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Experimental thermodynamic evaluation of a real cooling system with seawater-source and heat pump for a hotel in Turkey

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Abstract

It is noteworthy that energy consumption is very high in hotels in order to provide the comfort requirements of human beings. Especially in summer season, cooling is an important condition for comfort in a hotel. In hotel cooling, seawater is an indispensable heat sink. In this study, the thermodynamic performance of a hotel cooling system with seawater-source and heat pump used with the aim of cooling a hotel located by sea is investigated. Using hourly data collected from the existing system during the summer of 2016, exergy analysis is performed. As the cooling system is at an industrial-scale involving copious equipment, it is divided into five different cooling zones. For cooling of a selected hotel room, the heat pump and variable refrigerant flow system in the zone where the room was located is also assessed. The results indicated that the hotel cooling system has the coefficient of performance, exergy efficiency, and improvement potential of 4.37, 66.5% and 77.7% on average, respectively. The zone requiring priority improvement is the fifth cooling zone due to air-conditioning plants and long pipelines. The equipment with the highest exergy destruction is the seawater heat exchanger. Due to low temperature difference between hotel room and seawater, it is necessary to increase use of these systems and develop them in terms of technology.

KEYWORDS

exergy analysis, heat pump, hotel cooling, seawater source, thermodynamic performance

1 | INTRODUCTION

The interest in renewable energy resources is increasing each day due to the fact that the variety and amount of fossil-sourced energy on the earth will be consumed and also due to negative effects on the environment. Seawater (SW) is one of renewable resources. SW is a clean, abundant, reliable, and sustainable energy source for electricity production, heating, cooling, and distillation processes. However, SW is mostly used for cooling of many power plants in a variety of cycles and for space heating and cooling in low temperature heat pump (HP) applications. For cities located on sea coasts to achieve space cooling and heating, SW-sourced systems are perfect technology to benefit from SW energy. Thus, systems replace traditional cooling and heating and will reduce the energy consumption and CO_2 emissions of buildings.[1-3]

Generally, cities located on sea coasts are very popular in the tourism sector. The size and scope of the tourism sector is very important in terms of global resources and even small changes can lead to large effects. One of the most intense energy sectors in the field of

Abbreviations: BF, Bernoulli filter; BWIS, beach well infiltration intake system; COP, coefficient of performance; DX, direct-expansion; HCHE, helical coil heat exchanger; HVAC, heat, ventilation, and air-condition; SHE, seawater heat exchanger; SWHP, seawater-source heat pump; SW-HP, seawater-source and heat pump; VBWI, vertical beach well intake; VRF, variable refrigerant flow.

tourism is the hotel industry. Energy costs comprise the largest portion of general operating costs of a hotel after personnel costs. Generally, the annual average energy consumption per floor area for hotels is related to climate. Additionally, the energy performance of hotels has been investigated in many studies. A research based on the energy analyses data reported for European Hotels reports that hotels consume 200–400 kWh/m^2 each year.[4] For example; the energy use intensity was 389 kWh m^{-2} year^{-1} for only 32 hotels in the Antalya province in Turkey,[5] while it was 308 kWh m^{-2} year^{-1} for 31 hotels in Spain.[6] Theirs electricity consumptions accounts for 45–62% and 83% of theirs total energies, respectively. Typically, nearly half the electrical energy consumed is used for heat, ventilation, and air-condition (HVAC).[7,8]

Air-conditioning systems in hotels are generally responsible for the majority of the total electricity consumption due to solar radiation and ambient temperatures. The operation of the air-conditioning systems causes a large electricity consumption, therefore energy cost reduction programs in hotels focus on reducing the energy consumption of the HVAC systems. Within these programs, the majority consider low temperature HP applications. However, in HP applications heat is generally discharged from the air as a heat source to the water as a heat sink. In this situation, the choice of heat source or sink for the HP is very important. Hotels located on the coast, especially, may choose SW as the heat source or sink for HPs used to heat or cool hotels. Generally, the efficacy of SW as heat source for HPs may be explained by two principles. The first is that water is a much better heat transfer fluid compared to air and thus, heat is transported in a much more efficient fashion. The second is that SW is warmer than the external air temperature in winter and cooler than the external air temperature in summer and this makes it a more efficient heat source. In the case of HPs, the warmer source generally increases indoor temperature in the winter months and in summer the cooler source decreases the indoor temperature, henceforth contributing comfort levels.

SW-sourced air-condition systems can be divided into two main classes based on how SW is obtained by (a) direct/surface intake and (b) indirect/underground intake. Direct SW intake uses raw SW. The design and construction of the intake location is generally open and easy, and has unlimited intake capacity. These intake locations create problems due to corrosive effects of SW, freezing, blockage due to material at macroscopic and microscopic dimensions, and device breakdown. Indirect SW intake uses natural geology on the coastline or on the seafloor to filter raw SW. Indirect SW intake uses a variety of underwater systems like vertical wells, horizontal wells, corner wells, collecting wells, collector tunnels, seabed galleries, and beach galleries linked to the geological characteristics of the facility location. Especially, a vertical beach well intake (VBWI) significantly reduces the environmental effects with natural surface absorption.

In the literature there are many studies about a variety of heating and cooling systems operated with indirect intake of SW and with heat exchangers inserted directly in SW: Zhang et al[9] experimentally investigated the heat transfer performance of surface water-sourced HP system used square copper wound high-density polyethylene tube and compared it with smooth high-density polyethylene tube. The results showed that the coefficient of performance (COP) of the HP with coined tube is higher than that of the HP with the smooth tube. Haiwen et al[10] performed field measurements of a district heating system using HP units with stored SW on Dalia Island in China for a district heating system fed by coal. They stated in their works that the improvement potential for energy yield of the HP units, and their COP would increase by 24.2 and 34.6%, respectively. Si et al[11] presented a new model for calculating and optimizing the hourly water temperature at the condenser input of a river water-sourced HP on the Yangtze River in China. They provided a definite and effective method to calculate the energy consumption of the water cooling unit. Schibuola and Scarpa[12] carried out experimental studies based on year-round performance for heating and cooling processes with a surface water-sourced HP using lagoon water to provide HVAC requirements for monumental hotels in Venice/Italy. Compared with air-sourced HP systems, the lagoon water provided an energy saving of more than 20% for the energy consumption requirements of HVAC and a reduction in greenhouse gas emissions. Zheng et al[13] investigated the thermal performance of a seawater-sourced heat pump (SWHP) system with subsea helical coil heat exchanger (HCHE) inserted directly in SW numerically and experimentally. They investigated the effects of different parameters on the thermal performance of the system. Their results found that subsea heat exchangers were very helpful in the design. Zheng et al[14] experimentally and numerically examined the heat transfer performance of a SWHP with a subsea HCHE. They researched the effect of SW flow rate on the heat transfer performance of the subsea HCHE. The results showed that considering the SW flow rate in the operation of subsea heat exchangers was very close to reality and developing mathematical models would be very beneficial for design and optimization of subsea heat exchangers used in SWHP systems. Zheng et al[15] investigated the thermal performance of a SWHP for radiant floor heating during the whole heating season in cold regions of China. A high-density polyethylene HCHE was inserted directly into the flowing SW transferring the heat of the SW to the HP. Thus, freezing problems were prevented in the subsea heat exchanger and its heat transfer performance increased. Zou and Xie[16] developed a simplified method to calculate and estimate the energy consumption of a SWHP unit in lake water. They obtained models calculating the input water temperature for the lake water side.

