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2008

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Operating Characteristics of a Photovoltaic/Thermal Integrated Heat Pump System

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ABSTRACT

Hybrid photovoltaic/thermal (PV/T) collector, combining photovoltaic cells with a solar thermal collector, can simultaneously convert solar energy into electricity and heat energy. In this paper, a novel photovoltaic/thermal integrated heat pump (PV/T-HP) system was developed, in which the PV/T collector also acted as evaporator of heat pump. Two types of PV/T collectors with conventional copper evaporator tubes and multi-port flat extruded aluminum tubes were designed and investigated. Comparative results indicate that the latter structure obtains a better performance. Using the meteorological data in Nanjing, China, the simulation results based on 150 L water heating load show that such a PV/T-HP could simultaneously generate electricity and heat energy efficiently all year around. In addition, Optimizing operation by controlling the compressor speed on different weather conditions was proposed to achieve a better long-term performance.

1. INTRODUCTION

A hybrid photovoltaic/thermal (PV/T) collector, using a kind of fluid stream to remove heat generated in PV panel, can convert solar energy into electricity and heat simultaneously. It contains several advantages in contrast to separate PV and thermal systems, such as higher overall energy efficiency, lower cost, and smaller installation areas. Thus it has been increasingly attractive in solar utilizations.

In recent three decades, various PV/T collectors, commonly using air or water as heat exchange fluid, have been theoretically analyzed and experimental demonstrated. Modeling calculation and testing of mostly used water type sheet-and-tube PV/T collector were researched (Florschuetz, 1979; Chow, 2003; Huang et al., 2001). Comparative study on air type PV/T collectors with different configuration was carried out by Hegazy (2000). A comparison of seven different design types of PV/T collectors' efficiency and yield was evaluated to give guidance for system design (Zondag, et al., 2003). In addition, Ji et al. (2003) simulated a hybrid photovoltaic/thermal collector wall in residential buildings of Hong Kong. It is found that the fabric-integrated PV/T system achieved good electricity conversion and heat collecting efficiency, and meanwhile a fairy good performance on reducing space heat gain in summer. Up to now, several commercial parties has been beginning to accelerate the products development and demonstration (Thomas and Pierce, 2001).

However, existed PV/T systems mostly aim at obtaining electricity as much as possible, and overlook the heat energy output. The heat energy produced by conventional PV/T system is insufficient in quantity and temperature for building heat supply. As a matter of fact, the energy consumed for space heating and hot-water service accounts for a significant percentage of the total energy use in buildings. Therefore, it is essential for PV/T system to generate electricity efficiently at the same time to utilize heat effectively. Using heat pump technology, low-grade energy

from PV/T collector could be upgraded to an appropriate level for heating services. In this study, a novel PV/T integrated heat pump (PV/T-HP) system was proposed. Correspondingly, two types of PV/T collector suitable for this system were designed and comparatively investigated. Finally, system long-term operating characteristics were simulated and analyzed.

2. SYSTEM DESCRIPTION

Solar PV/T collector can be combined with heat pump in different types. This study will focus on the PV/T integrated heat pump (PV/T-HP) system as shown in Figure 1. The solar PV/T collector/evaporator connects PV-laminate with absorber plate of an uncovered sheet-and-tube collector. It also acts as evaporator of heat pump. Refrigerant expands directly in the evaporator tube adhered to the back of PV/T collector, and then released heat in condenser to heat water. It possesses several advantages: First, refrigerant evaporates in the PV/T collector keeping the PV module working on a uniform and relative lower temperature, and consequently improve the efficiency of electricity generation; Secondly, refrigerant directly expanded by absorbing solar irradiation in PV/T panel, and then released heat energy at a higher temperature in heat pump condenser, achieving a higher COP of heat pump as avoiding the internal heat exchange between PV/T collector and evaporator; Last but not least, the integrated configuration of PV module, solar thermal collector and evaporator reduces system installation areas and cost.

