwhile, the output power to grid fluctuates around 189.4 W to 500.3 W under various ambient temperatures.

Fig. 7. Influence of ambient temperature on heating power, output power to grid, compressor power, and electrical power of the system under the hybrid control method.

5.2 Solar radiation intensity

In this sub-section, the performance indices are carried out when the solar irradiation ranges from 200 W/m² to 800 W/m², ambient temperature is -10 °C, wind speed is 1 m/s, and mass flow rate ratio of injected vapor is 10%. This system could operate in one-stage mode (close expansion valve I) and two-stage mode.

Fig. 8(a) presents that the thermal efficiency rises from 41.49% to 46.99% but the electrical efficiency decreases linearly from 15.55% to 14.76% when the solar irradiation increases for the two-stage mode. Moreover, the overall efficiency increases from 82.41% to 85.83% with the increment of solar irradiation (from 200 W/m² to 800 W/m²). Higher solar irradiation would significantly increase the PV cells' temperature and then cause lower electrical efficiency. On the other hand, higher PV cells' temperature would benefit the thermal performance of the PVT module. The thermal efficiency outweighs the electrical efficiency, thus, it leads to an increasing trend of overall efficiency.

Fig. 8(b) shows that the system COP increases linearly from 2.30 to 6.32 with the increment of solar irradiation (from 200 W/m² to 800 W/m²) in two-stage mode. It could be found that the one-stage mode shows better system performance and higher evaporating temperature than two-stage mode when the solar irradiation exceeds 350 W/m² due to the adoption of the VI cycle. The vapor injection cycle would separate the mainstream of the refrigerant and a portion of the refrigerant would not absorb heat from the environment, and finally leads to a lower system COP. In this circumstance, the system should close expansion valve I to operate in one-stage mode for better performance when the solar irradiation exceeds than 350 W/m².

Fig. 8. Influence of solar radiation intensity on (a) electrical, thermal, and overall efficiencies; (b) system COP and evaporating temperature for one/two-stage modes.

Fig. 9 shows that the output power to grid would exceed 0 W when the solar irradiation is higher than 400 W/m² which means the system could operate off the grid. From this aspect, the system would consume electricity from the electrical grid to drive the compressor if the solar irradiation is below 400 W/m². The peak value of the output power to grid of the system is 1245.4 W, while the compressor power is 1016.3 W and the electrical power produced by PV cells is 2261.7 W. The heating power increases almost linearly from 2511.6 W to 8095.4 W with the increment of the solar irradiation (from 200 W/m² to 800 W/m²).

Fig. 9. Influence of solar radiation intensity on heating power, output power to grid, compressor power, and electrical power of the system under the hybrid control method.

The system could operate in three modes including parallel ASHP mode, two-stage mode, and one-stage mode through the control of valves. The solar radiation intensity, ambient temperature, COP, evaporating temperature variation curves under the hybrid control method at different times have shown in Fig. 10 for a typical winter day (January 18th) in a northern city (Lanzhou: 36.0°N/103.8°E). The hybrid control method is based on the environmental conditions, and different ambient temperatures and solar irradiation would influence the system's working mode. The parallel ASHP mode ensures the system could generate heat without solar irradiation. The peak value of

COP (6.09) occurs at 12:00 while the solar irradiation is 622 W/m², meanwhile, the evaporating temperature is -6.1 °C which is 5.8 °C higher than the ambient temperature. The proposed PVT based SAHP system with VI cycle operates in one-stage mode from 11:00 to 12:30 when the solar radiation intensity is high enough, and switch to the two-stage mode when the solar irradiation varies around 150~400 W/m². If the heat absorbed from the solar radiation could not support the running of the PVT SAHP cycle, then the system would operate in parallel ASHP mode and the system COP varies from 2.0~2.5 under different ambient temperatures (-16 °C~ -7°C).

Moreover, Fig. 11 illustrates the hybrid control method for the proposed system under different environmental conditions for better system performance. The system would switch the operation model by determining the value returned by the environment sensor.

Fig. 10. Solar radiation intensity, ambient temperature, COP, evaporating temperature variation curves under hybrid control method at different times on a typical winter day.

Fig. 11. Hybrid control method for the proposed system under different ambient temperatures and solar radiation intensity.

