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Full Length Research Paper

Integration of reverse osmosis and refrigeration systems for energy efficient seawater desalination

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Reverse osmosis (RO) systems have minimum energy consumption compared to other desalination methods and their energy consumption is only in the form of electric power. However, in addition to fresh water, many buildings and even many factories and industries also need air conditioning, which requires large amounts of energy. The aim of this study was to introduce a new method for combined cold and fresh water (CCW) generation by integration of a RO and a refrigeration system. In this new integrated system, the electrical power consumption of the pumps in the RO system was conserved by preheating the seawater through the waste heat recovery in the condenser of the refrigeration cycle. A further reduction in the electrical power demand of these pumps can be made by replacement of the expansion valve in the refrigeration cycle with a turbo expander. Related coding in energy efficient seawater (EES) software was used to study the feasibility study of the integration of the RO and refrigeration systems to co-generate 10 M^3/day fresh water and 3 refrigeration tons of cooling as an illustrative example and the best state for this combined system was determined. The results confirmed that the proposed method provides a reduction of about 7.6% of the power consumption used in the separate production of fresh water and cooling.

Key words: Reverse osmosis (RO), compression refrigeration cycle, seawater desalination, cogeneration, integration, energy saving.

INTRODUCTION

The earth contains a vast amount of water, but much of it is too salty for human use without advanced treatment. Nearly all of the earth's water is found in the world's oceans, while only about 2.5% exists as fresh water (Xie et al., 2009). At present, about 40% of the world's population is suffering from serious water shortage. By the year 2025, this percentage is expected to increase to more than 60% (Kilic and Kilic, 2010).

One of the best solutions to overcome water shortage problems in the coming decade is the desalination of sea and brackish waters (Ataei et al., 2009). Desalination is a water treatment process that removes salts or other dissolved minerals and contaminants such as dissolved metals, radio-nuclides, bacteria, and organic matter from high salinity water to produce fresh water (Greenlee et al., 2009; Khawaji et al., 2008; Medugu and Ndatuwong, 2009; Nkwonta and Ochieng, 2009; Yoo et al., 2010). Desalination processes fall into following main categories:

1. Thermal processes: Multi-effect distillation (MED), multi-stage flash (MSF) and Mechanical Vapor Compression (MVC).
2. Membrane processes: Reverse osmosis (RO) and electro dialysis (ED).

The market share of RO desalination was 43% in 2004 and is forecasted to increase up to 61% in 2015. This is because RO has many advantages including its low energy requirement, low operating temperature, small

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footprint, modular design, and low water production costs. Water desalination by the RO technique has proved to be the lowest energy consuming technique compared to other desalination processes (Alghoul et al., 2009; Ataei et al., 2009; Greenlee et al., 2009; Kim et al., 2009; Verhuelsdonk and Attenborough, 2010).

Several projects have been undertaken throughout the last century that are aimed at improving the performance of RO systems. Traube (1867), using information provided by the work of Graham (1862), was apparently the first investigator to prepare artificial membranes and demonstrated that they were capable of discriminating between water molecules and small solute molecules dissolved therein. This was a significant improvement in Dutrochet's concept of osmosis and was later supported by Wilhelm Pfeffer's accurate measurements of osmotic pressure using membranes described by Traube (Pfeffer, 1877). Ohya and Sourirajan (1969) introduced the first asymmetric cellulose acetate membrane with high salt rejection and water flux. Redondo (1997) improved upon this technique with a new FILMTEC RO membrane and Van Wagner et al. (2009) studied effects of feed pH on commercial RO performance. In the RO method, water moves from an area of higher concentration to an area of lower concentration. For seawater, the salt water is pumped into a closed vessel where it is pressurized against the membrane. Most of the energy required by this process is consumed due in the initial pressurization by the pump. Therefore, modern RO systems are built with energy recovery turbines (ERT) (Marcel et al., 2010; Oh et al., 2009; Yu et al., 2010). In addition, a small increase in the temperature of inlet seawater to the RO system can reduce the intensity of power consumption (Xue et al., 2009; Yu et al. 2010).

Many buildings and even many factories and industries require power and fresh water, but also need a cooling source. General Electric Company introduced the first electrical refrigerator in 1911, followed by Frigidaire in 1915. Eastman Kodak installed the first air conditioning system in 1891 in Rochester. The concept of a vapor-compression refrigeration cycle relies on heat absorption from a cold source and its repulsion to a hot source via a refrigerant. The refrigerant is compressed by a compressor, which consumes energy consumption and then is expanded by an expansion valve. A refrigerator can be adapted with a turbo expander that replaces the expansion valve to generate mechanical shaft work or electrical power for increasing the global efficiency of the system (Cho et al., 2008; Dientrich and Thummes, 2010; Subiantoro and Ooi, 2010).

Ordinarily, desalination systems, power generator units, and refrigeration cycles separately supply the fresh water, power, and cold demand, respectively, of many buildings and industrial plants. Because of the high energy consumption in these systems, much attention is now being paid to issues of fresh water and power cogeneration (Cgacartegui et al., 2009; Ferreire et al., 2010; Kosmadakis et al., 2010). However, little attention has yet been focused on simultaneous desalination and cold generation, even though this cogeneration may reduce costs and energy consumption in many buildings and industrial plants in some regions, such as the Middle East and North Africa.

The objective of the present study was to introduce a new method for combined cold and fresh water generation (CCW) by integration of a refrigeration cycle and a RO system. For increasing energy conservation, in addition to preheating the seawater through the waste heat recovery in the condenser of the refrigeration cycle, the expansion valve of refrigeration cycle is replaced with a turbo expander. This technique, called the Integrated Reverse Osmosis Cooling System (IROCS), accounts for the maximum energy conservation at the minimum cost.

An IROCS to generate $10M^3$/day fresh water and 3 refrigeration tons of cooling was considered in this study as an illustrative example and was encoded using EES software to obtain optimal values in the IROCS design method computations.

MATERIALS AND METHODS

Reverse osmosis (RO) system

In this study, a single-stage RO system is used, as shown in Figure 1 and Table 1. In the RO method, the seawater is pumped into the membranes, which are inserted in cylindrical pressure vessels. The water molecules pass through the membrane, increasing the solute concentration of the reject water while producing purified water on the other side. Salts or other dissolved minerals and contaminants such as dissolved metals, radio-nuclides, bacteria, and organic matter are removed, depending on the type of membrane used. Most of the energy is consumed by the pump for the initial pressurization. Therefore, the RO systems are built with energy recovery turbines. These turbines can be installed in pumps for direct assistance with pumping and generating an efficient desalination process for highly saline seawater. The energy recovery turbine can reduce the amount of energy used by the pump by as much as one-third (Oh et al., 2009; Marcel et al., 2010; Yu et al, 2010).