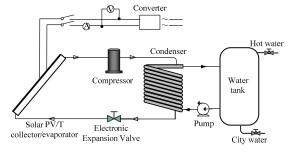


Figure 1: Schematic diagram of a PV/T integrated Heat Pump system

The prototype in this study is mainly composed of an uncovered PV/T collector/evaporator of 2 m² (1600×1250 mm), a variable-speed compressor with rated capacity of 370W, an electronic expansion valve (EEV), a water tank of 150 L connected to the tube-in-tube condenser by a pump, and other standard auxiliary components such as filter dryer, liquid receiver, converter, accumulator batteries. R22 was used as heat extracting fluid on the principle of reverse Rankine cycle. Prototype was designed working on a low temperature condition, namely collecting /evaporating temperature is lower than ambient air temperature. Thus, the PV/T collector obtains higher electrical and thermal efficiencies.

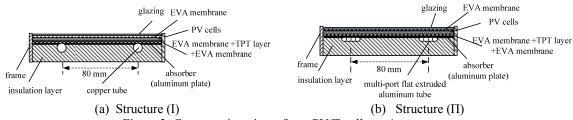


Figure 2: Cross-section view of two PV/T collector/evaporators

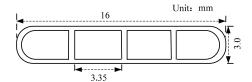


Figure 3: Dimensions of multi-port flat extruded tube

PV/T collector/evaporator pays an important role in a PV/T- HP system. Figure 2 showed two structures of sheet-and-tube PV/T collector/evaporator. For both structures, the PV module is the type of EVA encapsulated multi-crystalline silicon cells. Difference lies in the evaporator tubes. In the procedure of PV module encapsulation, aluminum plate was used as back plate instead of TPT materials in conventional PV module encapsulation. Glazing, EVA membrane, PV cells, EVA membrane + TPT layer + EVA membrane, absorber were stacked in order, and laminated together in laminator. Then evaporator tube with insulation layer was adhered to the back of absorber by using conductive glue. Finally, frame was fixed. In Structure (I) as shown in Figure 2(a), common copper tubes was used as evaporator tubes, while Structure (Π) shown in Figure 2(b) employs a kind of multi-port flat extruded aluminum tubes. Figure 3 shows dimensions of the latter evaporator tubes, which can be adhered to absorber more closely and easily because of a larger contact surface. This kind of multi-port flat extruded aluminum tube has been used in some automobile air-conditioning as condenser for its higher heat transfer performance. Moreover, collector of Structure (Π) is expected to obtain a high heat transfer performance as the refrigerant side heat transfer area is enlarged and contact heat transfer resistance between absorber an evaporator tubes is diminished.

3. SIMULATION OF PV/T-HP WITH TWO COLLECTOR STRUCTURES

3.1 Mathematical Model

A mathematical model for the PV/T-HP with two collector structures has been respectively developed and used subsequently for calculation procedure to predict their operating performances. Given that this study focus on the system performance at steady-state conditions, quasi-steady state modeling was used for all system components except for the storage tank where dynamic modeling was used.

3.1.1 Model of the PV/T collector/evaporator: As concluded by many researchers, performance of a photovoltaic module deteriorates with increase of its working temperature. This dependence of crystalline silicon cells' electrical efficiency (η_{el}) and thermal efficiency (η_{th}) on PV plate temperature (t_p) were typically given by Equation (1) and Equation (2) (Zondag, et al., 2003).

$$\eta_{el} = 0.097 - 0.00045(t_p - 25) \tag{1}$$

$$\eta_{th} = \alpha - \tau_{PV} \eta_{el} - U_L(t_p - t_a) / I \tag{2}$$

where heat loss coefficient, U_L , is mainly induced by the convective and radiation heat transfer between PV's surface glazing and ambient, which can be evaluated as Equation (3) to (5). α is the absorption coefficient of the absorber laminate, considered as 0.78, and τ_{PV} was considered as 1.0 according to experimental value obtained by Zondag et al. (2003).

$$U_{LC} = h_c + h_r \tag{3}$$

$$h_c = 2.8 + 3.0u_w \tag{4}$$

$$h_r = \varepsilon \sigma (T_p^2 + T_a^2)(T_p + T_a)$$
(5)