According to the literature review above-mentioned as well as to the knowledge of the authors, the thermodynamic performance of a seawater-source and heat pump (SW-HP) system has not been studied so far for the cooling of a hotel under operation. This is a main source of motivation for the present study. Furthermore, there is only one study on the SWHP system with VBWI in open literature. Xin et al[17] performed field tests to compare the performance of SWHP systems with VBWI and stored SW (direct intake) in the Liaodong Peninsula/China. The average unit COP was 2.99 and 4.66 for the stored SW and the VBWI systems, respectively, at 5.83 and 12.85°C of the average SW temperature. However, the study mentioned above is not a system operated by collecting SW through VBWI under a hotel with a large cooling load and by using a heat transfer fluid between the HPs and the SW line to transport heat over long distances. The aim of the present study is to assess and track the thermodynamic performance of an available hotel cooling system with SW-HP used for cooling of the Asia Beach Resort SPA Hotel in the Alanya county in Antalya/Turkey as a case study. With this aim, exergy analysis is applied to the hotel cooling system. Thus, a better understanding of the interactions between system equipment, and the potential for system improvement is provided. In the second section of the study, information about the hotel, the cooling system, and data collected from the system is introduced in the summer of 2016, while in the third section to assess the thermodynamic performance of the system the exergy analysis and its assumptions are mentioned. In the fourth section the results of the analyses for the hotel cooling processes are discussed. The general conclusions of the study are stated in the final section.

2 | DESCRIPTION OF THE SYSTEM

Turkey has an area of 780,580 km^2 with a total coastline of 8,430 km. On this long coastline, Antalya is a coastal city located in the south of Turkey (the Mediterranean Sea coastline). Because of the archaeological and natural richness of the area, it has significant potential for tourism development.[5] Therefore, comfort conditions for shelter and accommodation services in hotels gain importance. Cooling services are very important in the summer season. To achieve the aims of this study, the Asia Beach Resort SPA hotel in Alanya county in Antalya/Turkey which uses a SW-HP for hotel cooling is chosen. The hotel shown in Figure 1 has 32,000 m^2 enclosed area with 318 standard rooms, 2 disabled rooms, 24 family suite rooms, 12 family rooms, and 6 honeymoon suites for a total of 360 rooms. The hotel is a single building directly on the coast with 10 floors, 3 of which are underground, with open and enclosed parking area, and only open-air swimming pools.

The heating/cooling center for the hotel comprises the third basement floor. The facility is fed by appropriate shaft outputs. Cooling processes for the hotel use the following cooling system equipment; 122 cooling group external units, 560 variable refrigerant flow (VRF) system internal units, and 8 direct-expansion (DX) serpentine air-conditional stations. The cooling load of the hotel is calculated with the design criteria listed in Table 1 under stable state conditions using the Hourly Analysis Program[18] belonging to Carrier company. The design peak cooling load of the hotel is determined as 69 W/m^2.

The schematic flow diagram for the hotel cooling system with SW-HP operated for cooling of the hotel illustrated in Figure 1 is shown in Figure 2. Additionally, the technical specifications of the equipment in the hotel cooling system are given in Table 2. As seen in Figure 2, there are four SW collecting wells in the hotel cooling system. The wells when limited by underground seepage capacity, sea seepage capacity, and current hydrostatic pressure have VBWI[19] and are constructed 150 m from the coast and depths of 6 m below sea level to provide the input water amount. The wells are found in a sunken pumping station comprising 2 m cylindrical pieces on the coast and under the hotel complex, as in Figure 3. As can be shown in Figure 2, in the primary cycle loop SW pumps to a Bernoulli filter (BF) with 1.2 bar of pressure and 95 tons/hr of mass flow rate from SW pumps in each well. The filter removes foreign objects (weeds, silt, mussels, etc.) and the SW passes to a seawater heat exchanger (SHE) with 380 tons/hr of flow rate. In the SHE, the SW receives the heat from cold water in the secondary cycle loop and is discharged into the sea.

The cold water with reduced temperature in the secondary cycle loop is pumped to the cooling zones by secondary pumps with 380 tons/hr. The large cooling capacity of the hotel is divided into five different cooling zones due to the high number of rooms with cooling requirements and as a result the number of cooling equipment needed. The circulation pumps take part in the first, second, third, fourth, and fifth cooling zones have 72, 84, 72, 72, and 80 ton/hr flows pumping cooled water to the hotel cooling equipment (external units, HP, and air-conditioning station). The hotel rooms are cooled with a VRF system. In this study, to make the hotel cooling system clearer and more comprehensible to readers, the fourth cooling zone

FIGURE 1 A general view of the Asia Beach Resort SPA Hotel [Color figure can be viewed at wileyonlinelibrary.com]

TABLE 1 The design criteria selected to cool the hotel

Parameters	Values
Location	Alanya, Antalya, Turkey
Latitude	36° North
Altitude	6 m
Outdoor design conditions for summer	
Dry thermometer temperature	39°C
Wet bulb temperature	28°C
Daily temperature change	11.4°C
Relative humidity	80%
Indoor design conditions for summer	
Natural spaces (bedrooms, lobby, etc.)	
Cooling operation	24°C
Heating operation	22°C
Humidity control (min-max)	30–60%
Service spaces (kitchen, laundry, etc.)	
Cooling operation	26°C
Heating operation	20°C
Humidity control (min-max)	30–60%

is chosen. As a result, as seen in Figure 2, the operation, thermodynamic performance, and improvement topics for a VRF and HP are presented for a hotel room chosen in the fourth zone. The comfort conditions of the chosen room are 24°C and 30–60% as listed in Table 1.

