PERFORMANCE ANALYSIS OF A SOLAR-ASSISTED DUAL-TANK HEAT PUMP SYSTEM FOR CLIMATIC CONDITIONS IN TURKEY

by

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This study aimed to analyze the performance of a solar-assisted dual-tank heat pump system under climatic conditions in Turkey. This system and its components were modeled and simulated using transient system simulation software. The system was designed not only to supply domestic hot water for a restaurant, but also to heat it in winter and cool it in summer. The modeled system works on the principle that a water-to-water heat pump operating between dual tanks transfers the heat from the cold water tank (source side) to the hot water tank (load side). The hot water for both heating and domestic hot water is supplied from the hot water tank throughout all seasons, whereas, the cold water is supplied from the cold water tank for cooling the space in summer. A photovoltaic thermal collector was integrated into the cold water tank to support the source side of the heat pump and also to generate electricity for the system in winter, but was used only for producing electricity in summer. Analyses were carried out for five provinces (Istanbul, Ankara, Izmir, Hakkari, and Trabzon) located in five different regions of Turkey. According to the simulation results, the highest seasonal performance factor (2.65) was obtained for Izmir, whereas the lowest seasonal performance factor value (1.74) was obtained for Hakkari. The system worked 52% more efficiently in Izmir than in Hakkari. With the photovoltaic thermal collector, 17.68% of the total electrical energy consumption of the system was compensated for Izmir Province and 12.09% for Hakkari Province.

Key words: *photovoltaic thermal, transient system simulation, heat pump, simulation, solar-assisted, dual-tank*

Introduction

Solar energy systems play a notable role in meeting the energy requirements not only for supplying domestic hot water (DHW), but also for heating and cooling buildings. These systems have been integrated with heat pump systems to improve their efficiency.

Many studies on solar-assisted heat pump (SAHP) systems are available in the literature. These heat pump systems assist a solar collector in extracting additional energy from the solar loop, thus improving the solar fraction and reducing the source energy consumption. A solar thermal collector enhances the heat pump performance by raising the evaporator inlet temperature [1]. Higher collector efficiency can be achieved by working at a low temperature range and decreasing the heat loss from the collector to the surroundings [2]. The SAHP system can achieve a higher performance than a conventional-type air-source heat pump [3]. These sys-

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tems can be classified as direct or indirect SAHP systems according to how the solar energy is supplied from the collector, *i.e.*, by means of a flat plate solar collector or a photovoltaic thermal (PV/T) solar collector. To meet space requirements, it is enough to integrate only one tank into these systems for supplying heat and/or DHW. However, for the cooling of a space in addition heating and DHW requirements, one more tank should be integrated into the system. Many experimental and numerical studies on SAHP systems can be found in the literature. Most of the numerical studies have been conducted using transient system simulation (TRNSYS) software. Some of these studies are briefly included here. Using TRNSYS, Sterling and Collins [4] compared a dual-tank indirect SAHP system (SADHP) with traditional solar and electric DHW systems. Sterling [5] carried out a feasibility study with two indirect SAHP systems for DHW. Wagar [6] simulated and validated results from a single-tank SAHP system for DHW. Banister *et al.* [7] simulated and modeled a single-tank SAHP system using TRNSYS, and then validated the average temperature of the DHW tank using an experimental test set-up. Zahrani *et al.* [8] also applied TRNSYS to investigate using a heat pump for both heating and cooling. Chu [9] used TRNSYS to design and construct a SADHP system for a high performance house in order to investigate the energy consumption of the system, which was used to provide DHW, space heating, space cooling, and dehumidification under Canadian climatic conditions. The SADHP system was also designed and constructed by Chu *et al.* [10]. Moreover, Banister and Collins [1] developed a novel SADHP system configuration for DHW and space heating. Simulation results obtained via TRNSYS were validated experimentally by using a test apparatus. Croci *et al.* [11] also developed a SADHP, validated it, and compared it with conventional systems. Li *et al.* [12] compared an electrical water heating system with a conventional solar DHW system, a single-tank solar combi system, and a dual-tank solar combi system and assessed them from technical and economic aspects. The corresponding parameters of TRNSYS modules were also tested and validated using experimental data. Shrivastava *et al.* [13] investigated modelling and simulation of solar water heaters from a TRNSYS perspective. The heat pump with PV/T collectors with storage tanks potentially enable the development of solar energy systems that are more efficient [14], and therefore, in recent years, many researchers have been working on PV/T-integrated heat pump systems. Zang *et al.* [15] designed and fabricated a novel solar photovoltaic/loop-heat-pipe module-based heat pump system for both electricity and DHW. Tsai [16] presented a novel model for a refrigerant-based PV/T-assisted heat pump water heater system, in which the PVT evaporator simultaneously provided solar electricity and thermal energy to assist the system. The model accuracy was validated through experimental measurement with a test rig under real climate conditions. Zhang *et al.* [17] presented a simulation model of a PV/T system, as well as the simulation and experimental test results. The results were compared with numerical simulation data obtained via TRNSYS. Bellos *et al.* [18], also using TRNSYS, designed, simulated, and evaluated in energetic and financial terms four SAHP in order to detect the most suitable heating solution. Vallati *et al.* [19] investigated the potentiality of an energy system equipped with PV/T hybrid solar collectors, storage tanks, and a heat pump for the space heating of a small office for three different European city locations. Barbu *et al.* [20] analyzed the impact of the variation of some thermal parameters of a DHW tank (cold water main temperature, tank size, tank outlet flow, and consumer demand curve) on the electrical efficiency of a PV/T panel.

As can be seen from the literature studies reviewed here, applications of the PV/T-integrated SAHP systems have been developing rapidly. This study was conducted in order to encourage the widespread use of such renewable energy systems and to guide those working in this field. The single-tank SAHP system is currently used only for heating and domestic water in Turkey, however, unfortunately, for space cooling, an extra air conditioning system is needed. The SADHP system presented in this study is a novel system for our country because it meets cooling requirements in addition those of heating and DHW. Therefore, using TRNSYS, a SADHP system was designed and simulated to meet the space heating and cooling and DHW needs of a restaurant under climatic conditions of five provinces in Turkey. A performance analysis of the system was then carried out. In this study, all technical data for restaurants such as heat gains, losses, and schedules were taken from Coskun [21].