5.3 Mass flow rate ratio of injected vapor

In this sub-section, the performance indices are carried out under the working conditions: mass flow rate ratio of injected vapor varies from 5.0% to 15.0%, solar irradiation is 500 W/m^2 , ambient temperature is $-10 \text{ }^\circ\text{C}$, and wind speed is 1 m/s .

Fig. 12 shows the influence of the mass flow rate ratio of injected vapor on COP, overall efficiency, and mass flow rate of refrigerant. The system COP, mass flow rate of mainstream refrigerant, and the overall efficiency decrease while the mass flow rate of vapor injection refrigerant increases when the mass flow rate ratio of injected vapor increases. Under these specific conditions, a larger mass flow rate ratio of injected vapor means less mass flow rate in the mainstream and a larger mass flow rate in the injection bypass. Although larger injection flow would help to reduce the enthalpy of the mainstream refrigerant and increase the refrigeration effect at the PVT collector/evaporator inlet, the decrease of mainstream refrigerant's mass flow rate has a more adverse effect on the thermal absorption rate which means the heat extracted by the mainstream refrigerant in the PVT module would decline. On the other hand, the increasing total mass flow rate of refrigerant would cause higher compressor power. Consequently, a lower thermal absorption rate and higher compressor power result in a lower system COP. The maximum COP (4.30) occurs when the mass flow rate ratio of injected vapor is 5%, while the mainstream has the largest mass flow rate of refrigerant which is 45 g/s . The minimum COP (3.72) is obtained when the mass flow rate ratio of injected vapor is 15% while the mass flow rate of mainstream refrigerant is 42 g/s .

Fig. 12. Influence of mass flow rate ratio of injected vapor on COP, overall efficiency, and mass flow rate of refrigerant.

Fig. 13(a) shows that the evaporating pressure decreases smoothly when the mass flow rate ratio of injected vapor increases while the condensing temperature increases. The total compression ratio increases constantly from 4.46 to 5.34 when the mass flow rate ratio of injected vapor varies from 5% to 15%. The evaporating and condensing pressures are 0.48 Mpa and 2.57 Mpa when the mass flow rate ratio of injected vapor is 15%, respectively, which are 4% lower and 14.2% higher than that when the mass flow rate ratio of injected vapor is 5%.

Fig. 13(b) illustrates the effect of the mass flow rate ratio of injected vapor on evaporating

temperature, condensing temperature, evaporating-ambient temperature difference, and PV cells' temperature. The evaporating temperature reduces from -5.2°C to -6.6°C while the condensing temperature increases from 49.0°C to 54.9°C with the increase of the mass flow rate ratio of injected vapor (from 5% to 15%). The evaporating-ambient temperature difference decreases linearly from 4.8°C to 3.4°C while the PV cells' temperature increases from 2.3°C to 4.9°C .

Fig. 13. Influence of mass flow rate ratio of injected vapor on (a) evaporating pressure, condensing pressure, and total compression ratio; (b) evaporating temperature, condensing temperature, evaporating-ambient temperature difference, and PV cells' temperature.

Fig. 14 shows the influence of the mass flow rate ratio of injected vapor on output power to grid, compressor power, and electrical power. The output power to grid linearly decreases from 277.4 W to 114.0 W with the mass flow rate ratio of injected vapor. The minimum output power to grid is 114.0 W which occurs when the mass flow rate ratio of injected vapor is 15%, meanwhile, the compressor power (1387.5 W) reaches the peak value and the electrical power is 1501.6 W. A larger mass flow rate ratio of injected vapor would reduce the ability of the PVT collector/evaporator to harvest thermal energy from solar energy. Moreover, the heating power could be calculated by multiplying the compressor power by COP. The compressor power increases while the COP decreases with the increase of the mass flow rate ratio of injected vapor. However, the COP reduction is greater than the compressor power increase and then leads to lower heating power in this system. Therefore, the heating power reduces from 5331.2 W to 5160.6 W when the mass flow rate ratio of injected vapor increases from 5% to 15%.

Fig. 14. Influence of mass flow rate ratio of injected vapor on output power to grid, compressor power, and electrical power of the system.