The power consumed by the pump is calculated by Equation (1):

$$\dot{E}_{pump} = \frac{q_{F.W.} \cdot \rho_{pump} \cdot g \cdot h_{pump}}{24 \times 3.6 \times 10^{6} . \eta_{pump} . b} \tag{1}$$

The power produced by the energy recovery turbine is given by Equation (2) (Ataei and Yoo, 2010):

$$\dot{E}_{E.R.T.} = \frac{q_{F.W.} . h_{E.R.T.} . g . \rho_{E.R.T} . \eta_{E.R.T.}}{86400 \times 10^{3}} \left(\frac{1-b}{b}\right) \tag{2}$$

In Equations 1 and 2, flow rate unit is m^3/day and power unit is kW. The power required by the RO system is given by Equation (3):

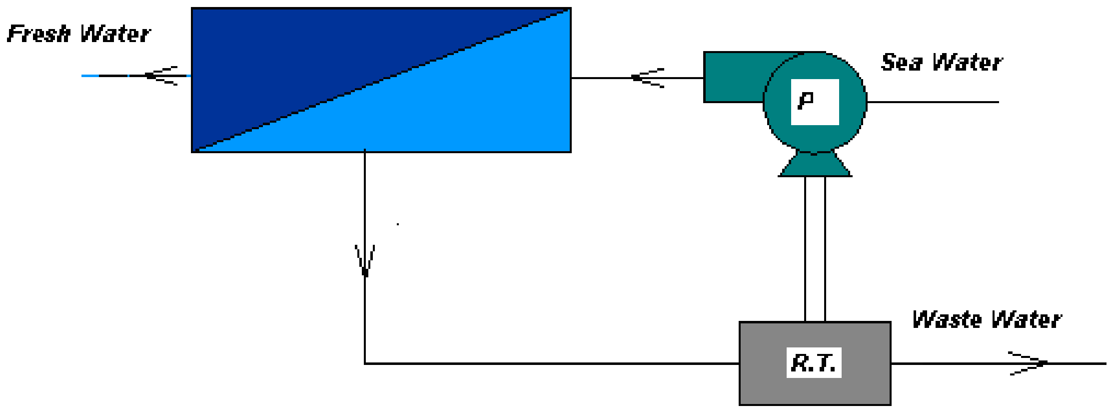

Figure 1. Structure of a single-stage RO system with an energy recovery turbine.

Table 1. Characteristics of the RO system used in this paper.

g (m/s^2)	h_{pump} (m)	b (%)	$\eta_{E.T.}(\%)$	$\eta_{com}(\%)$
9.81	500	40	85	85

$$E^{\cdot}_{RO} = E^{\cdot}_{pump} - E^{\cdot}_{E.R.T.} \quad (3)$$

The power required by the RO system, according to Table 1, is given by Equation (4):

$$E^{\cdot}_{RO} = \frac{q_{F.W.}.\rho_{pump}.g.h_{pump}}{34.56\times 10^{6}.\eta_{pump}} - \frac{q_{F.W.}.\rho_{E.R.T.}.h_{E.R.T.}.g.\eta_{E.R.T.}.(1.5)}{86400\times 10^{3}} \quad (4)$$

With regard to Equation (4), $E^{\cdot}_{RO}$ has a direct relationship to water density. Due to the use of a membrane in the RO method, this method can be used to desalinate seawater with a thermal range from 0 to 45ºC. The values of water density at different temperatures are shown in Figure 2.

By increasing seawater temperature, the energy saving percentage in the RO system is given by Eq. (5):

$$E.S.(\%) = \frac{E^{\cdot}_{(RO)T_{s.w.}} - E^{\cdot}_{(RO)T}}{E^{\cdot}_{(RO)T_{s.w.}}} \times 100 \quad (5)$$

The values of the energy saving percentage in the RO system versus the preheated seawater temperature, for several seawater temperatures, are shown in Figure 3.

As seen in Figure 3, the maximum energy savings occurs when the seawater is preheated to 45ºC.

Refrigeration system

Refrigeration is defined as "the transfer of heat from a lower temperature region to a region of higher temperature". Devices that produce refrigeration operate using a vapor-compression cycle. The base work of the vapor-compression refrigeration cycle is heat absorption from a cold source and its repulsion to a hot source via a refrigerant. This is done by compressing a refrigerant with energy consumption and then expanding it by an expansion valve. Refrigeration cycles can adopt a turbo expander that replaces the expansion valve in the expansion process to carry out mechanical work or to produce electrical power, thus increasing the global efficiency of the system (Dientrich and Thummes, 2010; Cho et al., 2008; Subiantoro and Ooi, 2010). A Vapor-Compression refrigeration cycle is shown in Figure 4 (Dincer and Kanogolu, 2010). Characteristics of the refrigeration cycle used in the current study are presented in Table 2.

By considering the useable thermal range of seawater in the RO system, the thermal range T_b is given by Equation (6):

$$20^{\circ}C < T_b < 65^{\circ}C \quad (6)$$

The compressor power required is given by Equation (7) (Enweremadu et al., 2008):

$$E^{\cdot}_{com} = m^{\cdot}_{R}(h_b - h_a) \quad (7)$$

The power produced by the turbo expander is given by Equation (8) (Ataei and Yoo, 2010):

$$E^{\cdot}_{E.T.} = m^{\cdot}_{R}(h_c - h_d) \quad (8)$$

The power consumed by the refrigeration cycle is given by Equation (9):