Description of system and simulation method

As seen in fig. 1, the modeled system consists of a water-to-water heat pump operating between dual tanks. The nominal heating capacity and nominal heating power are 21.7 kW and 5 kW, respectively. It is used to extract heat from the cold water tank (CWT) and transfer heat to the hot water tank (HWT), which is used for meeting the daily DHW consumption and, using a heating pump, for heating the restaurant during the winter months. At a constant temperature of 55 °C, 300 L of water is drawn from the HWT for 15 minutes four times every day (at 06:00 a. m., 08:00 a. m., 20:00 p. m., and 22:00 p. m.).

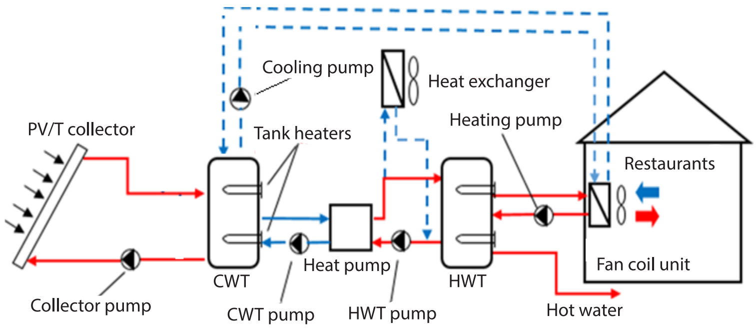

Figure 1. The SADHP system [22]

The restaurant, whose architectural plan is shown in fig. 2, has a volume of 337.5 m^3 and is open every day between 08:00 a. m. and 22:00 p. m. The room temperature is maintained at 21 ±1 °C in winter and 24 ±1 °C in summer, and the water temperature in the HWT is maintained at 55 °C. Detailed technical data can be obtained from Coskun [21].

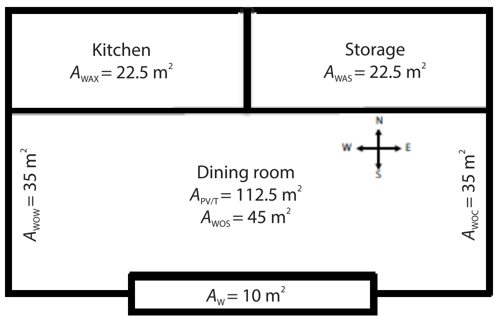

Figure 2. Architectural plan of restaurant [21]

By adding a PV/T collector to the system, the heat pump source temperature is increased in winter and some of the consumed electrical energy is met. This is accomplished by circulating water between the CWT and the PV/T collector using a collector pump. For cooling the restaurant in summer, the cold water in the CWT is pumped to the fan-coil unit in the restaurant via the cooling pump. In summer, the PV/T collector is used only for electricity generation. In summer, as in winter, the DHW for the restaurant is provided from the HWT. However, unlike in the winter months, in the summer months, the water temperature in the HWT increases only when domestic water is drawn from it. A heat exchanger (HEX) was added to the system in order to keep the condensing temperature and pressure values at the desired levels.

The SADHP system was modeled and simulated using TRNSYS, which is a graphically based software for simulating transient systems such as energy systems. It is primarily an equation-solving program based on standard numerical techniques. This program consists of two main parts. One is an engine that reads and processes the input file, iteratively solves the systems, determines convergence, and plots system variables. The other part is an extensive library of components, with each modelling the performance of one part of the system. These are the engineering models that must accurately reflect the true physics of the system being modeled. Many studies in the literature have focused on the validations of these models. The TRNSYS model of the system created in this study is shown in fig. 3 and the components and specifications used in the model are presented in tab. 1.

During the simulation process, the room temperature is controlled by the heating thermostat in the winter months and by the cooling thermostat in the summer months. Room temperature was set at 21 °C ±0.5 in the winter season and at 24 °C ±0.5 in the summer season. The heat pump and HWT and CWT pumps are activated when any of the heating or cooling signals is greater than 0. Whenever the heating signal is active, both the heating fan and the heating pump are activated, whenever the cooling signal is active, both the cooling pump and the cooling fan are activated. In winter, the heat pump source temperature drops due to heat withdrawal. In order to prevent the CWT temperature from falling below a specified value, two 5 kW auxiliary heaters are placed in the 2nd and 8th layers of the CWT and controlled by thermostats. The heater in the 2nd layer turns on when the water temperature in this layer drops to 10 °C and turns off at 15 °C, whereas the heater in the 8th layer turns on when the water temperature in this layer drops to –10 °C and turns off at 5 °C. The dead band temperature value for both heaters was chosen as 5 °C. In order to prevent freezing, 30% ethylene glycol was added to the water circulating between the CWT and the PV/T collector. In addition, by using the PV/T collector system during the winter months, the water temperature in the CWT is increased by using solar energy, and some of the electricity produced is used. Water between the CWT and the collector is circulated by the collector pump. The collector pump is activated when the heating signal is active and the collector outlet temperature is greater than the collector inlet temperature, whereas it is not activated in summer when the cooling signal is active.