The cooling and ventilation systems are created to ensure a comfortable indoor environment with the aim of cooling inside the hotel. Therefore, during the day in the summer months in the Alanya county of Antalya/Turkey, there is a need for cooling due to intense solar radiation. The changes in hourly ambient and SW temperatures during the multi-annual period (1970–2016) for Alanya, Antalya/Turkey are given in Figure 4. As seen in Figure 4, the ambient temperature usually ranges between 19 and 37°C during the summer season. The SW temperature also varies between 12 and 28°C. In the month of June, July, and August, the difference between SW and ambient temperatures are, respectively, 7.3, 7.5, and 6.1°C. For this reason, it is appropriate to use SW instead of the outdoor air as the environment where the heat is rejected by the hotel cooling system.

In this study, the pressure, temperature, and volumetric flow rate values for line numbers shown on Figure 2, were recorded momentarily by the hotel's Supervisory Control System in the cooling system tests performed within the month of June, July, and August, as of 2016. Thus, the data obtained were used to evaluate thermodynamic performance of the system. The system was continuously operated throughout summer. Additionally, ambient and SW temperatures were recorded.

3 | ANALYSIS

The fact that a thermodynamic system has maximum performance is a very important topic currently. Additionally, the performance of the system is evaluated by the first and second laws of thermodynamics in order to reach maximum performance. The energy analysis uses the first law of thermodynamics, while exergy analysis uses both the first and second laws. For the whole system and equipment, the mass, energy, and exergy balances may be, respectively, stated as follows:

$$\sum \dot{m}_{\text{in}} = \sum \dot{m}_{\text{out}} \tag{1}$$

$$\sum \dot{E}_{\text{in}} - \sum \dot{E}_{\text{out}} - \dot{E}_{\text{loss}} = 0 \tag{2}$$

and

$$\sum \dot{E}x_{\text{in}} - \sum \dot{E}x_{\text{out}} - \dot{I} = 0 \tag{3}$$

where $\dot{m}$, $\dot{E}$, $\dot{E}x$, and $\dot{I}$ denote mass flow rate, energy rate, exergy rate, and exergy destruction rate, respectively. Thus, the energy and exergy balances of the hotel cooling system may be written as:

$$\dot{Q}_{\text{SWs}} + \dot{W}_{\text{SWP}} + \dot{W}_{\text{BP}} + \dot{W}_{\text{ZP}} + \dot{W}_{\text{HP}} + \dot{W}_{\text{Fans}} - \dot{Q}_{\text{cl}} - \dot{Q}_{\text{loss}} = 0 \tag{4}$$

and

$$\dot{Q}_{\text{SW}}\left(1 - \frac{T_0}{T_{\text{SW}}}\right) + \dot{W}_{\text{SWPs}} + \dot{W}_{\text{BP}} + \dot{W}_{\text{ZPs}} + \dot{W}_{\text{HPs}} + \dot{W}_{\text{Fans}} - \dot{Q}_{\text{cl}}\left(1 - \frac{T_0}{T_{\text{cl}}}\right) - \dot{I} = 0 \tag{5}$$

or, more clearly,

$$\dot{E}x_{\text{SW}} + \dot{W}_{\text{SWPs}} + \dot{W}_{\text{BP}} + \dot{W}_{\text{ZPs}} + \dot{W}_{\text{HPs}} + \dot{W}_{\text{Fans}} - \dot{E}x_{\text{cl}} - \dot{I} = 0 \tag{6}$$

where $\dot{Q}$ and $\dot{W}$ is heat transfer rate and work rate, respectively. The subscript SW, SWP, BP, ZP, HP, and cl denote SW, SW pump, booster pump (BP), zone pump (ZP), HP, and cooling load. Additionally, $\left(1 - \frac{T_0}{T_k}\right)$ is remarkable for the readers. In cooling cycles, it contains more thermal energy $\dot{Q}$ than the system at T in k location if the system temperature is lower than reference T_0, and the exergy can flow into the system. Thus, the cooling exergy load, $\dot{E}x_{\text{cooling},k}$, is the required rate of exergy to be supplied to the kth system component at T to maintain the design conditions, as can be defined as.[21]

$\dot{E}x_{\text{cooling},k} = -\dot{Q}_k\left(1 - \frac{T_0}{T_k}\right)$ for $T_0 > T_k$(7)

For each state in any system, the exergy rate may be stated as follows;

$$\dot{E}x_i = (\dot{m} \cdot e)_i \tag{8}$$

where specific flow exergy, e, is;

$$e_i = [(h_i - h_0) - T_0(s_i - s_0)] \tag{9}$$

Exergy efficiency or the second law of thermodynamic efficiency, may be expressed as the ratio of total exergy output to total exergy input:

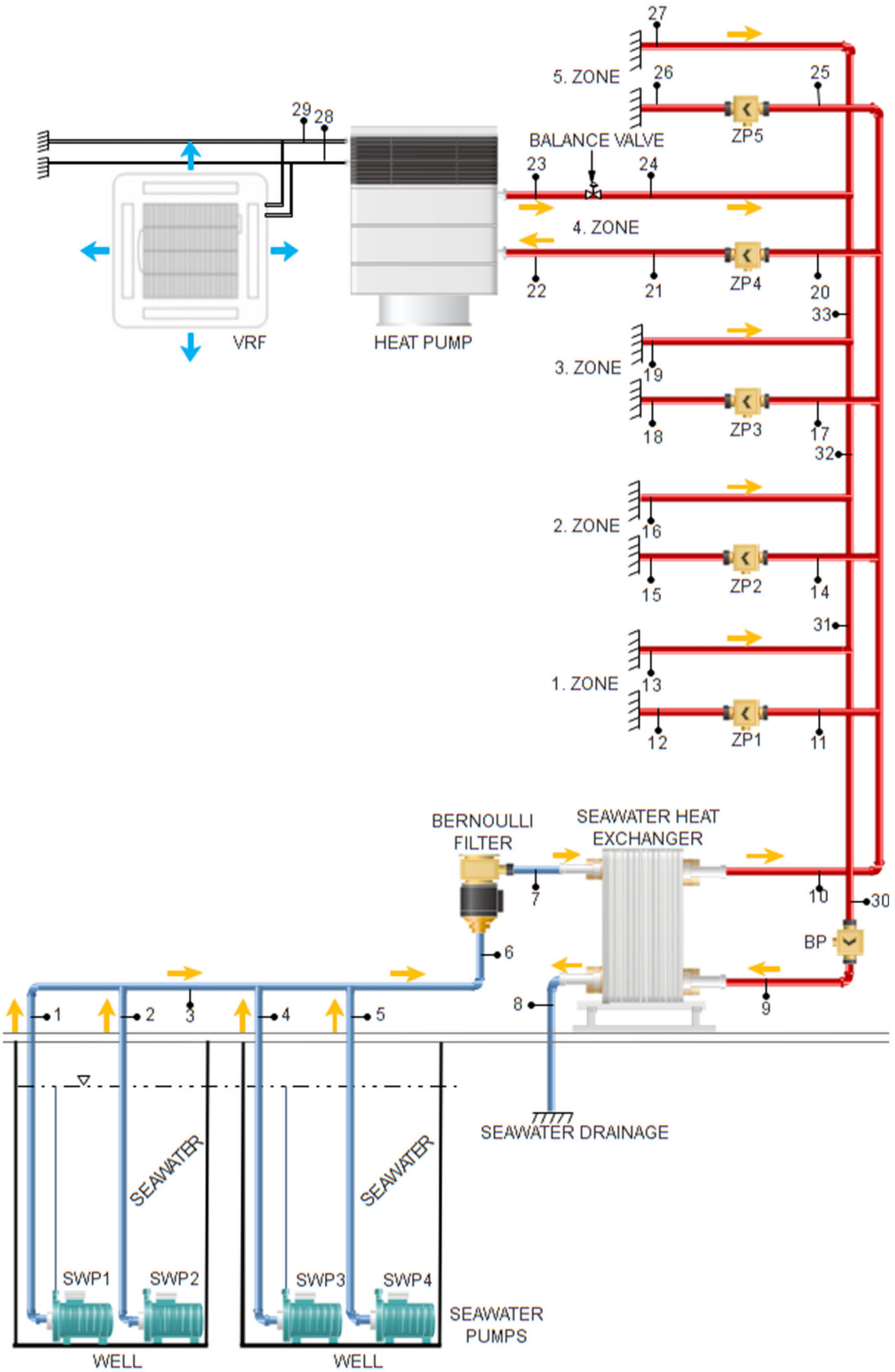

FIGURE 2 Flowchart of the hotel cooling system with seawater and heat pump [Color figure can be viewed at wileyonlinelibrary.com]

$$\varepsilon_k = \frac{\dot{Ex}_{\mathrm{out},k}}{\dot{Ex}_{\mathrm{in},k}} \tag{10}$$

The exergy efficiency for the whole system may be given below;

$$\varepsilon_{\mathrm{system}} = \frac{\dot{I}}{\dot{W}_{\mathrm{SWPs}} + \dot{W}_{\mathrm{BP}} + \dot{W}_{\mathrm{ZPs}} + \dot{W}_{\mathrm{HPs}} + \dot{W}_{\mathrm{Fans}}} \tag{11}$$

As mentioned above, using the mass, energy and exergy balances, the exergy balances, and exergy efficiencies for all equipment in the hotel cooling system are determined, and these are listed in Table 3.

Additionally, the following equation developed by Van Goal[22] is used to assess the improvement potentials of the whole system and its all equipment, as given below:

$$IP = (1-\varepsilon)\left(\dot{Ex}_{\mathrm{in}} - \dot{Ex}_{\mathrm{out}}\right) \tag{12}$$

For exergy analysis to assess the thermodynamic performance of the hotel cooling system, the following assumptions are made: (a) For SW, the thermodynamic properties of water are used. By doing so, any possible effects of salts and foreign matters that might be present

TABLE 2 Technical properties of the equipment in the hotel cooling system

Equipment (k)	Technical properties
Seawater well Pumps (SWPs)	Submersible pump, nominal pump power of 11 kW, volumetric flow rate of 95 m^3/hr, and pump pressure of 24 mSS
Bernoulli filter (BF)	DN300, volumetric flow rate of 380 m^3/hr, max. pressure drop of 0.25 bar, and filter precision of 200 μm
Seawater heat exchanger (SHE)	Plate heat exchanger from water to water, titanium, heat transfer capacity in 2,200 kW, counter-flow, logarithmic temperature difference of 4°C, pressure drop of 29.2 kPa, total heat transfer coefficient of 5,541 W/m^2K, and heat transfer area of 99.5 m^2
Booster pump (BP)	Nominal pump power of 9.3 kW, volumetric flow rate of 190 m^3/hr, and pump pressure of 14 mSS
Zone pumps (ZPs)	Nominal pump power of 3 kW, volumetric flow rate of 50 m^3/hr, and pump pressure of 12 mSS
Heat pump (HP)	Cooling capacity of 101 kW, power of 28.42 kW, and EER of 3.5
Variable refrigerant flow system (VRF)	Cooling capacity of 2.8 kW, power of 53 W, air flow rate of 500 m^3/hr, and R410A as refrigerant

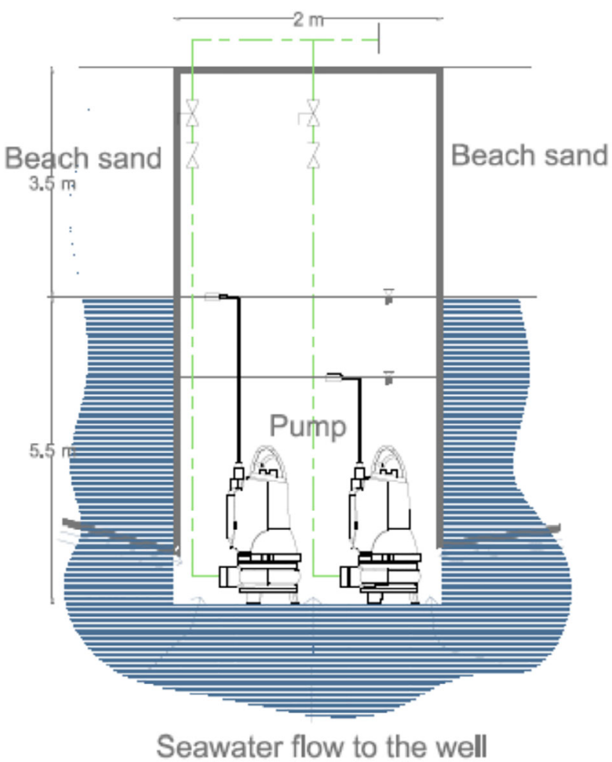

FIGURE 3 A general view of the hotel seawater well [Color figure can be viewed at wileyonlinelibrary.com]

in the SW are neglected. (b) Air-cooled VRF systems are commonly used. However, for efficient transfer of heat rejected from the indoor environment into SW, a water-cooled VRF system is

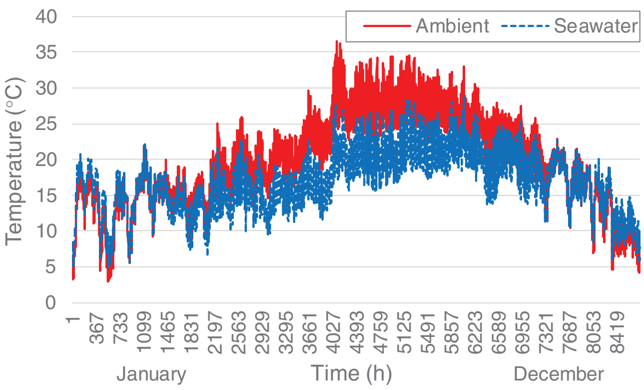

FIGURE 4 The average hourly ambient and seawater temperatures of the city of Antalya in Turkey for multi-annual periods (1970–2016)[20] [Color figure can be viewed at wileyonlinelibrary.com]

used in here. (c) In VRFs and HPs it is used R410A as cooling fluid. (d) The whole system is operated at full load under steady state, steady volume conditions. (e) All windows and doors in the hotel rooms are closed so infiltration is negligible. (f) The isentropic efficiencies of pumps and fans vary from 80 to 85% and 90 to 95%.