$$E^{\cdot}_{ref.} = E^{\cdot}_{com} - E^{\cdot}_{E.T.} \quad (9)$$

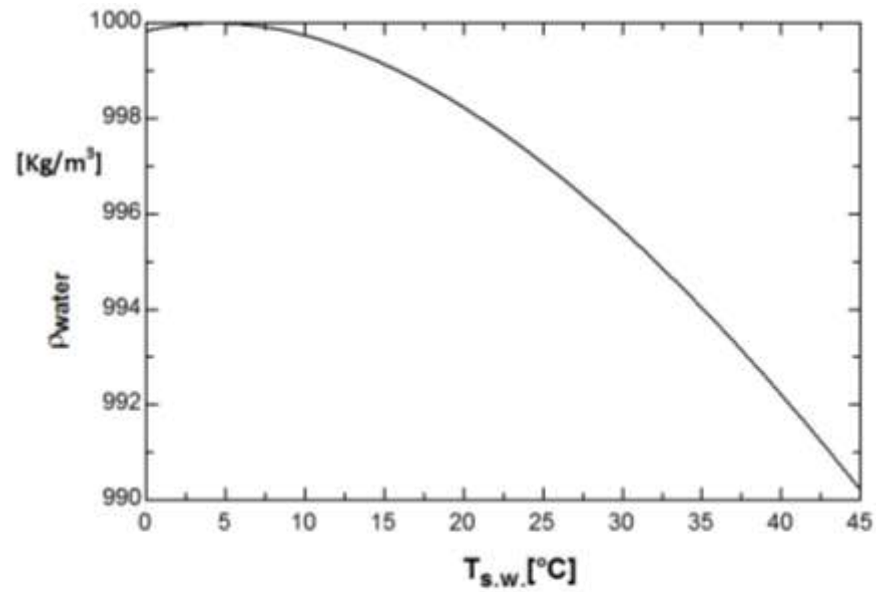

Figure 2. Water density vs. temperature.

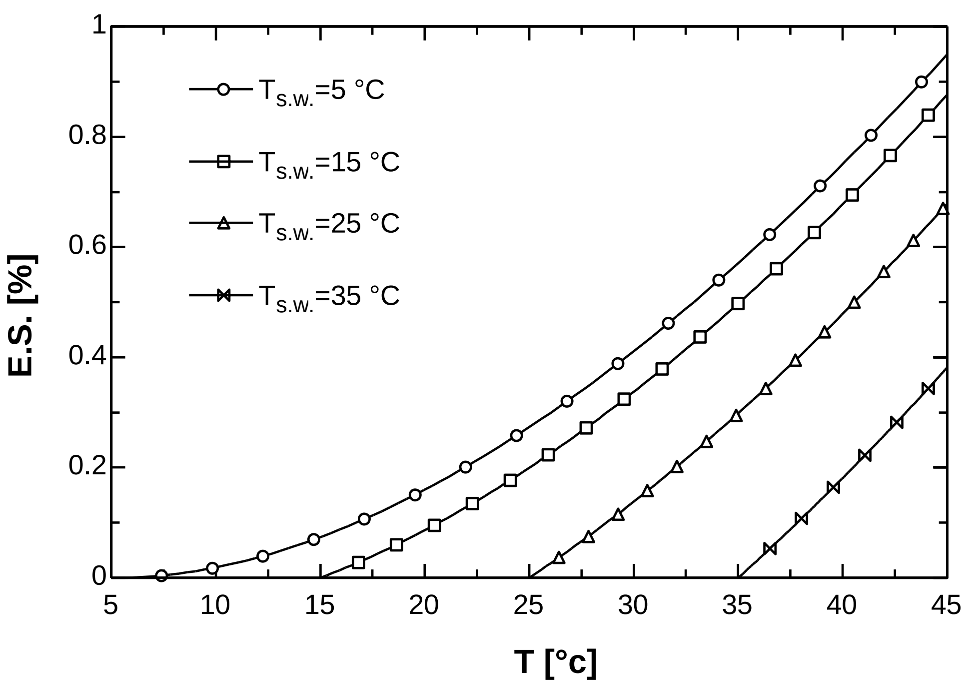

Figure 3. Energy saving percentage in the RO system vs. preheated seawater temperature.

The values of power consumed by the refrigeration cycle versus the turbo expander efficiency (in a sample operation condition) is shown in Figure 5.

As seen in Figure 5, when the turbo expander efficiency is zero (which means an expansion valve is substituted for the turbo expander), the energy consumption of the refrigeration system is maximal, but can be decreased by increasing the turbo expander efficiency.

Integration of the RO and refrigeration systems

With a slight increase in the temperature of the inlet seawater to the RO system, the intensity of power consumption can be reduced. By integrating this system and the refrigeration cycle, in addition to the possibility of providing a temperature increase for the inlet seawater in the desalination system, some shaft work needed by this system could also be recovered by replacement of the expansion valve in

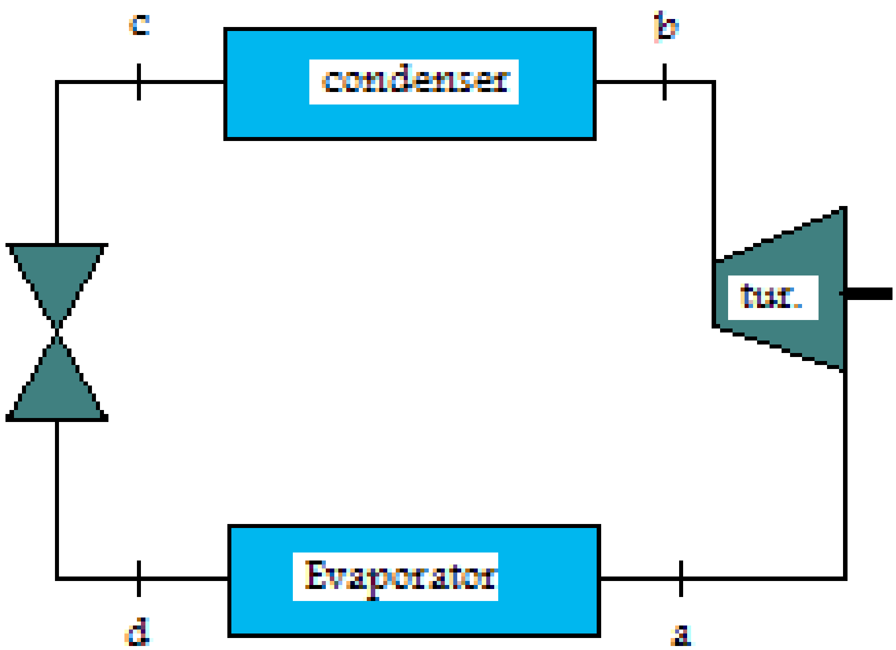

Figure 4. A Vapor-Compression refrigeration cycle.

Table 2. Characteristics of the refrigeration cycle used in this paper.