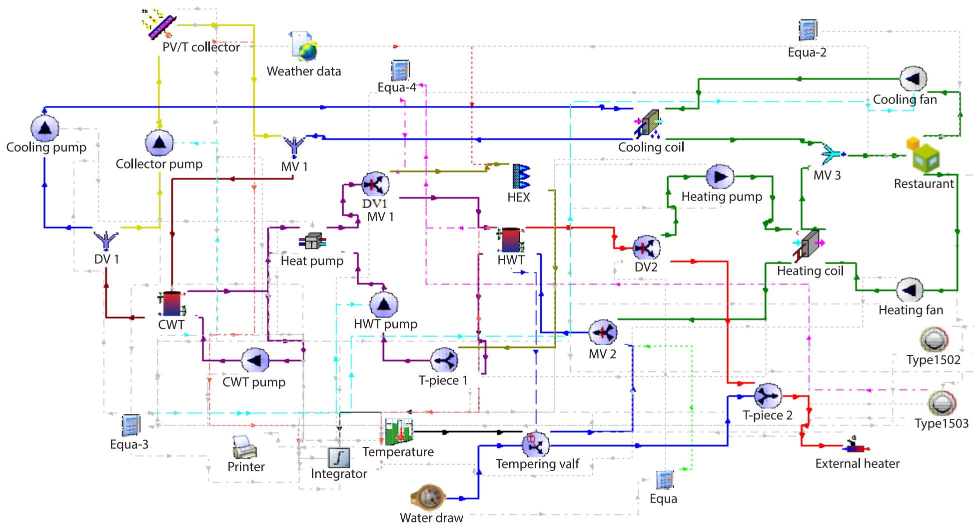

Figure 3. The TRNSYS model of the system

Table 1. Components and specifications used in the TRNSYS model

U-values of Building (Type 56)					
Roof, wall (outside), wall (inside), fllor [$Wm^{-2}K^{-1}$] (respectively)				0.453, 050, 0.497	
Heat pump (Type 927)		PV/T coolector (Type 563)		Fans (Cooling/Heating) (Type 112b)	
Rated heating capacity	21.73	Collector area [m^2]	13, 26	Maximum flow rate [kgh^{-1}]	6000
Rated heating power [kW]	5.05	Collector slope [°]	45	Power consumption [W]	900
Rated cooling capacity [kW]	15.28	Specific heat of fluid [$kJkg^{-1}K^{-1}$]	3.73		
Rated cooling power [kW]	3.89	Reflectance/ Emissivity [–]	0, 15/0, 9		
	Cold water tank (Type 4c)	Hot water tank (Type 4c)		Auxilary heaer (Type 6)/ Auxilary cooler (Type 1246)	
Tank volume [m^3]	0, 35	0, 50		Maximum heating capacity [°C]	5/40
Nuber of node [–]	10	10		Set point temperature [°C]	55/30
Number of auxilary	2	2			
Auxilary heater maximum heat-ing rate [kW]	5	2			
Collector pump (Type 3b)/ [CWT, HWT tank, heating/ cooling pumps (Type 3d)]		Cooling coil (Type 508c)		Heating coil (Type 670)	
Maximum flow rate [kgh^{-1}]	500/1440	Set point temperature [°C]	10	Set point temperature [°C]	35
Power consumption	40/60	Coil bypass fraction [–]	0, 15	Effectiveness [–]	0, 5

In winter, when the water temperature in the 2nd layer of the HWT drops below 50 °C or the water temperature in the 8th layer drops below 25 °C, two 2 kW electric heaters are activated in order to raise the water temperature. The water outlet temperature in the HWT is maintained at 55 °C. Hot water is directed to the heating coil for space heating and to the *T*-direction valve (*T*-piece 2) for water drawing via the controlled flow diverter valve (DV2). If there is no draft, hot water is directed directly to the heating coil, whereas if there is a draft, the signal to the DV2 valve opens the way to *T*-piece 2. In the meantime, the MTV valve is opened to feed the same amount of water to the tank as the hot water that was withdrawn. Instead of hot water drawn from the top layer of the tank, cold water at 15 °C is directed to the MTV valve to be sent to the bottom layer of the tank by the tempering valve. If the tank outlet water temperature is above 55 °C, the cold water coming from the tempering valve is mixed by *T*-piece 2. If the water temperature is below 55 °C at the *T*-piece 2 outlet, it is sent to the external heater to be heated and the domestic water temperature is raised to 55 °C.

In the summer months, because only DHW is drawn from the HWT, the water temperature in the tank can rise above 50 °C and as a result, the heat pump load temperature can exceed the nominal limits. To prevent this, a controlled flow diverter valve 1 (DV1) is placed between the heat pump and the HWT. The DV1 valve is activated when the cooling signal is greater than zero and the load temperature of the HWT is greater than 50 °C. When these conditions are met, the DV1 valve sends the water leaving the heat pump to the HEX instead of to the HWT for cooling. The designed system was simulated for one year (in the range of 0-8760 hours), with the simulation step interval taken as 0.125 hour. The daily operating time interval of the system was determined as between 08:00 a. m. and 22:00 p. m.

Control scenarios during all operation were:

– cooling or heating signal > 0, then all – CWT pump, HWT pump, and heat pump are activated,
– cooling signal > 0, then all – heat pump, cooling pump, and cooling fan are activated, whereas collector pump is deactivated,
– cooling signal >0 and $T_{\mathrm{HWT,load}} > 50$ °C, then DV1 is activated,
– heating signal > 0, then both heating pump and heating fan are activated,
– heating signal > 0 and ($T_{\mathrm{Coll\ out}} - T_{\mathrm{Coll,in}}$) > 0, then collector pump is activated,
– $\dot{m}_{\mathrm{w,drave}} = 0$, then only space heating (no flow *T*-piece 2) is activated, and
– $\dot{m}_{\mathrm{w,drave}} > 0$, then both MV2 and DV2 are activated (both space heating and water drawing).

Performance parameters

The performance parameters of the system were evaluated using the seasonal performance factor (SPF) calculated:

$$SPF = \frac{\dot{Q}_{\mathrm{load,\,tot}}}{\dot{W}_{\mathrm{net}}} \tag{1}$$

where $\dot{Q}_{\mathrm{load,tot}}$ is the sum of the required energy rate for heating, cooling and DHW:

$$\dot{Q}_{\mathrm{load,\,tot}} = \dot{Q}_{\mathrm{SH}} + \dot{Q}_{\mathrm{DHW}} + \dot{Q}_{\mathrm{SC}} \tag{2}$$

The total power consumption, $\dot{W}_{\mathrm{tot}}$, was obtained using:

$$\begin{aligned}\dot{W}_{\mathrm{tot}} = \dot{W}_{\mathrm{HP}} + \dot{W}_{\mathrm{CWT,pump}} + \dot{W}_{\mathrm{HWT,pump}} + \dot{W}_{\mathrm{p,coll}} + \dot{W}_{\mathrm{H,pump}} + \dot{W}_{\mathrm{C,pump}} + \\ + \dot{W}_{\mathrm{aux,HWT}} + \dot{W}_{\mathrm{aux,CWT}} + \dot{W}_{\mathrm{extr}} + \dot{W}_{\mathrm{H,fan}} + \dot{W}_{\mathrm{C,fan}}\end{aligned} \tag{3}$$

