An Experimental Thermal Performance Analysis & Comparison of a Direct Expansion Solar Assisted Heat Pump Water Heater With Unglazed and Single Glazed Collector

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Abstract
An experimental thermal performance analysis of a direct expansion solar assisted heat pump for domestic water heating is presented. This system uses flat plate collector as an evaporator with a surface area of 2.23m², a small R134a reciprocating-type compressor with rated input power 245 W, an insulated hot water tank having a volume of 30L and capillary expansion tube having a bore of 0.036". For water heating the energy sources are mostly liquefied petroleum gas and electricity. The use of heat pump or solar water heating, particularly the solar-assisted heat pump options, is not popular. In this paper, the potential application of direct-expansion solar-assisted heat pump (DE-SAHP) system was examined. Experimental studies measure the solar intensity, water inlet temperature, water outlet temperature, ambient temperature, time step, and heat gain at the condenser for finding out the coefficient of performance, also a performance comparison between glazed and unglazed collector was done. The COP for the DE-SAHP built in the present study lies in the range 3–7 depending on operating conditions.

Keywords

I	Solar radiation intensity (W/m²)
T_a	Ambient air temperature (°C)
T_w	Temperature of water in the storage tank (°C)
m_r	Mass flow rate of the refrigerant (kg/s)
h	Specific Enthalpy (kJ/kg)
c_{pw}	Specific heat of water (kJ/kgK)
M_w	Mass of the water in water tank (kg)
P_s	Pressure at suction (kg/cm²)
P_c	Pressure at condenser (kg/cm²)
t	Time (min)
Q_w	Heat gain at condenser

I. Introduction
Hot water and steam form an integral part of various industrial and commercial applications and with rising oil prices, there has never been a better time to look at heating water by harnessing energy from the sun. Hot water can be used in number of applications such as large scale hot water usage for bathing and laundry applications in hostels, hospitals and high rise apartment buildings, pre feed for boilers for steam generation, for various industrial applications, milk dairies for applications such as pasteurization, condensation, cleaning, leather processing industry for drying and tanning. Hot water also used in metal finishing industry for degreasing and phosphating, resin emulsification in polymer industry, drying and related processes in pharmaceutical industry, solar drying through air heating is an area of growing interest and swimming pool heating is a popular concept in India and abroad. In the direct expansion solar assisted heat pump water heater (direct SAHP) system, the solar collector serves as an evaporator where the refrigerant absorbs the incident solar energy and energy rejected by the condenser contributes to water heating. Since the solar collection system can supply energy at temperatures higher than the ambient outdoor air, the capacity and coefficient of performance improves. In virtue of its above mentioned advantages, the direct SAHP is expected to have a huge potential market in daily life. In the direct SAHP, condenser can be arranged as an external heat exchanger supplying a hot water or arranged as an immersed coil in the hot water storage tank. The main advantage of direct SAHP as compared to SAHP is the better thermal performance due to direct expansion and evaporation of the refrigerant in the collector. In direct SAHP, long supply and return lines are required between roof mounted solar collector and the thermal storage located in the heated interior of the building. As these lines are charged with refrigerant and require specialized mechanics, this makes the installation difficult. The study however concluded that this system configuration offered significant performance and cost benefits over the solar assisted heat pump water heater. In this paper, we will discuss and compare the performance of direct expansion solar assisted heat pump with and without glazing.

II. Literature Survey
The concept of the DESAHP was firstly considered by Sporn and Ambrose in 1955 [1]. The DE-SAHP principle is one of the most promising techniques, so much research has focused on DE-SAHP systems, including system structure ,thermal performance, working fluid characteristics , operational control, numerical simulation, economic analysis ,etc. since the overall performance of a solar system is influenced significantly by the changes in the climate conditions and load demands, the real system matching in a whole year is hardly realized without the guide of a reasonable theoretical analysis [2].
Chaturvedi (1980) [3] carried out a theoretical analysis for the instantaneous operation of a SAHP and shows that the evaporating temperature of a SAHP, Te, depends on the solar radiation I and the ambient temperature Ta. Te may be higher than Ta (heat dissipated to the ambient air from the collector) or lower than Ta (heat gained from the ambient air), depending upon the design and the operating conditions.
Chaturvedi (1998) [4] further shows theoretical and experimentally that a SAHP using a bare collector and a variable-frequency compressor has an optimum performance provided the collector temperature (related to the evaporating temperature Te) is maintained in a temperature range of 5–108oC above ambient. Ito et al. (1999) also designed a SAHP with a bare collector operating at Te-Ta.
Theoretical and experimental studies were made by ito, miura and wang [5] on the thermal performance of a heat pump that used a bare flat plate collector as the evaporator. The analysis used empirical equations to express the electric power consumption of the compressor and Coefficient of Performance (COP), as functions of temperature of evaporation at the evaporator and that

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of the heat transfer medium (water) at the inlet of the condenser. The model having been established, the influence of the collector area, the pitch of the tube for the refrigerant flow, and the collector plate material were also investigated analytically.