Refrigerant	$\eta_{E.T.}(\%)$	$\eta_{com}(\%)$
R-12	85	85

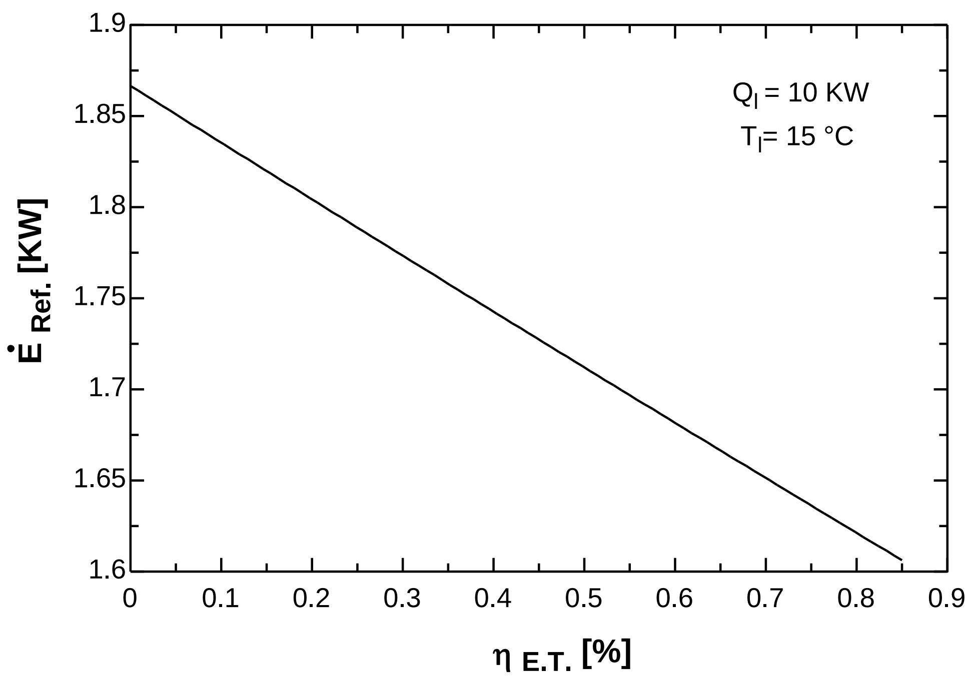

Figure 5. Power consumed by the refrigeration cycle versus the turbo expander efficiency.

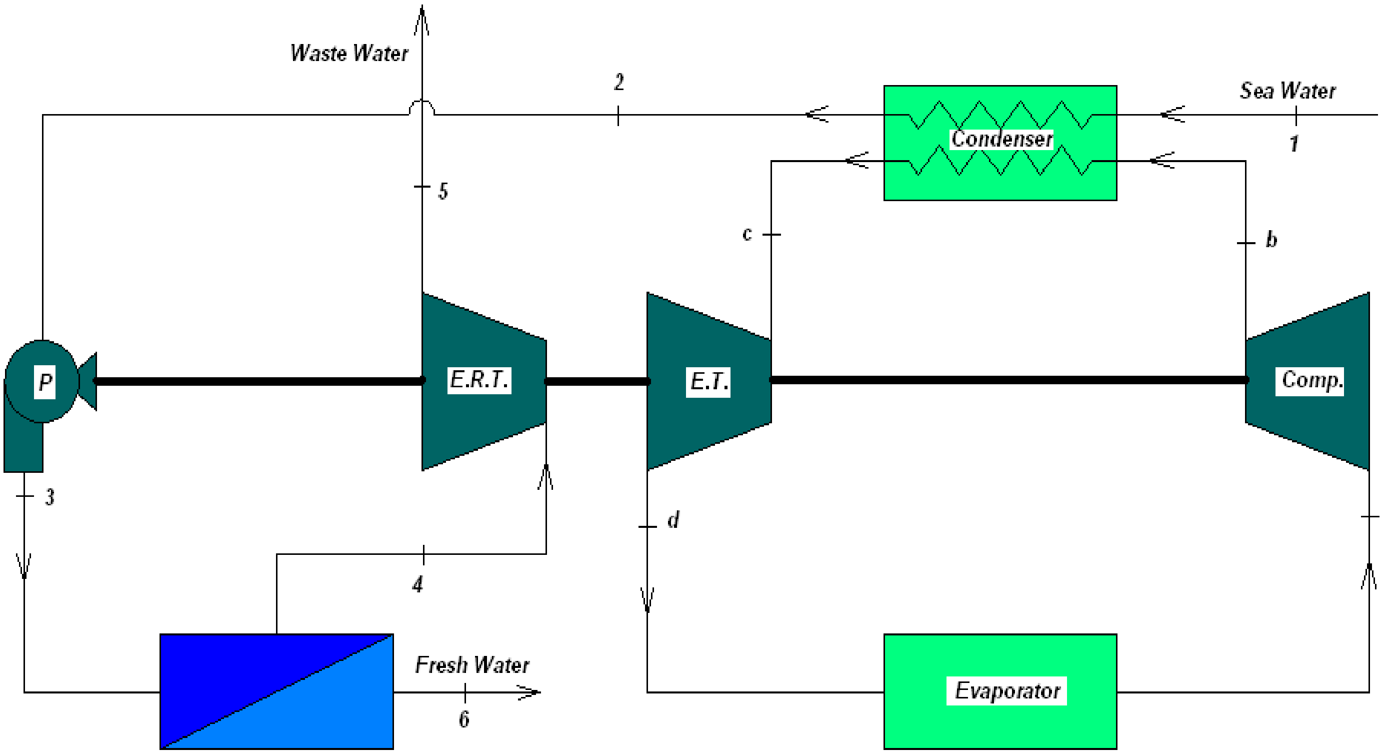

Figure 6. Structure of the combined cold and water (CCW) System.

the refrigeration cycle with a turbo expander.

Therefore, the combined cold and water (CCW) Generation resulting from the integration of the RO and refrigeration systems affords energy savings in comparison to running the two systems separately.

For this purpose, the RO and refrigeration systems are integrated according to the new system presented in Figure 6.

Calculation of heat rate required for the RO system in the integrated system

The heat rate required by the RO system to preheat seawater flow up to temperature T_2 by considering the fresh water flow rate is given by Equation (10):

$$\dot{Q}_{h(RO)}=q_{F.W.}.\rho_{s.w.}.(h_2-h_1)/b \qquad (10)$$

The values of the heat rate required versus the seawater preheating temperature for several seawater temperatures, taking into consideration the fresh water flow rate, are shown in Figure 7.