The net energy consumption, $\dot{W}_{\mathrm{net}}$, was obtained using:

$$\dot{W}_{\mathrm{net}} = \dot{W}_{\mathrm{tot}} - \dot{W}_{\mathrm{PV/T}} \tag{4}$$

The solar electric fraction (the ratio of electrical energy obtained from the sun to the total electrical energy consumption) was calculated using:

$$f_{\mathrm{sol,\,el}} = \frac{\dot{W}_{\mathrm{PV/T}}}{\dot{W}_{\mathrm{tot}}} \tag{5}$$

The solar thermal fraction (the ratio of solar energy obtained to the total electrical energy consumption) was calculated:

$$f_{\mathrm{sol,\,th}} = \frac{\dot{Q}_{\mathrm{PV/T}}}{\dot{W}_{\mathrm{tot}}} \tag{6}$$

The coefficients of performance of the heat pump for cooling and heating were expressed as $\mathrm{COP_c}$ and $\mathrm{COP_h}$, respectively, and calculated:

$$COP_{\mathrm{h,HP}} = \frac{\dot{Q}_{\mathrm{HP,L}}}{\dot{W}_{\mathrm{HP}}} \tag{7}$$

$$COP_{\mathrm{c,HP}} = \frac{\dot{Q}_{\mathrm{HP,S}}}{\dot{W}_{\mathrm{HP}}} \tag{8}$$

Analysis

The data obtained as a result of the simulation are presented in the form of graphics. In order to examine the performance of the system under different climatic conditions, the system was simulated for the climatic conditions of five different provinces (Istanbul, Ankara, Izmir, Trabzon, and Hakkari) located in five different regions in Turkey, fig. 4.

Provinces	Regions	Latitude and longitude
Istanbul	Marmara	41° 0' 49.8'' N / 28° 56' 58.8'' E
Izmir	Aegen	38° 24' 45.8'' N / 27° 8' 18.2'' E
Ankara	Central Anatolia	39° 55' 11.5'' N / 32° 51' 15.4'' E
Trabzon	Black sea	41° 0' 18'' N / 39° 43' 37'' E
Hakkari	Eastern Anatolia	37° 34' 38.7'' N / 43° 44' 12.4'' E

Figure 4. Locations and co-ordinates of provinces in Turkey used in the study [21]

Operation in winter season (heating and DHW for first week of February)

The period from 30th April to 27th September was designated as the cooling season and from 22nd November to 11th March as the heating season. Figure 5 shows the data of the heat pump operation for the first day of February under Istanbul climatic conditions.

According to the simulation results, the heat pump was heavily activated between 08:00 in the morning and 16:00 in the afternoon and the heating signal was active. The heat pump, which consumes an average of 3.7-5 kW of power, drew 10.5-21.5 kW of heat from the CWT during the day and transferred 15.5-26 kW of heat to the HWT. During the day, the heat pump source outlet temperature varied between 12 °C and –4 °C, whereas the load outlet temperature varied between 45 °C and 50 °C.

As seen in fig. 6, the ambient temperature on the first day of February was around 8 °C, and the room temperature was kept constant at around 21 °C between 08:00 a. m. and 22:00 p. m. (restaurant operating hours).

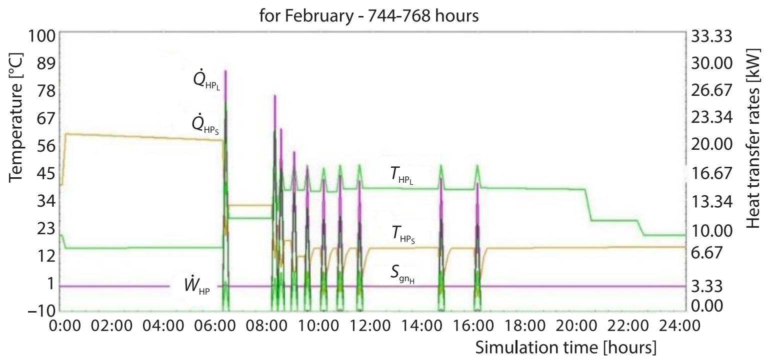

Figure 5. Changes of temperature and heat transfer rates of the heat pump

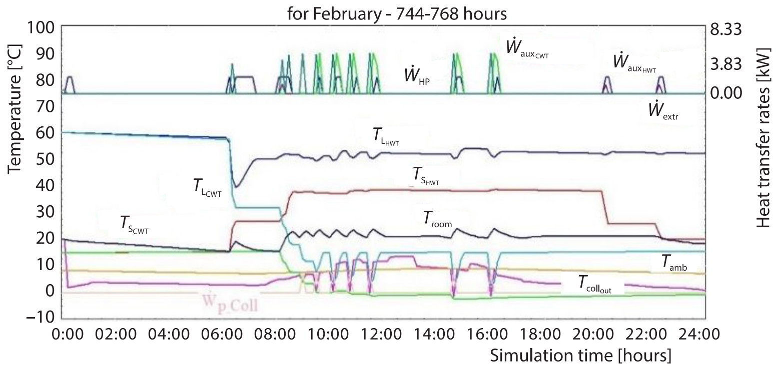

Figure 6. Temperature changes in HWT and CWT

The heat pump was frequently activated between 08:00 a. m. and 12:00 a. m., and after 16:00 p. m. it turned off with the increase in space-heat loads. Each time the heat pump was switched on, the water temperature value in the upper layer of the tank decreased as the heat was drawn from the CWT.

The CWT was divided into 10 layers, and two 5 kW electric heaters and thermostats were placed in the 2nd and 8th layers. When the water temperature in the 2nd layer dropped to 10 °C, the heater in this layer was activated and turned off at 15 °C. On the other hand, it was activated when the water temperature in the 8th layer dropped to –10 °C and was deactivated at –5 °C.