It should be noted that refrigerant R-12 R-22 was used in those experiments. On the other hand, the analysis of the paper can be applied in the same way to heat pumps using other kinds of refrigerants. Our experimental analysis is based on R-134a.

II. System Description

Fig. 1 shows the schematic diagram of the DE-SAHPWH system. The system consists of detachable glazing flat plate solar collector which acts as an evaporator, a hot water storage tank with an immersed heat exchanger as condenser, a capillary expansion tube and a small hermetically sealed reciprocating compressor. The figure also shows the pressure gauges and temperature probes as and where located in the system.

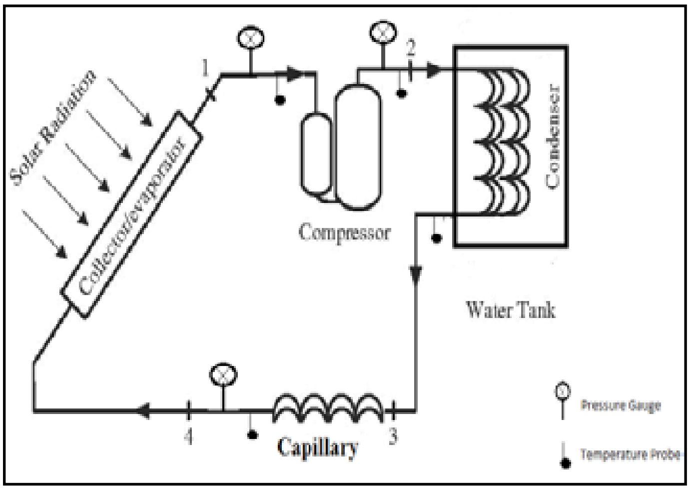

Fig. 1: Schematic Diagram

A. Flat Plate Solar Collector/Evaporator

The DE-SAHPWH uses a detachable glazing flat plate solar collector with total area 2.23m². It consists of a copper absorbing plate of 0.3mm thickness. Copper tubes of OD 5/16 inches soldered to the plate with a pitch of 100mm. Collector has an aluminium sheet at the back, insulation in between with aluminium foil wrapped over it.

B. Compressor

A small R-134a reciprocating type hermetically sealed compressor with piston swept volume 5.79cc and rated input power 245W is adopted. The compressor has the self-protection circuit to prevent coil overheating and over current.

C. Condenser/Hot Water Storage Tank

The condenser is made up of a copper tube OD 3/8” with length 32ft, which is immersed in the domestic hot water tank. In order to improve the system performance factors such as heat conduction within the tank as well as side losses are considered and the storage tank is thus designed to minimize such losses. The water tank has a capacity of 30 liters.

D. Capillary Tube

The expansion device used in the system is a capillary tube of 0.036 inches OD and length of 10ft. Two parallel 5ft capillaries are used with a copper filter. Use of capillary has an advantage that, when the compressor stops the refrigerant continues to flow into the evaporator and equalizes the pressure between the high sides and low side of the system this considerably decreases the starting load on the compressor. Thus a low starting torque motor can be used to derive the compressor, which is a great advantage. Also the refrigerant charge in the capillary tube section is critical, therefore no receiver is necessary.

The photograph of the setup is shown in the fig. 2.

Fig. 2: Experimental Setup

III. Experimental Procedure

In order to investigate the various parameters of DE-SAHPWH, tests were performed in two stages. In the first stage, an unglazed DE-SAHPWH was experimentally investigated and all the data was recorded. In the second stage a single glazed DE-SAHPWH was experimentally investigated and all the data was recorded. Data was recorded after steady state condition was established in the system. Experimental test were performed on different days, in different solar intensity and with different ambient temperatures. The readings are recorded at the interval of 10-15 minutes during the day time in the month of May 2013.

Ambient temperature, solar radiation, condenser water temperature, refrigerant temperatures and pressures at various points were recorded. In order to avoid damage to the compressor, the experiment was stopped as soon as the condenser water temperature reached about 50oC. The test conditions for the various experiments conducted without water draw off from the storage tank.

Energy balance with the immersed condenser yields

$$Q_w = M_w \, c_{pw} \, (dT_w/d_t) \qquad (1)$$

where Q_w is the heat gain at the condenser, which is also the heat transfer rate released to the hot water in tank by the condenser. M_w is the total mass of the water in domestic water tank, c_{pw} is the specific heat of the water, Tw is the temperature of the water, t is the time.