The solar energy collected by the collector/evaporator per unit time, Q_s , was calculated as follows:

$$Q_s = A_c \eta_{th} \tag{6}$$

(a) Heat transfer in evaporator of Structure (I): In the two-phase region, the local heat-transfer coefficient which was related to the refrigerant vapor quality, x, may be evaluated by:

$$h_{tp,1}(x) = \frac{3.0}{X_{tt}^{2/3}} h_{t.e} \tag{7}$$

where X_{tt} was Martinelli Number, $h_{l,e}$ was the local heat transfer coefficient based on liquid refrigerant-only in the two-phase region, and can be evaluated by the standard Dittus-Boelter correlation. Then the numerical integration method was used over the entire range of vapor quality inside the two-phase region to obtain the averaged heat transfer coefficient. Thus, the PV/T plate temperature could be calculated as Equation (8).

$$t_{p,1} = Q_s \left(\frac{1}{h_{tp,1} A_{in,1}} + \frac{\delta_1}{\lambda_1 A_{in,1}} + r_1' \right) + t_0$$
 (8)

where r'_1 is the contact heat transfer resistance between absorber plate and evaporator copper tube, t_0 is evaporating temperature of refrigerant.

(b) Heat transfer in evaporator of Structure (Π): Based on the experimental results of boiling heat transfer with different refrigerants in several small-diameter horizontal channels (Tran et al., 1996; Yu et al., 2002), boiling heat transfer coefficients could be correlated as a function of boiling number Bo, liquid Weber number We_l , and density ratio q_l/q_v as following:

$$h_{p,2} = 6400000 (B_o^2 W e_1)^{0.27} (\rho_l / \rho_v)^{-0.2}$$
(9)

where $B_o = \frac{q''}{Gi_{fg}}$, $We_l = \frac{G^2D}{\rho_l\sigma}$.

Thus, the PV/T plate temperature in Structure (b) could be calculated as Equation (10).

$$t_{p,2} = Q_s \left(\frac{1}{h_{tp,2} A_{in,2}} + \frac{\delta_2}{\lambda_2 A_{in,2}} + r_2' \right) + t_0$$
 (10)

where r'_2 is the contact heat resistance between absorber plate and multi-port flat extruded tube, which expected to be smaller than r'_1 of Structure (I).

3.1.2 Model of heat pump cycle: The refrigerant mass flow rate, m_r , was determined as following:

$$m_r = \frac{\eta_v V_h r}{60 v_1} \tag{11}$$

An energy balance in the collector/evaporator and condenser yields:

$$Q_e = m_r (h_1 - h_4) = Q_s (12)$$

$$Q_c = m_r (h_2 - h_3) (13)$$

Modeling of heat pump condenser was the same as previous study (Xu et al., 2006), which not presented in detail here.

3.1.3 Model of water heating cycle: Assuming that there was no heat loss from the water tank and pipe to ambient, the energy balance for the water tank was:

$$Q_c = m_w C_{p,w} \frac{dt_w}{dt} \tag{14}$$

3.2 Simulation and Comparison

Using the mathematical model above, the operating performance of PV/T-HP with two different collector structures has been simulated and compared. Assumptions in this simulation were: refrigerant at the exit of the collector /evaporator was saturated; the degree of refrigerant sub-cooling at the exit of condenser was 5 °C; solar irradiation and ambient temperature were constant at their respective average values throughout the simulated operational period, because this study focused on the average performance of the whole water-heating process; initial temperature of water in the tank was at ambient temperature. In addition, compressor speed was fixed at rated capacity.