Calculation of waste heat rate of the refrigeration system in the integrated system

Waste heat rate of the refrigeration system is given by Equation (11):

$$\dot{Q}_{h(ref.)}=\dot{m}_R(h_b-h_c) \qquad (11)$$

The values of the waste heat rate versus the compressor outlet temperature for several cold environment temperatures, by considering the cooling load, are shown in Figure 8.

Calculation of the energy savings in the integrated system

The value of the power consumed in the integrated system is given by Equation. (12):

$$\dot{E}_{Integrated}=(\dot{E}_{RO}+\dot{E}_{ref})_{Integtared} \qquad (12)$$

Equation 13 is obtained by combining Equations. 4, 9, and 12:

$$\dot{E}_{Integrated}=[(\frac{q_{F.W.}.\rho_{pump}.g.h_{pump}}{34.56\times10^6.\eta_{pump}}-\frac{q_{F.W.}.\rho_{E.R.T.}.h_{E.R.T.}.g.\eta_{E.R.T.}.(1.5)}{86400\times10^3})+\dot{m}_R(h_b-h_a-h_c+h_d)]_{Integrated} \qquad (13)$$

The total power consumed when the systems operate separately (the refrigeration cycle is without turbo expander) is given by Equation (14):

$$\dot{E}_{seprate}=\dot{E}_{RO}+\dot{E}_{ref.} \qquad (14)$$

The percentage of the energy savings by the integrated system is calculated by Equation (15):

$$E.S.(\%)=\frac{\dot{E}_{separate}-\dot{E}_{Integrated}}{\dot{E}_{separate}}\times100 \qquad (15)$$

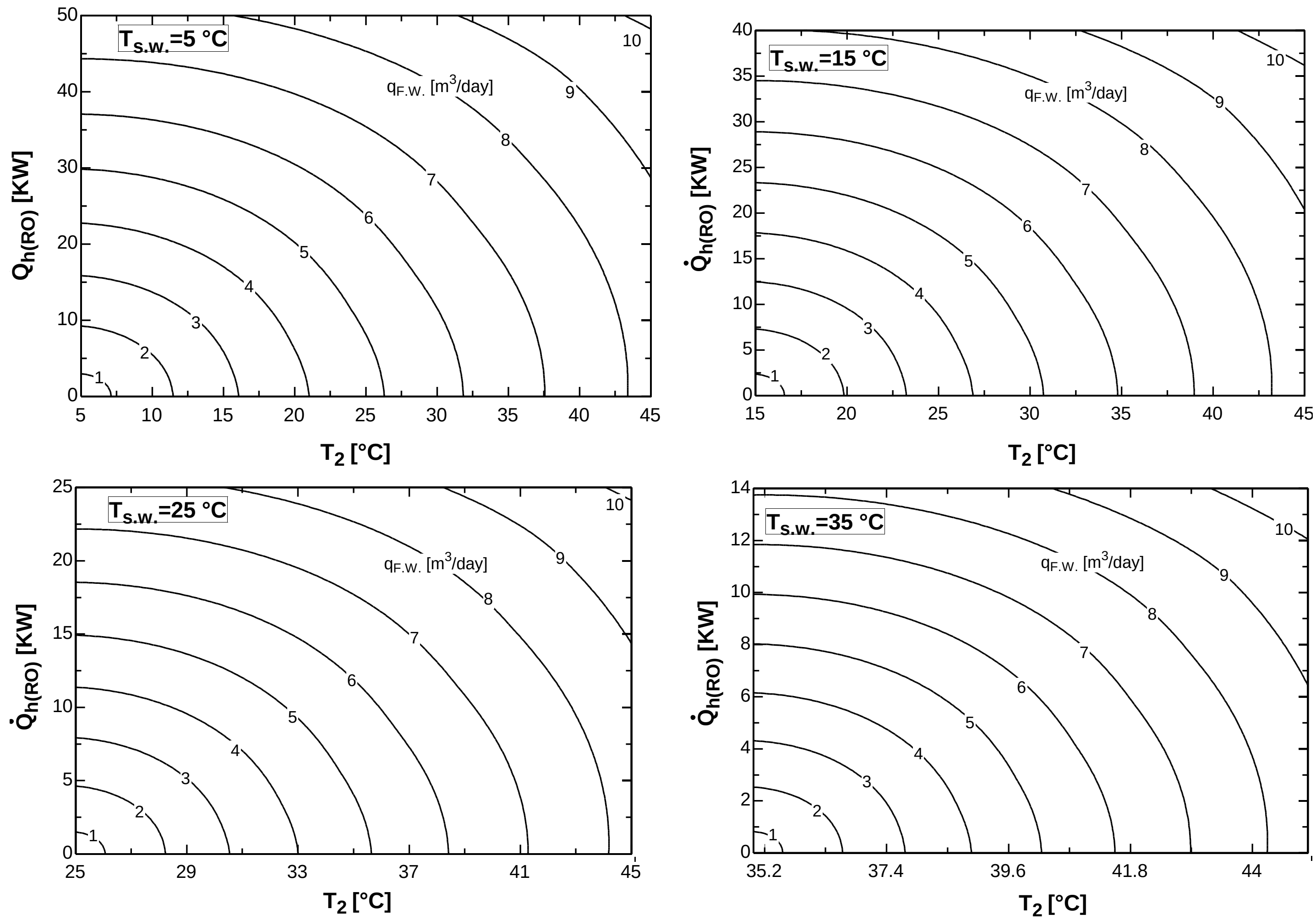

Figure 7. Heat rate required vs. the sea-water preheating temperature.

RESULTS AND DISCUSSION

Feasibility of integration of the RO and refrigeration systems integration in the presented system

Integration of the RO and refrigeration systems is possible when $\dot{Q}_{h(RO)}$ equals to $\dot{Q}_{h(ref.)}$.

The values of $\dot{Q}_{h(RO)}$ for preheating seawater up to temperature T_2 (the useable thermal range of the RO system) and also the values of $\dot{Q}_{h(ref.)}$ (to prepare the cooling load) are shown in Figures 9 and 10, based on the conditions used in Table 3.

As shown in Figure 9, the $\dot{Q}_{h(RO)} = \dot{Q}_{h(ref.)}$ line has intercepted temperature curves from 34.45 to 45˚C. This means that the integration of the refrigeration and RO systems (for the conditions 1 presented in Table 3) is possible in the preheating thermal range from 34.45 to 45˚C. However, as shown in Figure 10, none of the temperature curves have been intercepted by the $\dot{Q}_{h(RO)} = \dot{Q}_{h(ref.)}$ line, which means that integration of the refrigeration and RO systems (for the conditions 2 presented in Table 3) is impossible.