4 | RESULTS AND DISCUSSION

In this study, the thermodynamic performance of a cooling system with SW-HP of a hotel operated in reality for cooling processes is investigated. The study is designed as a case study using the Asia Beach Resort SPA hotel located in Alanya county of Antalya, Turkey. Thus, the exergy analysis for the whole hotel cooling system and detailed evaluations are performed. Furthermore, the cooling system is divided into five cooling zones and the fourth cooling zone among them is investigated. Additionally, the HP and VRF system used in a room selected within the fourth cooling zone are taken into account. From the state numbers given in Figure 2 on the hotel cooling system operating at full load, hourly data were collected in June, July, and August as of 2016.

Regard in Figure 4, which are trends of SW and ambient temperatures during the summer period, the average SW, and ambient temperatures of the system are ~20 and 27°C, respectively. The difference of these temperatures is 7°C. Furthermore, this difference is increasing from the beginning to the end of the season. During the summer season as hourly, the changes of the SW temperature and the COP of the hotel cooling system are drawn in Figure 5. From Figure 5, while the average values of SW temperature in July and August are close to each other (21 and 22°C, respectively), the value (18°C) in June are lower than their values. As observed in Figure 5, the COP curves show similar changes. The mean COP values for June, July, and August are found as 3.32, 3.75, and 3.91, respectively. With the increase in SW temperature, the COP of the hotel cooling system rises.

TABLE 3 Exergy balance and efficiency equations for the equipment of the hotel cooling system

Component (k)	Exergy balance	Exergy efficiency
Seawater well Pumps (SWPs)	$\dot{I}_{SWP1}=\dot{W}_{SWP1}-\left(\dot{E}x_1-\dot{E}x_{sw}\right)$ $\dot{I}_{SWP2}=\dot{W}_{SWP2}-\left(\dot{E}x_2-\dot{E}x_{sw}\right)$ $\dot{I}_{SWP3}=\dot{W}_{SWP3}-\left(\dot{E}x_4-\dot{E}x_{sw}\right)$ $\dot{I}_{SWP4}=\dot{W}_{SWP4}-\left(\dot{E}x_5-\dot{E}x_{sw}\right)$	$\varepsilon_{SWP1}=\frac{\dot{E}x_1-\dot{E}x_{sw}}{W_{SWP1}}$ $\varepsilon_{SWP2}=\frac{\dot{E}x_2-\dot{E}x_{sw}}{W_{SWP2}}$ $\varepsilon_{SWP3}=\frac{\dot{E}x_4-\dot{E}x_{sw}}{W_{SWP3}}$ $\varepsilon_{SWP4}=\frac{\dot{E}x_5-\dot{E}x_{sw}}{W_{SWP4}}$
Bernoulli filter (BF)	$\dot{I}_{BF}=\dot{E}x_6-\dot{E}x_7$	$\varepsilon_{BF}=\frac{\dot{E}x_7}{\dot{E}x_6}$
Seawater heat exchanger (SHE)	$\dot{I}_{SWHE}=\left(\dot{E}x_7-\dot{E}x_8\right)-\left(\dot{E}x_{10}-\dot{E}x_9\right)$	$\varepsilon_{SWHE}=\frac{\dot{E}x_{10}-\dot{E}x_9}{\dot{E}x_7-\dot{E}x_8}$
Booster pump (BP)	$\dot{I}_{BP1}=\dot{W}_{BP1}-\left(\dot{E}x_9-\dot{E}x_{30}\right)$	$\varepsilon_{HP}=\frac{\dot{E}x_9-\dot{E}x_{30}}{W_{BP1}}$
Zone pumps (ZPs)	$\dot{I}_{ZP1}=\dot{W}_{ZP1}-\left(\dot{E}x_{12}-\dot{E}x_{11}\right)$ $\dot{I}_{ZP2}=\dot{W}_{ZP2}-\left(\dot{E}x_{15}-\dot{E}x_{14}\right)$ $\dot{I}_{ZP3}=\dot{W}_{ZP3}-\left(\dot{E}x_{18}-\dot{E}x_{17}\right)$ $\dot{I}_{ZP4}=\dot{W}_{ZP4}-\left(\dot{E}x_{21}-\dot{E}x_{20}\right)$ $\dot{I}_{ZP5}=\dot{W}_{ZP5}-\left(\dot{E}x_{26}-\dot{E}x_{25}\right)$	$\varepsilon_{ZP1}=\frac{\dot{E}x_{12}-\dot{E}x_{11}}{W_{ZP1}}$ $\varepsilon_{ZP2}=\frac{\dot{E}x_{15}-\dot{E}x_{14}}{W_{ZP2}}$ $\varepsilon_{ZP3}=\frac{\dot{E}x_{18}-\dot{E}x_{17}}{W_{ZP3}}$ $\varepsilon_{ZP4}=\frac{\dot{E}x_{21}-\dot{E}x_{20}}{W_{ZP4}}$ $\varepsilon_{ZP5}=\frac{\dot{E}x_{26}-\dot{E}x_{25}}{W_{ZP5}}$
Heat pump (HP) for Zone 4	$\dot{I}_{HP}=\left(\dot{E}x_{22}-\dot{E}x_{23}\right)-\left(\dot{E}x_{29}-\dot{E}x_{28}\right)+\dot{W}_{HP}$	$\varepsilon_{BP1}=\frac{\dot{E}x_{29}-\dot{E}x_{28}}{W_{HP}+\left(\dot{E}x_{22}-\dot{E}x_{23}\right)}$
Variable refrigerant flow system (VRF) for Zone 4	$\dot{I}_{WRF}=\left(\dot{E}x_{29}-\dot{E}x_{28}\right)-\left(\dot{E}x_{34}-\dot{E}x_{35}\right)$	$\varepsilon_{VRF}=\frac{\dot{E}x_{34}-\dot{E}x_{35}}{\dot{E}x_{29}-\dot{E}x_{28}}$
Cooling zones (Zs)	$\dot{I}_{Z1}=Q_{Z1}\left(1-\frac{T_0}{T_{Z1}}\right)-\left(\dot{E}x_{12}-\dot{E}x_{13}\right)$ $\dot{I}_{Z2}=Q_{Z2}\left(1-\frac{T_0}{T_{Z2}}\right)-\left(\dot{E}x_{15}-\dot{E}x_{16}\right)$ $\dot{I}_{Z3}=Q_{Z3}\left(1-\frac{T_0}{T_{Z3}}\right)-\left(\dot{E}x_{18}-\dot{E}x_{19}\right)$ $\dot{I}_{Z4}=Q_{Z4}\left(1-\frac{T_0}{T_{Z4}}\right)-\left(\dot{E}x_{21}-\dot{E}x_{24}\right)$ $\dot{I}_{Z5}=Q_{Z5}\left(1-\frac{T_0}{T_{Z5}}\right)-\left(\dot{E}x_{26}-\dot{E}x_{27}\right)$	$\varepsilon_{Z1}=\frac{\dot{E}x_{12}-\dot{E}x_{13}}{Q_{Z1}\left(1-\frac{T}{T_{Z1}}\right)}$ $\varepsilon_{Z2}=\frac{\dot{E}x_{15}-\dot{E}x_{16}}{Q_{Z2}\left(1-\frac{T}{T_{Z2}}\right)}$ $\varepsilon_{Z3}=\frac{\dot{E}x_{18}-\dot{E}x_{19}}{Q_{Z3}\left(1-\frac{T}{T_{Z3}}\right)}$ $\varepsilon_{Z4}=\frac{\dot{E}x_{21}-\dot{E}x_{24}}{Q_{Z4}\left(1-\frac{T}{T_{Z4}}\right)}$ $\varepsilon_{Z5}=\frac{\dot{E}x_{26}-\dot{E}x_{27}}{Q_{Z5}\left(1-\frac{T}{T_{Z5}}\right)}$