The HWT was also divided into ten layers, and two 2 kW electric heaters and thermostats were placed in the 2nd and 8th layers. When the water temperature in the 2nd layer dropped to 50 °C, the heater in this layer was activated and turned off at 55 °C. Conversely, it was activated when the water temperature in the 8th level dropped to 25 °C and was switched off at 30 °C. When the heat pump was activated, the water temperature value in the lower and upper layers of the HWT increased.

If the water temperature of the upper layer of the tank did not rise above 50 °C despite the heat pump being activated, then the electrical heaters located in the HWT were activated and raised the water temperature to 50 °C. If the water temperature could not be increased to 50 °C with the activation of the heaters in the HWT, then the external heater was activated. The collector pump was activated seven times between 09:00 a. m. and 16:00 p. m. in order to benefit from solar energy.

Figure 7 shows the PV/T collector inlet and outlet water temperature changes, the useful energy gain from the sunand the amount of power produced by the photovoltaic cells during the day. At this hour, the temperature of the water entering the collector was –0.87 °C, and the water leaving the collector was 13.6 °C. The highest amount of useful energy gained from the sun (1464 W) was obtained at 12:00 p. m., whereas the highest energy produced by the photovoltaic cells (584 W) was obtained at 13:00 p. m.

As seen in fig. 8, the heating pump and the heating fan were activated at 06:00 a. m. with the start of water withdrawal and the room temperature was kept constant at 21 °C between 08:00 a. m. and 22:00 p. m. The heating pump pumped 1440 Lph of hot water at approximately 50 °C from the HWT to the heating coil. Here, the water, which had given its heat to the room air, had cooled down to around 10-15 °C and was returned to the HWT. Room air of approximately 21 °C was heated (in the range of 19-25 kW) at 6000 kg per hour in the heating coil and was sent to the room at 35 °C.

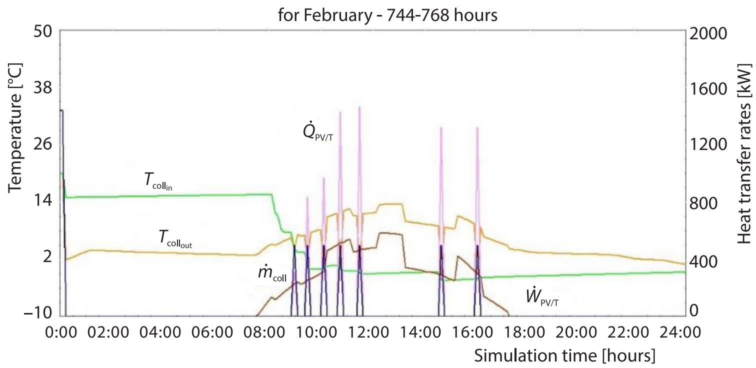

Figure 7. Changes in temperature, energy, and power of the PV/T collector

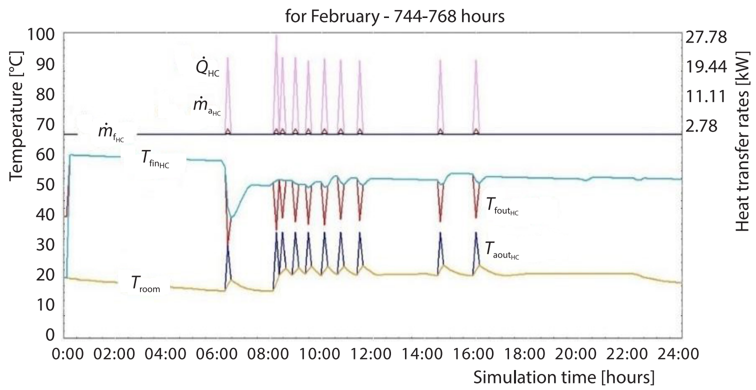

Figure 8. Changes of temperatures and heat transfer rates in the heating coil

Operation in summer season (cooling and DHW for first week of July)

On the first day of July, the heat pump was activated frequently between 10:00 a. m. and 22:00 p. m., fig. 9. When the heat pump was activated, it consumed an average of 4.2 kW of power, drawing 18.3-20.3 kW of heat from the CWT, and in the meantime sending 22-25 kW of heat into the HWT. Meanwhile, during the day, the source temperature varied between 8-14 °C, whereas the load temperature varied between 42-45 °C. Figure 10 shows that the

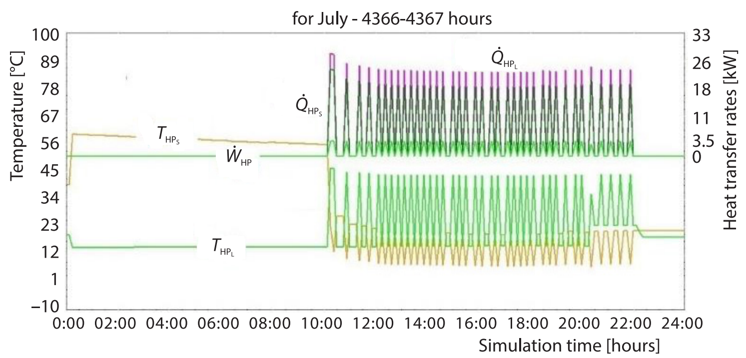

Figure 9. Changes of temperature and heat transfer rates of the heat pump

lowest ambient daytime temperature on the first day of July was 18 °C and the highest 27 °C. In order to keep the room temperature constant at 24 ±0.5 °C, the heat pump was activated frequently between 10:00 a. m. and 22:00 p. m. With the activation of the heat pump, the water temperature in the CWT bottom layer dropped to 8 °C, whereas the water temperature returning from the cooling coil to the CWT was approximately 21 °C. The heaters in the HWT were activated when the upper layer water temperature dropped below 50 °C. The HWT heaters were activated to provide the desired constant water temperature between 06:00-08:00 a. m. and to assist the heat pump between 20:00 and 22:00 p. m. The external heater was activated three times: at 08:00 a. m., 20:00 p. m., and 22:00 p. m.