5.4 System performance comparison

Fig. 15 presents the performance comparison of the COP between the proposed system and the ASHP system with VI cycle [44] under different ambient temperatures and the corresponding performance improvement when the solar irradiation is 500 W/m^2 . The proposed PVT SAHP system with VI cycle could operate more efficient than the ASHP system with VI cycle under different conditions. That is because the PVT collector/evaporator would obtain a higher evaporating temperature and a lower heat loss coefficient compared with the fin-tube evaporator due to its thermal insulation layer in the rear side of the panel. The ambient air is the low-temperature heat source during the evaporating process in the fin-tube evaporator, while the PV cells are the low-temperature heat source in the PVT module. The PV cells would reach a higher temperature which could easily exceeds the ambient temperature in the case of irradiation. Therefore, the evaporating temperature of the refrigerant in the PVT module would higher than that in the fin-tube evaporator, and then leads to higher COP of the proposed PVT SAHP system with VI cycle. Moreover, the proposed system is controlled by the hybrid control method for better system performance. In this case, the solar irradiation is 500 W/m^2 .

The performance improvement can reach 111.1% when the ambient temperature is $-20 \text{ }^\circ\text{C}$, in this circumstance, the COP of the proposed PVT SAHP system with VI cycle is 3.76 while it is 1.78 of the ASHP system with VI cycle. The performance improvement decreases rapidly from 111.1% to 63.6% with the increase of ambient temperature (from $-20 \text{ }^\circ\text{C}$ to $-10 \text{ }^\circ\text{C}$), and maintains around 63.5% when the ambient temperature ranges from $-10 \text{ }^\circ\text{C}$ to $0 \text{ }^\circ\text{C}$. The proposed system would switch two-stage mode to one-stage mode when the ambient temperature exceeds $1.5 \text{ }^\circ\text{C}$. The performance of the proposed PVT SAHP system with VI cycle is improved more obviously under cold environmental conditions. The highest COP of the proposed system is 6.26 at the ambient temperature of $10 \text{ }^\circ\text{C}$ while the COP of the ASHP system with VI cycle is 3.69. The average performance improvement in COP is 70.6% which reveals the wide application potential of the proposed system.

Fig. 15. Performance comparison of the COP of the proposed system and ASHP system with VI cycle under different ambient temperatures and the corresponding performance improvement.

The experimental parameters and COP results of the existing PVT SAHP system with VI cycle are obtained from Lu's [32] paper. The proposed system adopts a novel direct expansion evaporator with an optimized fluid channel which has higher thermal efficiency. As shown in Fig. 16, the COP of the proposed system is higher than the existing system under different cases. The lowest ambient temperature is -3.17 °C (Case 5) and the solar irradiation is 828.72 W/m², in this circumstance, the COP of the existing system is 3.75 while the COP of the proposed system is 4.38, and the performance improvement is 16.8% (lowest value). The maximum performance improvement is 42.2% which occurs in Case 2 when the solar irradiation is 300.14 W/m² and the ambient temperature is 1.89 °C. The average performance improvement is 33.3% of these six cases which reflects that the proposed system shows better performance compared with the existing system.

Fig. 16. Comparison results of the COP of the proposed system and the existing PVT SAHP system with VI cycle under different conditions and the corresponding performance improvement.

5.5 Economic analysis

The levelized cost of heat (LCOH) is an economic evaluation index to assess the costs of heat generated by heating systems. As a comprehensive economic assessment index, LCOH is appropriate to be adopted to compare the economic performance of different heating systems. In this regard, the LCOHs of four heating systems have been carried out to evaluate their economic performance: (a) proposed vapor-injection heat pump employing PVT collector/evaporator (PVT SAHP system with VI cycle); (b) air source heat pump employing VI cycle (ASHP system with VI cycle); (c) gas water heater (GWH); (d) electric water heater (EWH).

As shown in Fig. 17, the LCOHs of PVT SAHP system with VI cycle, ASHP system with VI cycle, GWH, and EWH are 0.054 \$/kWh, 0.097 \$/kWh, 0.104 \$/kWh, and 0.111 \$/kWh, respectively. The GWH and EWH are widely used for domestic hot water supply. Although the initial costs of these two systems are lower than the PVT SAHP system with VI cycle, the operating costs are much higher than those of PVT SAHP system with VI cycle. Therefore, the LCOH of PVT SAHP system with VI cycle (0.054 \$/kWh) is the lowest which is 48.5% lower than that of GWH and 51.5% lower than that of EWH, respectively. After comparing the LCOH, the proposed PVT based SAHP system with VI cycle shows better economic performance than the other three domestic hot water systems.