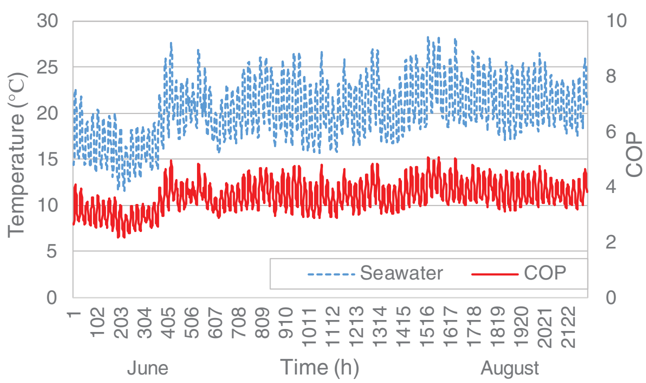

FIGURE 5 Changes in seawater temperature and COP for the hotel cooling system during the summer as of 2016. COP, coefficient of performance [Color figure can be viewed at wileyonlinelibrary.com]

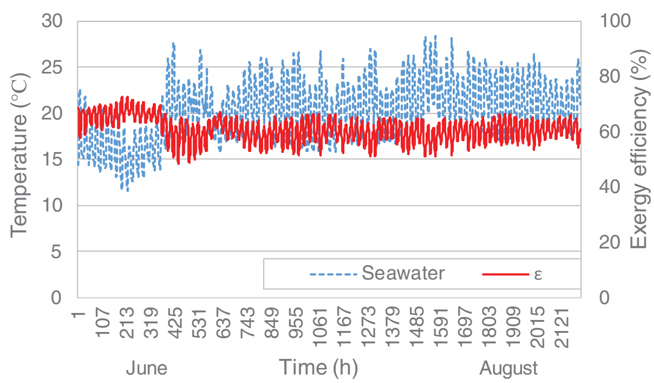

FIGURE 6 Changes in seawater temperature and exergy efficiency for the hotel cooling system during the summer as of 2016 [Color figure can be viewed at wileyonlinelibrary.com]

The exergy analysis is used to assess the thermodynamic performance of the system for long term. It is performed with the Engineering Equation Solver program using the data collected consistently from the system and the equations listed in Table 3. Considering Figure 5 regarding SW temperature change, Figure 6 shows the change in exergy efficiency (ε) of the hotel cooling system during the summer. As shown in Figure 6, from the beginning of the season to the end, the SW and ambient temperatures increase, while the exergy yield of the system decreases. The average exergy efficiencies of the system for June, July, and August are 68, 65, and 64%, respectively.

TABLE 4 Thermodynamic parameters and exergy rates for each state in the hotel cooling system from hourly average data of June, July, and August months as of 2016

State no, i	Fluid type	Temperature, T_i (°C)	Pressure, P_i (kPa)	Enthalpy, h_i (kj/kg)	Entropy, s_i (kj/kg K)	Mass flow rate, $\dot{m}_i$ (kg/s)	Exergy rate, $\dot{E}x_i$ (kW)
1	Seawater	19.5	120	81.85	0.289	26.39	23.04
2	Seawater	19.5	120	81.85	0.289	26.39	23.04
3	Seawater	19.5	120	81.85	0.289	52.78	46.07
4	Seawater	19.5	120	81.85	0.289	26.39	23.04
5	Seawater	19.5	120	81.85	0.289	26.39	23.04
6	Seawater	19.5	120	81.85	0.289	105.56	92.14
7	Seawater	19.5	60	81.8	0.289	105.56	85.78
8	Seawater	20.5	30	85.95	0.3033	105.56	61.29
9	Cooling water	23.6	540	99.46	0.3473	105.56	81.5
10	Cooling water	22.3	480	93.9	0.3288	105.56	89.93
11	Cooling water	22.3	400	93.83	0.3288	20	15.43
12	Cooling water	22.3	520	93.98	0.3289	20	17.82
13	Cooling water	24.3	400	102.2	0.357	20	11.37
14	Cooling water	22.3	400	93.83	0.3288	23.33	18
15	Cooling water	22.3	502	93.97	0.3289	23.33	20.36
16	Cooling water	24.9	400	104.7	0.3654	23.33	12.1
17	Cooling water	22.3	400	93.83	0.3288	23.33	15.43
18	Cooling water	22.3	530	93.99	0.3289	20	18.02
19	Cooling water	23.3	400	98.01	0.3429	20	13.26
20	Cooling water	22.3	400	93.83	0.3288	20	15.43
21	Cooling water	22.3	520	93.98	0.3289	20	17.82
22	Cooling water	22.4	520	94.36	0.3302	2	1.761
23	Cooling water	24	400	100.9	0.3528	2	1.19
24	Cooling water	22.8	400	95.92	0.3359	20	14.31
25	Cooling water	22.3	400	93.83	0.3288	22.23	17.15
26	Cooling water	22.3	520	93.98	0.3289	22.23	19.8
27	Cooling water	22.1	400	92.99	0.326	22.23	17.67
28	R410A	0	750	282.9	1.06	0.00489	0.03349
29	R410A	11	17,500	90.18	0.2932	0.0231	0.2507
30	Cooling water	23.6	400	99.26	0.3472	105.56	66.81
31	Cooling water	23.6	400	99.26	0.3472	85.56	54.15
32	Cooling water	23.6	400	99.26	0.3472	62.23	39.39
33	Cooling water	23.6	400	99.26	0.3472	42.23	26.73
34	Air	27.6	101	301.2	5.704	0.1908	−271.9
35	Air	21.1	101	294.6	5.682	0.191	−272.1