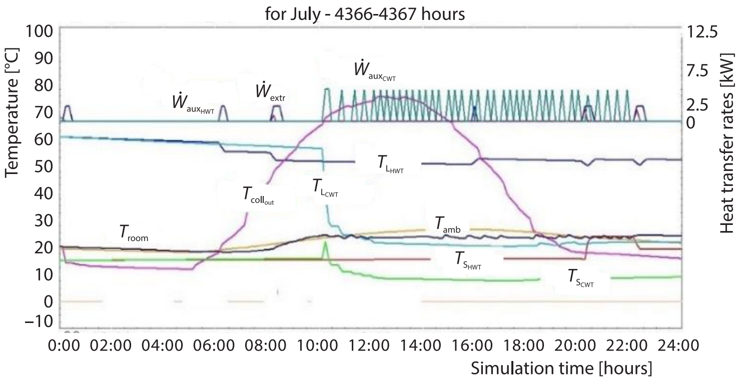

Figure 10. Temperature changes in HWT and CWT

Although the water temperature at the PVT collector outlet reached 75 °C, the collector pump did not activate because, in July, the PVT collector was used only for electricity generation. Due to the long summer days and high radiation, 7.61 kW of electricity was produced from the PV/T collector between 05:00 a. m. and 20:00 p. m., fig. 11.

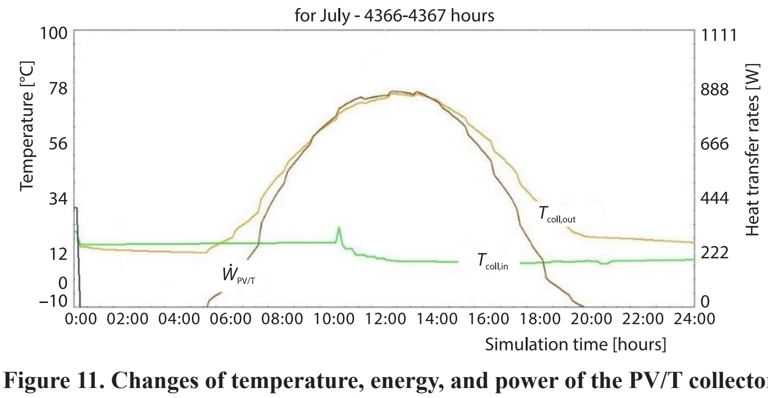

Figure 11. Changes of temperature, energy, and power of the PV/T collector

As seen in fig. 12, the collector pump was activated only in the winter months, whereas electricity generationok place throughout the year.

In order to keep the room temperature constant at 24 °C in summer, the heat pump was activated and drew heat from the CWT. The water in the tank, into which 30% glycol had been added, was then pumped to the cooling coil where it was cooled down to about 8-13 °C during the day. It was then pumped to the CWT again via the cooling pump at 1440 kg per hour, fig. 13. Room air at 6000 kg per hour was passed through the cooling coil. Heat in the range of

16-20 kW was drawn from the room air and the room relative humidity was reduced to 58% during the day.

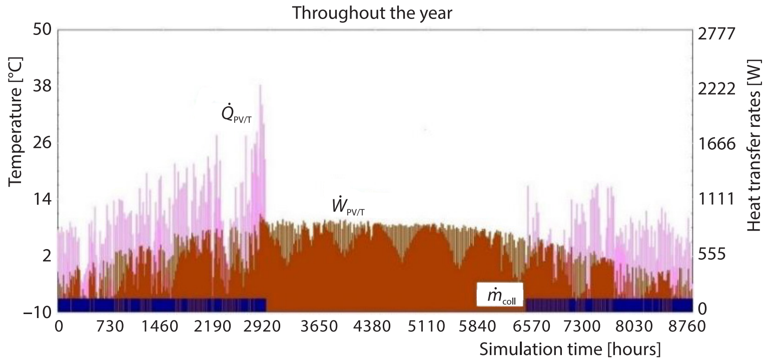

Figure 12. Changes of temperature, energy, and power of the PV/T collector throughout the year

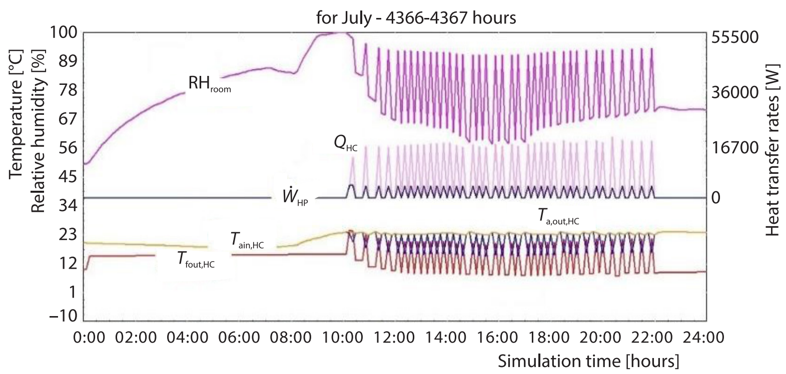

Figure 13. Changes in temperature and amount of heat transfer in the cooling coil

In this system, the problem of excessive rise in water temperature in the HWT during the summer months was the most important handicap. In these months, because only domestic water is drawn from the HWT, sufficient heat cannot be removed. As seen in fig. 14, 1440 kg of hot water per hour entered the HEX at approximately 43 °C and left at 30 °C. Meanwhile, an average of 22000-25000 W of heat was released into the ambient air. Thus, the tank load temperature was kept constant at 50 °C.

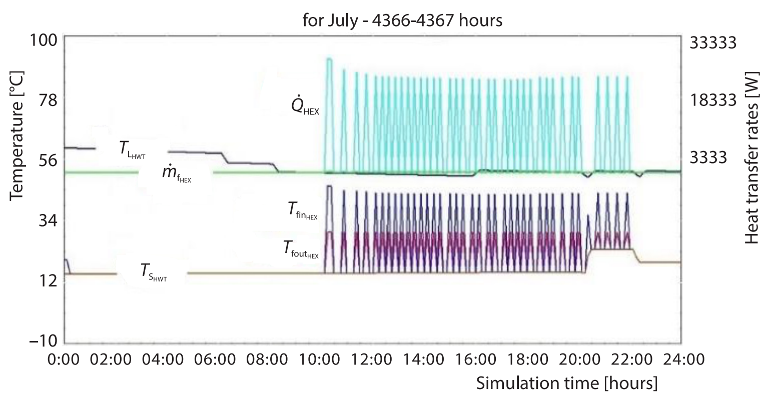

Figure 14. Changes of temperature and heat transfer rates of HEX

Figure 15 shows that throughout the year, HEX was only activated in the summer months.