The Coefficient of performance is defined as ratio of heat gain (Q_w) in the condenser to the compressor electric power consumption. Compressor consumption is found energy-meter reading.

Experimentally evaporative load can be calculated as

$$Q_e = m_r(h_1 - h_4) \qquad (2)$$

Calculation of COP:

$$COP = Q_w/W_{cm} \qquad (3)$$

Also, $$COP = 1 + Q_e/W_{cm} \qquad (4)$$

IV. Results and Discussion

For the purpose of analysis and comparison, two best readings for both stages were selected.

In case of unglazed evaporator many experimental tests were conducted from 16/05/2013 to 20/05/2013. Data was recorded for all that tests, but out of the recorded data a single test data has been taken in which heat pump is giving best performance using unglazed evaporator. On 18/05/2013 heat pump gave best performance between 10:10 am to 11:22 am where intial water temperature was 29°C (Table 1).

Similarly in case of glazed evaporator many experimental tests were conducted from 22/05/2013 to 26/05/2013. Data was recorded for all that tests. A single test data has been taken in which heat pump is giving best performance using glazed evaporator. On 24/05/2013 heat pump gave best performance between 10:10 am to 11:18 am where initial water temperature was 29°C (Table 2).

Table 1. Experimental Results With Unglazed Collector

I (W/m²)	t (min)	Ta (°C)	Tw (°C)	PC Kg/cm²	PS Kg/cm²	Qw (kWh)	Wcm (kWh)
748	10	42	30	8.9	1.2	0.034	0.030
773	20	42	32	9.6	1.6	0.104	0.061
805	30	42	34	10.2	1.8	0.174	0.091
828	40	43	37	11.0	2.1	0.278	0.122
835	50	43	40	12.0	2.3	0.383	0.152
856	60	43	45	12.6	2.5	0.557	0.183
879	70	43	48	13.8	3.2	0.662	0.213
885	72	44	49	14.0	3.5	0.697	0.220

Table 2: Experimental Results With Glazing

I (W/m²)	T (min)	Ta (°C)	Tw (°C)	PC Kg/cm²	PS Kg/cm²	Qw (kWh)	Wcm (kWh)
751	08	43	31	9.2	1.6	0.069	0.028
775	15	43	34	10.4	1.9	0.174	0.056
809	24	43	37	11.0	2.2	0.278	0.085
830	34	44	40	11.5	2.5	0.383	0.113
834	40	44	42	12.2	2.8	0.453	0.142
858	48	44	45	12.9	3.2	0.557	0.170
880	56	44	48	13.6	3.4	0.662	0.199
887	62	44	50	14.2	3.6	0.732	0.220

Variation of the evaporation temperature with the tank water temperature

Fig. 3 shows the variation of the evaporation temperature with respect to the tank temperature. It is evident from the figure that when the evaporation temperature increases the tank water temperature (condenser) of the DESAHP cycle increases.

This means that the evaporation in the DESAHP cycle takes place at a higher temperature if the temperature of the heat source is higher. The change is almost linear. The temperature of the evaporator is not allowed to increase to higher values since the compressor can be damaged by exceeding its operation limits. The performance characteristics of the system have been investigated in this range. In case of unglazed evaporator the evaporation temperature could not exceed than 35 °C which is below ambient temperature, but in case of glazed evaporator the evaporation temperature exceeded more than ambient temperature 47 °C and hence better performance of DESAHP was achieved.

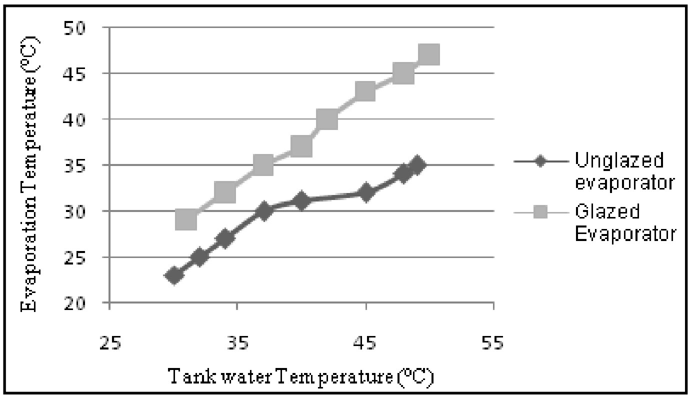

Fig. 3: Variation of the Evaporation Temperature With the Tank Water Temperature

Effect of evaporation temperature on Actual COP values fig. 4 shows the variation of the Actual COP value of the DESAHP with the evaporation temperature. When the solar radiation increases, the evaporation temperature increases. The amount of heat gain at condenser, i.e. Qw, increases with increase in evaporation temperature. Thus, the COP of the heat pump increases with increasing evaporation temperature. The maximum COP value of heat pump in case of unglazed evaporator reaches 3.16 and in case of glazed evaporator the maximum COP values reaches 3.32. Hence it is shown from figure that heat pump with glazed evaporator gives higher COP values than unglazed evaporator.