The thermodynamic parameters for each state of the system from hourly average data of 3 months in 2016 and their exergy rates are listed in Table 4. During this time, the hotel cooling system runs at full load. Data collected on different times from the system need to be verified. Therefore, a sensitivity analysis is performed, and the sensitivities of the temperature, pressure, and volumetric flow rate listed in Table 4 are ±1.8, ±0.7, and ± 2.9%, respectively. On the basis of these values, the error ratio found for the exergy rate is ±0.51%. As seen in Table 4, when the SW temperature is 19.5°C, the temperature and humidity of the cooled room are measured as 21.1°C and 52%, respectively. The temperature difference of the SHE and VRF system is 1 and 4.5°C, respectively. As a result, the COP of the HP is calculated as 2.94. During this time there is ~92.2 kW of exergy input rate from the SW to the system. Of this amount of exergy input rate, nearly 66.5% (61.3 kW) is SW heated by the SHE, and it discharges back into the sea. The remaining 30.9 kW is the exergy destruction rate occurred in the system equipment.

In the hotel cooling zones, there is an exergy demand rate of 58 kW. Of the exergy demand rate, 57% are exergy loss rate occurring in the zones and the remaining 43% is exergy output rate produced by the cooling system for the zones in average 25.1 kW. Thus, changes in exergy loss rate and exergy efficiency for each of the cooling zones in the system are shown in Figure 7. From Figure 7, the highest exergy loss rate among the cooling zones in the hotel occurs in the fifth zone for 8.53 kW with 20% of exergy yield. This is followed by the first, third, second, and fourth zones. The reason that the highest exergy loss rate in the hotel occurs in the fifth zone is that this has the longest pipeline. And insulation problems cause such high exergy loss rates. The data used in the calculation are collected during cooling of all spaces/rooms in the hotel by the system operated at full capacity. Therefore, the fourth zone at 4.97 kW with 41% of exergy yield is the region with the lowest exergy loss rate. The cooling process using a HP and VRF system are selected for a room in this zone. As seen in Figure 7, the zone with the highest exergy efficiency is the second region with 58%. Then, it is ranked as by the first zone with 45% and the third zone with 43%. The second zone is one of the cooling zones with the highest exergy yield. The reason for this is that the zone where the useful exergy rate is the most in the exergy input rate entering the zone. It shows that the cooling requirements in this zone are provided in the best manner. It can be clearly observed from Figure 7, that the fifth zone has priority for improvement among all the zones.

Change in exergy destruction rate and exergy efficiency for all equipment in the hotel cooling system are given in Figure 8. The total exergy destruction rate of system equipment included in the account is 30.9 kW. The highest exergy destruction rate among equipment is determined as the SHE with 10.6 kW for 34% of total exergy destruction rate. This is followed by the BF with 6.4 kW (20% of its) and the BP on the secondary cycle loop with 5.7 kW (18% of its). The equipment with the lowest exergy destruction rate is the VRF system with 0.02 kW. The exergy destruction rate of the HP is also 0.35 kW. Additionally, there is average 0.73 kW of exergy destruction rate for each of the ZPs used in the cooling zones. The SHE is the priority equipment requiring improvement among all the system equipment. The problems due to the BF responsible for catching foreign matter (weeds, silt, mussels, etc.) entering the SHE with the SW have been effective. This problem is due to large variations in pressure during filtration. Therefore, it causes both equipment to have the highest exergy destruction rate. Regarding Figure 8, although the exergy destruction rate of the BF is high, it has the highest exergy efficiency with 93% among system equipment. Thus, the exergy rate entering the BF provides the highest exergy destruction rate with 6.36 kW. Due to the high thermodynamic performance of the BF, the improvement priority comes after the BP. On the other hand, although the exergy destruction rate of the VRF system is very small, its exergy yield is 92%. The SHE has the highest amount of exergy destruction. However, since it can only produce 44% of the exergy rate entering SHE, it has the lowest performance among the equipment. The other equipment with the lowest exergy destruction rate is the HP and its exergy yield is 38%.

Using the Equation (12) given in Reference 22 the potentials for improvement of hotel's cooling system equipment are calculated and presented in Figure 9. The results obtained for the BF on Figure 8 show that the potential for improvement of the BF has as low as 0.4 kW. However, the potential for improvement of the SHE is

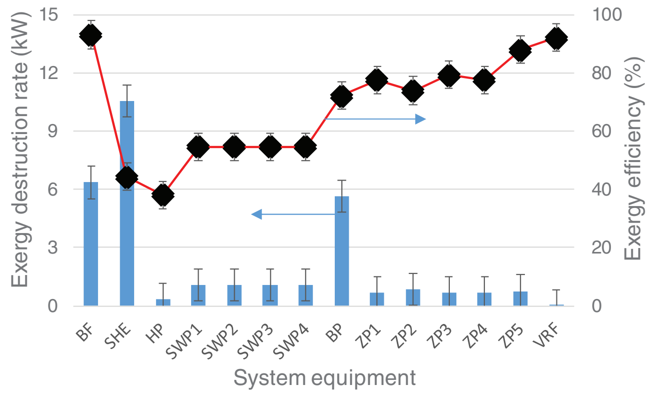

FIGURE 8 Changes in exergy destruction rate and exergy yield for the equipment of the hotel cooling system on average for June, July, and August as of 2016 [Color figure can be viewed at wileyonlinelibrary.com]

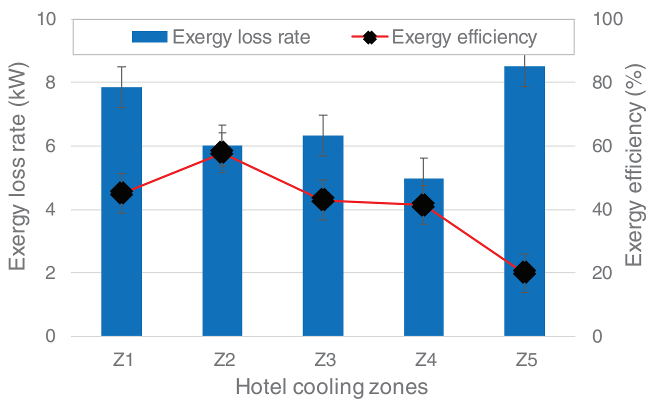

FIGURE 7 Changes in exergy loss rate and exergy yield for the cooling zones of the hotel on average for June, July, and August as of 2016 [Color figure can be viewed at wileyonlinelibrary.com]