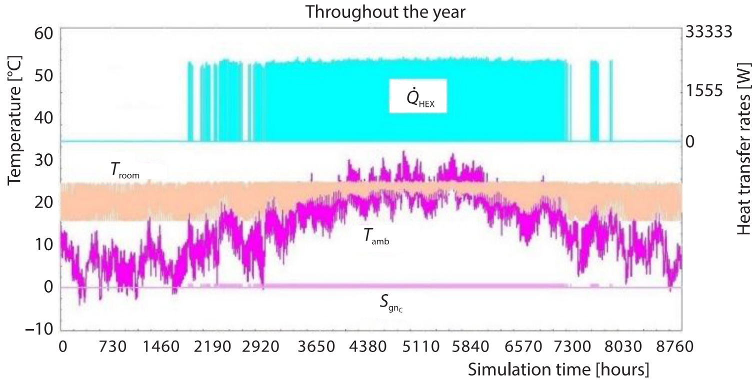

Figure 15. Activation times of HEX throughout the year

Results and discussion

The performance of the heat pump system using a dual-tank system with an integrated PV/T collector was simulated for five different provincial climate conditions in Turkey and the results are presented comparatively in tab. 2. According to the results, moving from the eastern to the western provinces, the net electricity consumption decreased with the effect of climate. The device that consumed the most electricity in the system was the heat pump, followed by the HWT and CWT water tank auxiliary heaters. For Izmir, which is the hottest province, 44.87% of the total power consumption was expended by the heat pump and 44.02% by all auxiliary heaters, whereas the remainder was consumed by the pumps, and that of the pumps was consumed by the fans. For Hakkari, as the coldest province, 31.59% was consumed by the heat pump and 60.69% by all auxiliary heaters, whereas the rest was consumed by the pumps and the fan. Despite the fact that Hakkari is the province that benefits most from the sun with both electricity generation and thermal energy, the highest energy consumptionok place here because of the very cold winter season. In Hakkari, the energy spent for heating the HWT and the CWT was very high. Although the amount of PV/T collector utilization was very high, the solar electrical fraction was low because of the harsh winter conditions. In Trabzon, the amount of electricity produced from the PV/T collectors was the lowest because of the climate of the region. Most days are cloudy and rainy. However, due to the mild climate of the province, annual energy consumption was also extremely low.

The province of İzmir, located in the hottest climate zone, consumed the most energy for the annual cooling requirement, and Trabzon, in the north where the summer months are mostly rainy, was the province with the lowest cooling needs. The province of Hakkari is located in the east and is exposed to heavy snowfall in winter. Thus, Hakkari consumed the most energy for annual heating and also had the closest amounts of annual heating and cooling energy consumption.

As seen from fig. 16 the Mediterranean climate provides the most suitable conditions for the use of the SADHP system. In addition the Mediterranean climate, the Black Sea climate, with an average annual temperature of 13-15 °C, also presents suitable conditions for the use of the system. The performance values obtained for the province of Trabzon also support this argument. For Izmir, 17.68% of the total energy consumption was obtained from the sun via the PV/T collector. This value was 12% for Hakkari Province.

Table 2. Annual energy consumption and heat transfer rates

	Izmir		Trabzon		Istanbul		Ankara		Hakkari	
	[kW]	[%]	[kW]	[%]	[kW]	[%]	[kW]	[%]	[kW]	[%]
$\dot{W}_{\text{all aux.heaters}}$	40.72,71	44.02	4310.44	52.65	5042.45	51.17	6935.94	57.37	8760.21	60.69
$\dot{W}_{\text{fans and pumps}}$	1027.45	11.11	766.92	9.37	950.84	9.65	1015.44	8.40	1113.01	7.71
$\dot{W}_{\text{HP}}$	4151.77	44.87	3109.82	37.98	3861.26	39.18	4138.07	34.23	4560.01	31.59
$\dot{W}_{\text{tot}}$	9251.93	100.00	8187.18	100.00	9854.55	100.00	12089.46	100.00	14433.23	100.00
$\dot{W}_{\text{PVT}}$	1635.81		1356.07		1644.37		1707.10		1744.93	
$\dot{W}_{\text{net}}$	7616.12		6831.11		8210.18		103382.37		12688.31	
Heat transfer rates										
$\dot{Q}_{\text{PVT}}$	67.92		80.50		85.04		98.48		113.57	
$\dot{Q}_{\text{HP,L}}$	21506.98		15949.99		19527.53		19632.37		20661.81	
$\dot{Q}_{\text{HP,S}}$	17355.21		12840.17		15666.27		15494.29		16101.80	
$\dot{Q}_{\text{SC}}$	15489.87		10809.76		13109.01		11689.62		11007.21	
$\dot{Q}_{\text{SH}}$	3447.89		3722.00		4803.55		7455.06		9920.93	
$\dot{Q}_{\text{DHW}}$	1256.95		1224.37		1231.42		1157.68		1128.42	

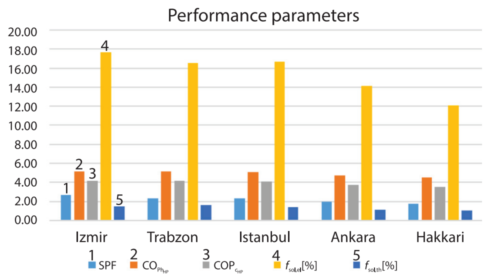

Figure 16. System performance parameters for all provinces

Conclusions

In this study, a designed PV/T collector-integrated dual-tank heat pump system was modeled and simulated using the TRNSYS program, and performance analyses were performed for five provinces in five different regions in Turkey. The following conclusions were reached by the study.