Exploration of the optimum preheating temperature in the feasible-integration region

The values of the power consumed by the integrated systems (calculated by Equation 19) versus the seawater preheating temperature in thermal range from 34.45 to 45˚C (for conditions 1 presented in Table 3) are shown in Figure 11. The values of the energy saving percentage afforded by the integrated system (calculated by Equation 15) versus the seawater preheating temperature in the thermal range from 34.45 to 45˚C are shown in Figure 12. As shown in Figure 12, the integration of the RO and

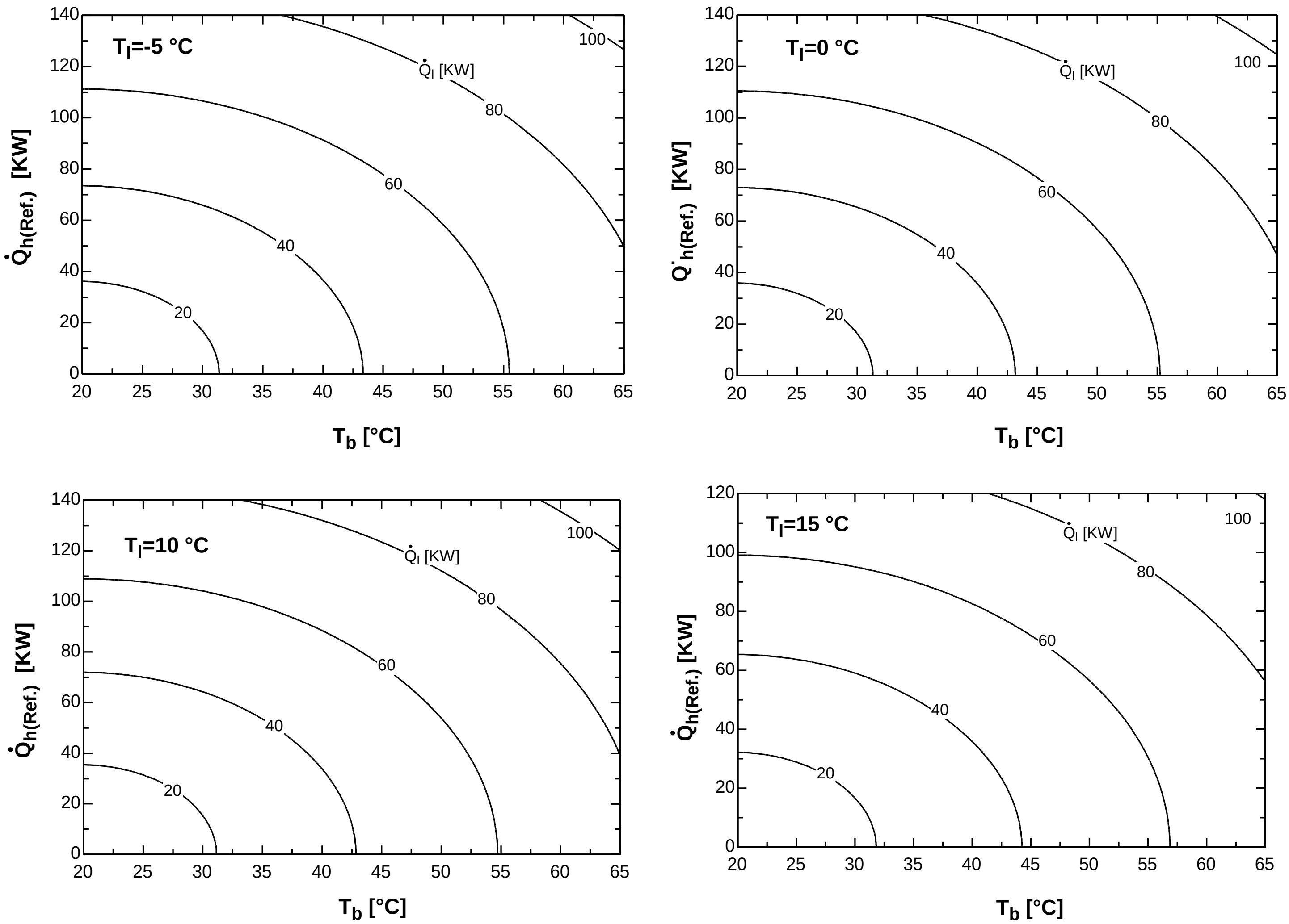

Figure 8. Values of waste heat rate vs. the compressor outlet temperature.

refrigeration systems is not desirable for all of the preheating temperatures from 39.4 to 45˚C. The integration is only desired in the preheating thermal range from 39.45 to 39.4˚C, because the energy saving percentage is more than zero. The maximum energy savings is 7.6% at a temperature of 34.45˚C.

Conclusion

Due to the need by many buildings and industries for fresh water and cooling at the same time, integration of RO and refrigeration systems can be used to cogenerate both fresh water and cold in order to reduce costs and energy consumption. With just a slight increase in the temperature of inlet seawater to the RO system, the intensity of power consumption can be reduced. By integrating this system with the refrigeration cycle, not only is there the possibility to increase the inlet seawater temperature into the desalination system, but some shaft work needed for this system could be recovered by replacement of the expansion valve in the refrigeration cycle with a turbo expander. In this paper, we describe a new method for combined cold and fresh water generation (CCW) by the integration of compression refrigeration and RO systems. In this new structure (CCW), the electrical power consumption of the pumps in the RO system is conserved by preheating the seawater through the waste heat recovery in the condenser of the refrigeration cycle. Furthermore, a reduction in electrical power demand of these pumps is possible by replacement of the expansion valve in the refrigeration cycle with a turbo expander that helps to drive the RO pump

Using code developing in EES software, the feasibility of integration of the RO and refrigeration systems to cogenerate 10 cm^3 per day of fresh water and 3 refrigeration tons of cooling was studied and the best