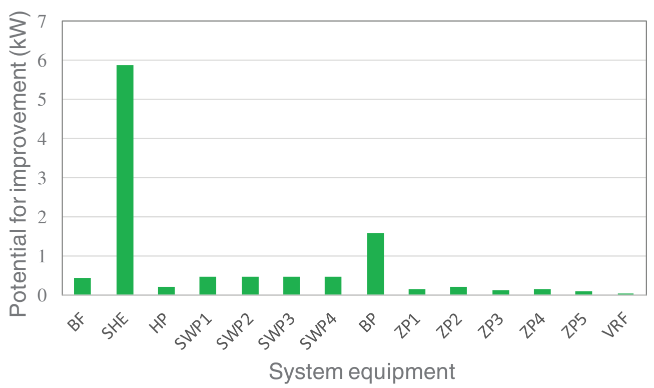

FIGURE 9 Change in potential for improvement of the equipment in the hotel cooling system on average for June, July, and August as of 2016 [Color figure can be viewed at wileyonlinelibrary.com]

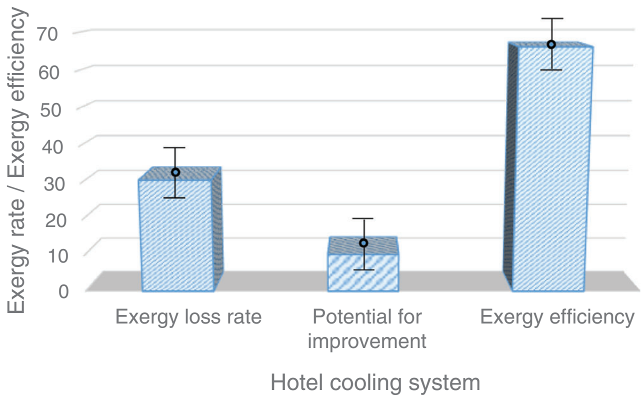

FIGURE 10 The results of exergy analysis for the hotel cooling system on average for June, July, and August as of 2016 [Color figure can be viewed at wileyonlinelibrary.com]

determined as 5.9 kW. This value indicates the highest improvement possible for equipment mentioned. If, of the exergy destruction rate in the SHE with 10.57, 5.9 kW may be improved; therefore, its exergy efficiency can be increased from 44 to 75%. The equipment with the next highest potential for improvement is the BP with 1.58 kW. Therefore, the number of pumps used can be increased. As can be seen in Figure 9, compared to the ZPs of the cooling zones, the SWPs have greater potential for improvement. Finally, it is understood that the best operating equipment is the VRF system.

Figure 10 summarizes the exergy analysis results performed for the hotel cooling system dealt with in this study. Of the SW exergy rate entering the hotel cooling system with 92.2 kW, it comprises the exergy destruction rate of 30.9 kW caused by the equipment. Thus, the exergy yield of the whole system is determined as 66.5%. Improvement strategies for 30.9 kW of the exergy destruction rate resulting from the equipment of the entire system will be improved by 34%, reducing the exergy destruction rate of 10.4 kW. This value is called as the potential for improvement of the whole system according to Van Gool.[22] Thus consequently, the exergy efficiency of the whole system will reach 77% through improvement strategies.

5 | CONCLUSION

In this study, the thermodynamic performance of a hotel cooling system with SW-HP operating in the Asia Beach Resort Spa Hotel located in Alanya county, Antalya/Turkey is investigated. This system was operated at full load for the purpose of cooling the hotel with 360 rooms in the June, July, and August as of 2016. This study uses average hourly data collected from June to August of the summer season in 2016. Exergy analysis is applied to it in order to evaluate the thermodynamic performance of the hotel cooling system during this time. Since a cooling system of the hotel under consideration has hundreds of rooms involving much equipment and a discussion of it would not be beneficial to a reader, the details of the cooling system and of the calculations are omitted in here. In order to avoid boring the readers with the complex details of the hotel and its cooling system and very detailed calculations, the significant topics are addressed in this study. As the hotel cooling system is quite involved it is also divided into five different cooling zones. The cooling process using a HP and VRF system is dealt with for a chosen room in the fourth cooling zone. The general results of the study show that the COP values for the whole cooling system and HP used with the aim of cooling a hotel with a SW-HP are calculated as 4.37 and 2.94, respectively. The highest exergy destruction rate among cooling zones in the hotel occurs in the fifth zone with 8.53 kW. The fourth zone has the lowest exergy destruction rate with 4.97 kW. Therefore, the cooling process with the HP and VRF is chosen for a room in the fourth zone. The exergy destruction rate for all system equipment is 30.9 kW. The analysis results show that the highest exergy destruction rate is from the SHE, BF, and BP of the secondary cycle loop. Their exergy destruction rates are 10.57, 6.36, and 5.68 kW, respectively. Thus, the equipment with priority for improvement are ranked. When ranked according to the highest exergy efficiency, it is the BF, VRF system, and BP of the secondary cycle loop. Their values are found as 93, 92, and 82%, respectively. The total exergy yield of the hotel cooling system is 66.5%. The best operated equipment is determined to be the VRF system. The equipment with the highest potential for improvement is the SHE. All improvement strategies conducted for the system should be focused on the SHE. To reduce pressure losses in the SHE, precautions should be taken based on flow regulation or construction and material choice, and additionally flow imbalances should be corrected. In hotels, which mainly use air-sourced cooling systems, selection of SW-sourced cooling systems as a renewable energy source is important in terms of physical, economic, and environmental aspects. In a hotel cooling system with SW-HP, this is due to the low temperature difference between the heat source (hotel room) and heat sink (SW). Thus, it is necessary to increase the use of such systems and to develop the technological aspects.

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NOMENCLATURE

e	specific flow exergy (kJ/kg)
$\dot{E}$	energy rate (kW)
$\dot{E}x$	exergy rate (kW)
h	specific enthalpy (kJ/kg)
$\dot{I}$	exergy destruction rate (kW)
IP	potential for improvement (kW)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (kPa)
$\dot{Q}$	heat transfer rate (kW)
s	specific entropy (kJ/kg K)

T temperature (°C or K)

$\dot{W}$ work rate (kW)

GREEK SYMBOLS

ε exergy efficiency (%)

SUBSCRIPTS

BP booster pump

cl cooling load

HP heat pump

i successive number of elements

k component/location

SW seawater

SWP seawater pump

ZP zone pump

0 reference state

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