- Although the heat pump was activated more frequently in hot climates, the electric heaters were activated more frequently in cold climates. As a result, the total energy consumption increased.
- The PV/T collector compensated for 17.7% of the total electrical energy consumption of the system in Izmir Province, but for only 12.09% in Hakkari Province. This ratio could be improved by using more PV/T modules. The number of PV/T modules should be selected by considering the initial investment cost and operating costs.
- As a result of the analyses made for the five different provinces, Izmir was determined to be the most suitable province for the SADHP system. The highest SPF value (2.65) was obtained for the province of Izmir and the lowest (1.74) for the province of Hakkari. The system worked 52% more efficiently in Izmir than in Hakkari. In order to expand the use of such systems employing renewable energy sources, it is important to promote them in all provinces with climatic characteristics similar to the province of Izmir (Mediterranean climate).

Nomenclature

A – area, [m^2]
$COP_{\mathrm{h,HP}}$ – coefficients of performance of the heat pump for heating, [–]
$COP_{\mathrm{C,HP}}$ – coefficients of performance of the heat pump for cooling, [–]
$f_{\mathrm{sol,el}}$ – solar electric fraction, [%]
$f_{\mathrm{sol,th}}$ – solar thermal fraction, [%]
$\dot{m}_{\mathrm{a,HC}}$ – air-flow rate passing though the heating coil, [kgh^{-1}]
$\dot{m}_{\mathrm{coll}}$ – mass-flow rate of fluid passing through the PV/T collector, [kgh^{-1}]
$\dot{m}_{\mathrm{f,HEX}}$ – mass-flow rate of hot water passing through the heat exchanger, [kgh^{-1}]
$\dot{m}_{\mathrm{w,draw}}$ – amount of water drawn, [kgh^{-1}]
$\mathrm{RH}_{\mathrm{room}}$ – room relative humidity, [%]
Sgn_{C} – cooling signal, [–]
Sgn_{H} – heating signal, [–]
SPF – seasonal performance factor of the system, [–]
$\dot{Q}_{\mathrm{DHV}}$ – rate of energy transfers for domestic hot water preparation, [kWh]
$\dot{Q}_{\mathrm{HC}}$ – heat transfer rate of heating coil, [kW]
$\dot{Q}_{\mathrm{HEX}}$ – heat transfer rate of heat exchanger, [kW]
$\dot{Q}_{\mathrm{HP,\,L}}$ – heat transfer rate of the load-side fluid transferred by the heat pump, [kWh]
$\dot{Q}_{\mathrm{HP,\,S}}$ – heat transfer rate from the source fluid transferred by the heat pump, [kWh]
$\dot{Q}_{\mathrm{load,\,tot}}$ – total heat transfer, [kWh]
$\dot{Q}_{\mathrm{PV/T}}$ – useful energy gain from sun via PV/T collector, [kW]
$\dot{Q}_{\mathrm{SH}}$ – space heating heat transfer rate, [kWh]
$\dot{Q}_{\mathrm{SC}}$ – space cooling heat transfer rate, [kWh]
$T_{\mathrm{ain,HC}}$ – inlet air temperature of heating coil, [°C]
$T_{\mathrm{ain,CC}}$ – inlet air temperature of cooling coil, [°C]
$T_{\mathrm{aout,HC}}$ – outlet air temperature of heating coil, [°C]
T_{amb} – ambient temperature, [ºC]
$T_{\mathrm{coll,in}}$ – inlet fluid temperature of PV/T collector, [°C]
$T_{\mathrm{coll,out}}$ – outlet fluid temperature of PV/T collector, [°C]
$T_{\mathrm{fin,HEX}}$ – inlet fluid temperature of heat exchanger, [°C]
$T_{\mathrm{fin,HC}}$ – inlet fluid temperature of heating coil, [°C]
$T_{\mathrm{fout,HEX}}$ – outlet fluid temperature of heat exchanger [°C]
$T_{\mathrm{fout,HC}}$ – outlet fluid temperature of heating coil, [°C]
$T_{\mathrm{fout,CC}}$ – outlet fluid temperature of cooling coil, [°C]
$T_{\mathrm{HP,\,L}}$ – heat pump load temperature, [ºC]
$T_{\mathrm{HP,\,S}}$ – heat pump source temperature, [ºC]
$T_{\mathrm{L,CWT}}$ – CWT load temperature [°C]
$T_{\mathrm{L,HWT}}$ – HWT load temperature, [°C]
T_{room} – room temperature, [ºC]
$T_{\mathrm{S,CWT}}$ – CWT source temperature, [°C]
$T_{\mathrm{S,HWT}}$ – HWT source temperature, [°C]
$\dot{W}_{\mathrm{aux,HWT}}$ – energy consumption of auxiliary heaters in HWT, [kWh]
$\dot{W}_{\mathrm{aux,CWT}}$ – energy consumption of auxiliary heaters in CWT, [kWh]
$\dot{W}_{\mathrm{C,fan}}$ – energy consumption of cooling fan, [kWh]
$\dot{W}_{\mathrm{C,pump}}$ – energy consumption of HWT pump, [kWh]
$\dot{W}_{\mathrm{extr}}$ – required heating rate of external heater, [kW]
$\dot{W}_{\mathrm{H,fan}}$ – energy consumption of heating fan, [kWh]
$\dot{W}_{\mathrm{HP}}$ – energy consumption of heat pump, [kWh]
$\dot{W}_{\mathrm{CWT,pump}}$ – energy consumption of CWT pump, [kWh]
$\dot{W}_{\mathrm{H,\,pump}}$ – energy consumption of heating pump, [kWh]
$\dot{W}_{\mathrm{HWT,pump}}$ – energy consumption of HWT pump, [kWh]
$\dot{W}_{\mathrm{net}}$ – net energy consumption of system,[kWh]
$\dot{W}_{\mathrm{p,coll}}$ – power consumption collector pump, [kWh]
$\dot{W}_{\mathrm{tot}}$ – consumption, total energy, [kWh]

Subscripts

WAK – wall adjacent kitchen
WAS – wall adjacent store room
WDW – western wall of the dining room
WDE – eastern wall of the dining room
WDS – southern wall of the dining room

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