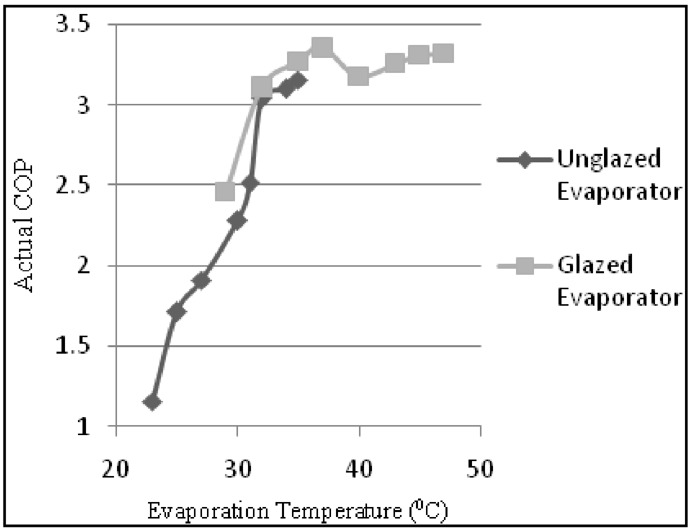

Fig. 4: Effect of Evaporation Temperature on Actual COP Values

Variation of Energy Consumption with heat gain at condenser

Fig 5 compares the Energy Consumption with Heat gained by water in condenser for a glazed and unglazed DESAHP. It is evident from the figure that for any particular value of heat gain (Qw), unglazed DESAHP consumes more energy than glazed DESAHP. Hence glazed DESAHP performs better in comparison to unglazed. This variation will be more visible when system is operated in winter conditions.

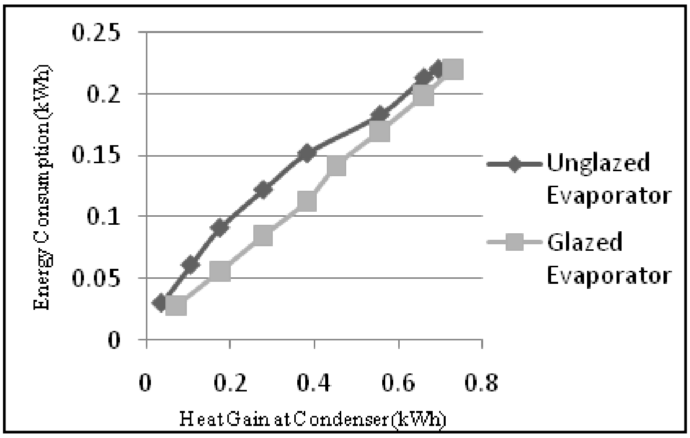

Fig. 5: Variation of Energy Consumption With Heat Gain at Condenser

V. Conclusion

A Direct Expansion Solar Assisted Heat Pump Water Heater using a glazed and unglazed collector has been investigated for hot water application. Experimental results show that there is increase in the COP of the glazed DESAHP in comparison to unglazed DESAHP. Experiments were made between 29-50°C tank temperatures and a performance analysis of the system was carried according to the experimental results. The COP's of DESAHP were calculated. The results showed that in the unglazed DESAHP 30 litres of water could be heated from 29°C to 49°C in about 72 minutes and energy consumption is about 0.22kWh whereas in glazed DESAHP 30 litres of water could be heated from 29°C to 50°C in about 62 minutes and energy consumption is 0.22kWh.

The glazed DESAHP system improves the COP of the unglazed DESAHP from 3.16 to 3.32, Energy consumption remains the same. The experimental investigation also verified that for any particular value of heat gain at condenser (Qw), unglazed consumes more energy than glazed DESAHP.

The efficiency of the DESAHP is measured by the coefficient of performance. A high coefficient of performance implies less electrical energy input for a given thermal energy output of the heat pump. A 100% efficient electrical heater would have a COP of 1 i.e. heat input equal heat output. It can be seen that DESAHP system has an advantage over electrical resistance heating. For a DESAHP operating with COP of 3, it is somewhat descriptive to say that one unit of electrical energy input results in delivery of three units of thermal energy by the heat pump. Hence this system can completely replace conventional water heating systems in near future.

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