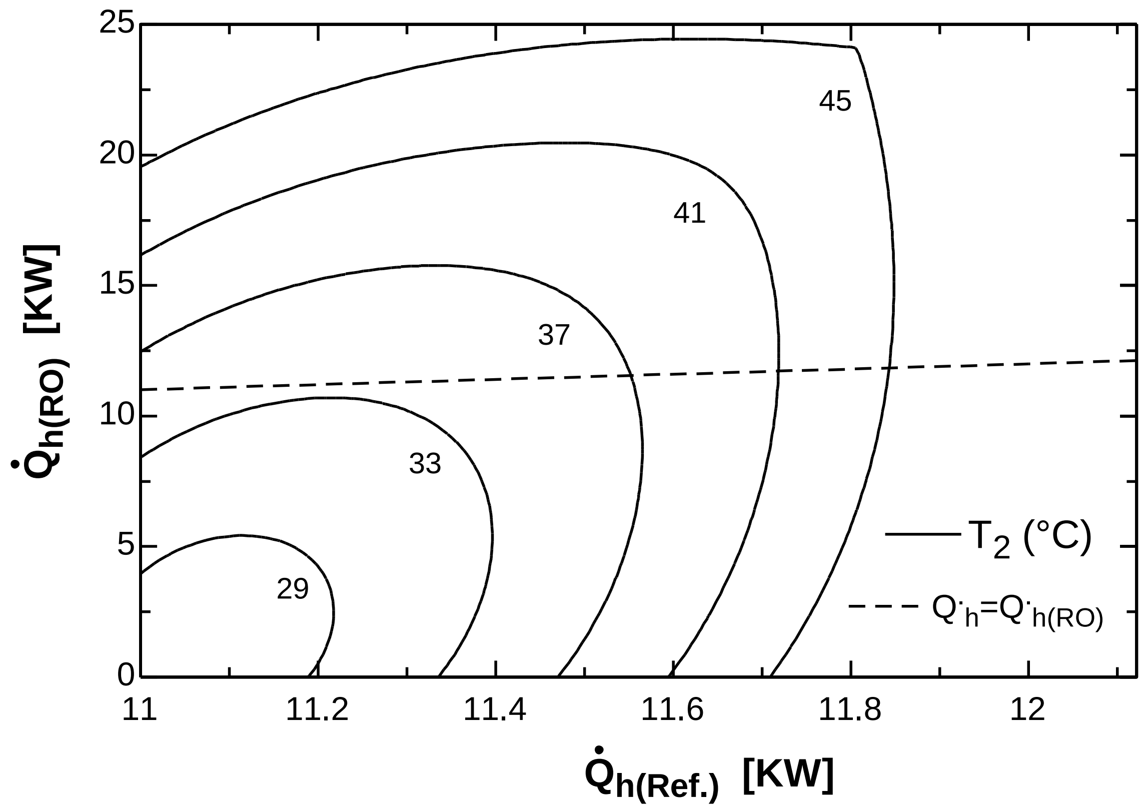

Figure 9. Integration Feasibility curve for condition 1.

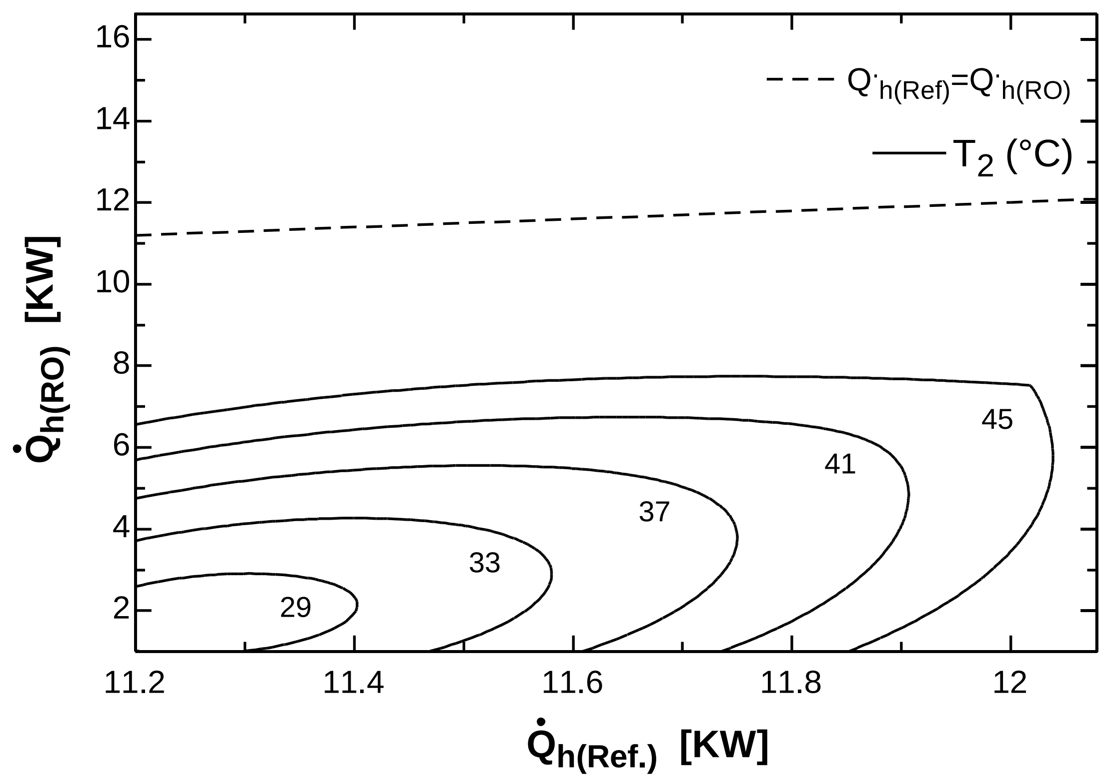

Figure 10. Integration Feasibility curve for condition 2.

Table 3. Operational conditions.

	$q_{F.W.}\ (m^3/d)$	$T_{s.w.}\ (^\circ C)$	$Q_l^\cdot\ (RT)$	$T_l\ (^\circ C)$
Condition 1	10	25	3	15
Condition 2	25	20	3	10