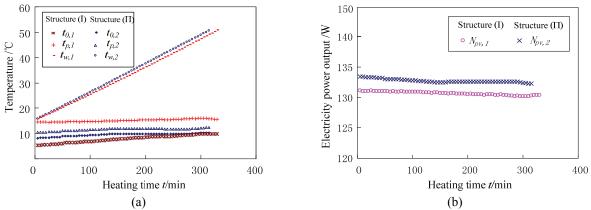


Figure 4: Variations of operating parameters with heating time for PV/T-HP with two structures

Figure 4 and Figure 5 shows the simulated operating performances on sunny days in spring or autumn, when t_a was assumed at 14.8 °C, and I at 645 W/m². Obviously, for PV/T-HP with both two structures, evaporating temperature t_0 slightly increase with the increase of water temperature t_w during the heating process, and heat pump coefficient of performance (COP) decrease. It can be seen that for PV/T-HP with collector of Structure (I), it would need 330 minutes to heat water from 14.8 °C to 50 °C, while 315 minutes was need for Structure (II). Evaporating temperature of collector Structure (II), $t_{0,I}$, with averaged value of 9.51 °C is higher than that of Structure (I) with averaged value of 7.86. Therefore it achieves a higher COP with averaged value of 4.81 than that of Structure (I) with averaged value of 4.56. Meanwhile, simulation results shows that the temperature difference between refrigerant and PV/T absorber for Structure (II), $\Delta t_{p,I}$ - $t_{p,I}$ - $t_{0,I}$, is about 2 °C. It is larger than that of Structure (I), $\Delta t_{p,2}$, which is about 8~10 °C. That means $t_{p,2}$ < $t_{p,I}$ resulting $N_{pv,2}$ > $N_{pv,I}$. It proves that PV/T collector with multi-port flat extruded aluminum tube obtains a better heat transfer performance and consequently improves both the electrical efficiency and thermal efficiency of system.

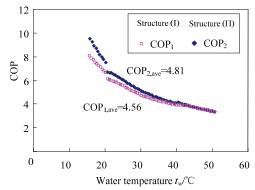


Figure 5: Variations of COP with water temperature for PV/T-HP with two structures

4. LONG-TERM OPERATING PERFORMANCE

The simulated monthly operating performances of the PV/T-HP prototype for generating electricity and heating 150L water up to 50 °C using the meteorological data in Nanjing, China, are shown in Figure 6. PV/T-HP works with highest COP of about 5.07 and highest electricity power output of 153 W in summer days, meanwhile with lowest COP of 4.03 and electricity power of 96 W in winter days.

Figure 7 indicates that the monthly mean evaporating temperature is about 10 °C lower than ambient air temperature of corresponding month. On this kind of operating condition, the heat loss of PV/T collector to ambient is slight and even may absorbs heat from ambient air. Although the thermal efficiency and electrical efficiency are improved, COP of heat pump is restricted which means more power would be consumed by heat pump. Figure 8 indicates that the monthly mean heating time needed is much larger than Sunshine Hours in winter, while is much shorter in summer. As a result, the PV/T-HP system with rated capacity couldn't produce sufficient heat energy to meet the required heat load in winter, on the contrary, system capacity exceed the required heat load in summer causing amounts of available solar energy was abandoned.

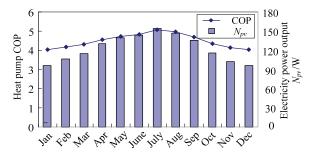


Figure 6: Simulated monthly mean performance of PV/T-HP in Nanjing, China

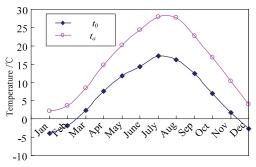


Figure 7: Simulated monthly mean evaporating temperature t_0 compared to ambient air temperature t_a

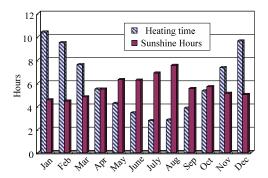


Figure 8: Simulated monthly mean heating time compared to Sunshine Hours

The conflict discussed above is caused by the mismatch between PV/T-HP system's capacity and weather conditions. As the PV/T-HP operated under a wide range of conditions, the load of collector/evaporator would fluctuate greatly, the use of fixed-capacity compressor was not advantageous. On the contrary, if the compressor speed was controlled to regulate the refrigerant flow rate at different weather conditions, the overall efficiency could be improved significantly. The optimizing operating method is generally concluded as following:

- When the solar radiation and ambient temperature is getting higher, such as in sunny summer days, the compressor speed should be reduced, leading to a lower mass flow rate of refrigerant and a higher t_0 . This in turn improved COP, and the energy consumption decreased. Also, the increase of t_0 resulted in a longer water heating time for make full use of the sunshine.
- When the solar radiation and ambient temperature is getting lower, such as in winter days or cloudy days, the compressor speed should be increased, leading to a higher mass flow rate of refrigerant and a lower t_0 . It can help to increase heat output to meet required heat load and shorten the water heating time, although COP of heat pump decreased.

5. CONCLUSIONS

Combination of solar PV/T collector with heat pump was a promising solar utilization technology. The uncovered sheet-and-tube PV/T collector with c-Si solar cells laminated on the surface and with refrigerant as working fluid is well-suited to be integrated with heat pump. Simulation of the PV/T-HP system with two collector structures indicate that collector using multi-port flat extruded aluminum tubes obtains a better performance compared to collector using conventional copper evaporator tubes.

The simulated long-term performance of PV/T-HP has demonstrated that this novel system can generate electricity and heat energy efficiently all year around. With the designed fixed-capacity of compressor, system operates on the condition of evaporating temperature lower than ambient air. Since PV/T-HP would be operated under a wide range of external operating conditions, the use of a compressor with variable capacity would be advantageous. A general method for optimizing operation by controlling compressor speed was discussed.

NOMENCLATURE

| A_c | Area of PV/T collector | m^2 | Subscri | ipts |
|--------------------------|--|--------------------|---------|--------------------|
| A_{in} | inside surface area evaporator tube | m^2 | a | ambient |
| Bo | boiling number | | c | condenser |
| COP | coefficient of operating performance | | e | evaporating |
| C_p | specific heat of water | J/(kg·K) | el | electrical |
| \vec{G} | mass velocity of refrigerant | $kg/(m^2 \cdot s)$ | in | inlet |
| D | tube diameter | m | l | liquid |
| h | heat transfer coefficient | W/m^2K | PV | photovoltaic cells |
| | /refrigerant enthalpy | J/kg | p | collector plate |
| i_{fg} | latent heat of refrigerant | J/kg | r | refrigerant |
| h_c | convective heat loss coefficient | W/m^2K | S | solar |
| h_r | radiation heat loss coefficient | W/m^2K | th | thermal |
| I | solar irradiation | W/m^2 | tp | two-phase region |
| N | electricity power | W | v | vapor |
| m_w | mass flow in tank | kg | w | water |
| N_{PV} | electrical output of PV cells | W/m^2 | 0 | evaporating |
| Q | heat energy | W | 1 | Structure 1 |
| q'' | heat flux | W/m^2 | 2 | Structure 2 |
| r_1' | contact heat transfer resistance | $(m^2 \cdot K)/W$ | 1,2,3,4 | state point in the |
| t | temperature | $^{\circ}$ C | | heat pump cycle |
| U_L | heat loss coefficient | W/m^2K | | |
| u_w | wind speed | m/s | | |
| V_h | theoretic displacement of compressor | m^3 | | |
| v | specific volume | m ³ /kg | | |
| We_l | liquid Weber number | | | |
| X_{tt} | Martinelli Number | | | |
| x | vapor quality | | | |
| Greek letters | | | | |
| α | absorption coefficient of the absorber | | | |
| δ | tube thickness | m | | |
| τ | transmission coefficient | | | |
| η | efficiency | | | |
| $\stackrel{\cdot}{\rho}$ | density | kg/m ³ | | |
| λ | thermal conductivity | W/(m·K) | | |
| σ | surface tension | N/m | | |
| | | | | |

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ACKNOWLEDGEMENT

This research was supported by The National High Technology Research and Development Program of China (863 Program), Excellent Doctoral Dissertation Foundation of Southeast University, China, and The Hong Kong Polytechnic University.