Fig. 17. Levelized cost of heat (LCOH) of four kinds of domestic hot water systems.

6 Conclusions

Performance improvement of vapor-injection heat pump system employing PVT collector/evaporator for residential heating in cold climate region was investigated. The detailed mathematical model was developed and verified to evaluate the system performance and theoretically reveal its merits. The main conclusions are as follows:

(1) The proposed system performs better than the ASHP system with VI cycle and the existing PVT SAHP system with VI cycle when ambient temperature ranges from -20 °C to 10 °C. For instance, the COP of the proposed system is 4.42 at the ambient temperature of 0 °C while the solar irradiation is 500 W/m², in the meantime, the COPs of the ASHP system with VI cycle and the existing PVT SAHP system with VI cycle are 2.70 and 3.57, respectively.

(2) Higher ambient temperature and solar irradiation would increase the overall efficiency and better for the system performance. The system could realize self-operation when the solar radiation

intensity exceeds 400 W/m², meanwhile, the system COP is 3.42 and the electrical, thermal, and overall efficiencies are 15.2%, 43.8%, and 83.9% when the ambient temperature is -10 °C respectively.

(3) The proposed system has better performance when the mass flow rate ratio of injected vapor is low. The maximum COP (4.30) occurs when the mass flow rate ratio of injected vapor is 5%, solar irradiation is 500 W/m², and ambient temperature is -10 °C. A hybrid control method including three modes has been proposed for better system performance.

(4) The levelized cost of heat of the proposed system is 0.054 \$/kWh which is 44.7% lower than that of air source heat pump employing VI cycle (0.097 \$/kWh), 48.5% lower than that of the gas water heater (0.104 \$/kWh), and 51.5% lower than that of the electric water heater (0.111 \$/kWh), respectively.

The mathematical model established in this paper could also be used to further theoretically assess the system performance under different working conditions.

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Nomenclature:

Abbreviations

<i>VI</i>	vapor injection
<i>PVT</i>	photovoltaic/thermal
<i>ASHP</i>	air source heat pump
<i>SAHP</i>	solar assisted heat pump
<i>COP</i>	coefficient of performance
<i>LCOH</i>	levelized cost of heat
<i>DX SAHP</i>	direct expansion solar assisted heat pump
<i>DXE</i>	direct expansion evaporator
<i>GWH</i>	gas water heater
<i>EWB</i>	electric water heater

Symbols

<i>A</i>	area (m ²)
<i>L</i>	length (m)
<i>W</i>	width (m)
<i>h</i>	enthalpy (J/kg)
<i>U</i>	heat loss coefficient (W/m ² ·K)
<i>T</i>	temperature (°C)/year
<i>I</i>	solar radiation intensity (W/m ²)
<i>Q</i>	heat transfer rate (W)
<i>P</i>	power (W)

p	pressure (Mpa)
m	mass flow rate (kg/s)
N	number (-)
R	thermal resistance (m·K/W)
V	volume (m ³)/voltage (V)

Greek symbols

δ	thickness (m)
τ	transmittance (-)
a	absorption ratios (-)
β	packing factor (-)
ε	emissivity (-)
η	efficiency (-)
ρ	density (kg/m ³)
γ	mass flow rate ratio of injected vapor (-)

Subscripts

pv	PV panels
e	electrical/evaporating
i	injected vapor/intermediate
c	PV-glazing cover/condensing
eva	evaporator
ref	refrigerant
th	thermal
$ideal$	ideal
res	flow resistance
abs	absorb
ove	overall
$cond$	condensation
com	compressor
cv	convection
cd	conduction
rd	radiation
rb	roll-bond panel
rc	reference
ei	electrical insulation
a	ambient
EVA	Ethylene Vinyl Acetate grease
L	lost
u	useful

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Declaration of Interest Statement

We declare that there are no conflicts of interest in the work we submitted.

Credit Author Statement

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