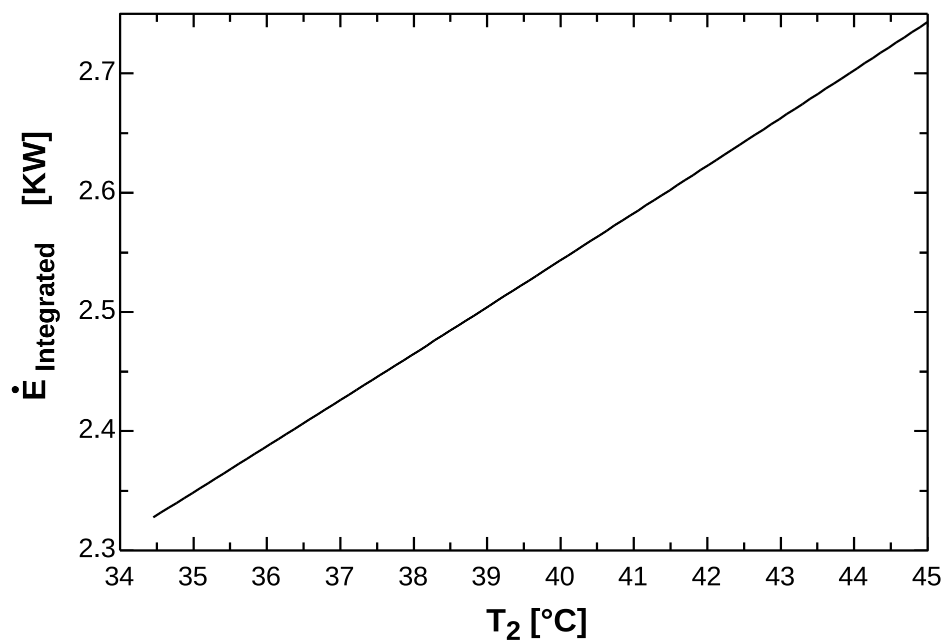

Figure 11. Values of the power consumed by the integrated systems.

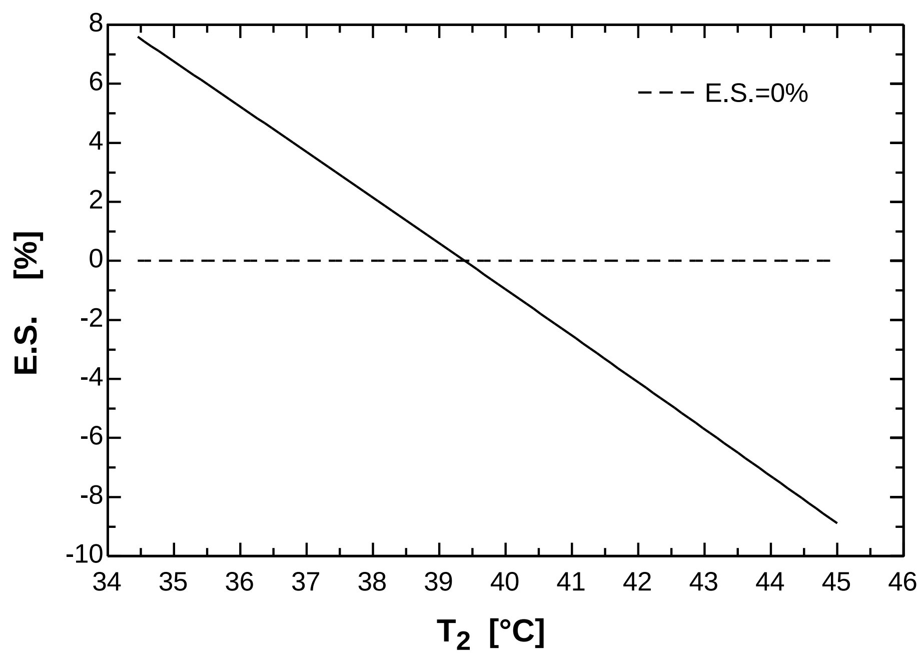

Figure 12. Energy saving percentage vs. the seawater preheating temperature.

state for this integrated system was determined. The method provides a reduction in power consumption of about 7.6% in comparison with the separate production of fresh water and cooling.

ACKNOWLEDGEMENT

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Nomenclature

b : Fresh water recovery in RO (%).

$\dot{E}_{RO}$: Power consumption in RO (kW).

$\dot{E}_{Integrated}$: Power consumption in the integrated system (kW).

$\dot{E}_{seprated}$: Power demand in the separated systems(KW).

$\dot{E}_{(RO)_{Ts.w.}}$ Power demand in the RO system without sea water preheating (kW).

$\dot{E}_{(RO)_{T}}$: Power demand in the RO system with sea water preheating (kW).

$\dot{E}_{pump}$ **:** Power consumption in the pump (kW).

$\dot{E}_{com.}$ **:** Power consumption in the compressor (kW)

$\dot{E}_{E.T.}$ **:**Power produced by the turbo expander (kW).

$\dot{E}_{E.R.T.}$ **:** Power produced by the energy recovery turbine (kW).

$\dot{E}_{ref.}$ **:** Power consumption in the refrigeration cycle (kW).

$E.S.$ **:** Energy saving percentage (%).

g: Gravity acceleration (N/kg).

h_{pump}: Head of the pump (m).

$h_{E.R.T.}$ **:** Head of the energy recovery turbine (m).

h_a **:** Compressor inlet flow enthalpy (kj/kg).

h_b : Compressor outlet flow enthalpy (kj/kg).

h_c : Turbo expander inlet flow enthalpy (kj/kg).

h_d : Turbo expander outlet flow enthalpy (kj/kg).

h_1 : Enthalpy of sea water (kJ/kg).

h_2 : Enthalpy of preheated sea water (kJ/kg).

$\dot{m}_R$ **:** Refrigerant mass flow rate (kg/s).

$\dot{Q}_l$ **:** Cooling load (kW).

$\dot{Q}_{h(RO)}$ **:** Heat rate required for sea water preheating (kW).

$\dot{Q}_{h(ref.)}$ **:** Waste heat rate in the condenser (kW).

$q_{F.W.}$ **:** Fresh water flow rate (kg/s).

$T_{S.W.}$ **:** Sea water temperature ($^{\circ}C$).

T **:** Preheated sea water temperature ($^{\circ}C$).

T_2 : Preheated sea water temperature in the heat exchanger of the CCW system ($^{\circ}C$).

T_l **:** Cold environment temperature ($^{\circ}C$).

T_b **:** Compressor outlet temperature ($^{\circ}C$).

Greek Letters

η_{pump} **:** Pump efficiency (%)

$\eta_{com.}$ **:** Compressor efficiency (%)

$\eta_{E.R.T.}$ **:** Energy recovery turbine efficiency (%)

$\eta_{E.T.}$ **:** Turbo expander efficiency (%)

$\rho_{E.R.T.}$ **:** Density of water inlet to energy recovery turbine (kg/m^3)

ρ_{water} **:** Water density (kg/m^3)

ρ_{pump} **:** Pump Inlet Sea water density (kg/m^3)

$\rho_{s.w.}$: Sea water density (kg/m^3)

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