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Technological options for thermal distillate cooling and their limitations subject to field implementation constraints and climatic conditions

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A B S T R A C T

Several cities in the Kingdom of Saudi Arabia often receive very hot water (46°C) from the water distribution network, which is supplied by a thermal desalination plant. In this work, an effort has been made to find suitable techno-economic solutions to cool hot distillate produced from thermal desalination before being sent to the water distribution network. The main constraints considered in the study were a cooled distillate temperature of 38°C when air temperatures reach 45°C and relative humidity reaches 70%. Several options, such as once-through seawater cooling, distillate flashing and condensing, and cooling towers have been evaluated. Wet cooling towers were able to reduce the temperature to 38°C most of the time, with the cooled temperature exceeding 40°C only 7.5% of the time. However, when seawater cooling was used, about 10% of the time the seawater temperature exceeded 35°C, which limited the cooling to 3.57°C only, that is cooled water temperature was only 40.43°C because of the constraints on the amount of seawater that could be supplied. After giving due consideration to the climatic factors, field/plant constraints and their impact on the efficiency of the solutions proposed, cooling towers were found to be the most suitable option.

Keywords: Hot thermal distillate; Seawater cooling; Wet cooling tower; Heat rejection; Relative humidity

1. Introduction

Many processes are used to produce potable water at a temperature higher than what is fit for human consumption and then cooled. In the Middle East, most of the water is desalinated using thermal desalination and membrane desalination processes. The membrane desalination process needs feed water at temperatures less than 40°C, thereby producing permeate at almost the same temperature. The conventional thermal desalination process on the other hand produces distillate that has slightly elevated temperatures and is cooled by exchanging heat with seawater, a part of which is used as make-up to the feed. Ozair et al. [1] have considered product water temperatures below 40°C in their assessment of MED-TVC units at Yanbu, while others have considered temperature below 42°C as the product water temperature. The final condenser pressure determines the product (distillate) temperature in thermal desalination plants. Many coastal cities in the Eastern Province receive water from thermal desalination plants. During peak summers, the water supplied to consumers is very hot, indicating the failure, under-design, or absence of distillate cooling systems.

Saroosh et al. [2] in their studies on cogeneration using CSP-MED considered a 10°C rise in seawater temperatures in the condenser at various steam extraction pressures. Kim and Jeong [3] in their studies on nuclear power plants (NPPs) considered 0.1235 bar (50°C) as the final condenser pressure in the Rankine power cycle, where seawater cooling was used in the Arabian Gulf. They discussed the effect of seawater cooling on the power plant efficiency, which can vary by about 10% between 10°C and 35°C seawater

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temperatures. The effect of seawater salinity and temperature on the condenser pressure was discussed by Ibrahim and Badawy [4]. Boland [5] discussed the steam condensing pressure range for direct water cooling of condensers, water cooling with a wet cooling tower, air-cooled condensers, and water cooling with dry cooling tower.

In the absence of a cooling mechanism for the distillate produced, cooling towers can be used. The concept of cooling by evaporation is used in cooling towers. Cooling towers can be broadly classified as atmospheric and mechanical draft towers. Depending on whether the fans are located on the air stream entry side or on the exit side, the mechanical draft towers can be categorized as forced draft or induced draft towers, respectively. Induced draft towers are widely used due to the wide range of flow rates they can handle and the higher air discharge velocity [6]. Depending on the direction of air flow, the cooling towers can be further classified as induced draft crossflow towers and induced draft counterflow towers.

Saroosh et al. [2] results indicated that the efficiency was about 0.6%–1% higher in the case of seawater cooling than that obtained by using wet cooling towers (for seawater temperature 30°C, dry bulb temperature 45°C, and wet bulb temperature 30.1°C).

Cooling towers have been extensively used in Saudi Arabia to achieve desired temperatures within the permissible limit, which in turn is controlled by many factors such as air temperature, relative humidity, and wet bulb temperature. Varied applications of cooling towers have been observed by the authors in Saudi Arabia. These have been used in cooling groundwater pumped from 1,200–1,500 m depth and having 60°C–70°C temperature range with a total dissolved solids of about 1,450 mg/L in the Riyadh Province of Saudi Arabia. The water was cooled to 30°C–35°C prior to being used as feed in reverse osmosis (RO) treatment plants [7]. Cooling towers with cooling capacities of 113.54 MW each have been used in the Salboukh (60,000 m^3/d) and Buweib (60,000 m^3/d) water treatment plants in Riyadh [8]. Cooling towers are also used in wastewater treatment plants to control the temperature of the feed water going into the anaerobic internal circulation (IC) reactors for maintaining proper growth conditions for the anaerobic bacteria that produce biogas (paper industry—personal communication). Here, the cooling towers are used to bring the temperature down to 38°C from an inlet temperature of 42°C–45°C. The towers are bypassed during the period from November to April because of the colder influent stream owing to the cooler ambient air temperatures.

The main parameters that determine the performance and size of a cooling tower are the range, the approach, the wet bulb temperature, and the heat load [9]. Range is the difference in cooling tower inlet and outlet water temperature. Approach is the difference between the cooling tower outlet water temperature and the ambient wet bulb temperature (Fig. 1). Wet bulb temperature (T_w) is the temperature seen when a wetted thermometer is exposed to air flow and it is a function of dry bulb temperature (ambient air temperature, T) and relative humidity (RH) [10]. At 100% relative humidity, the wet bulb and the dry bulb temperatures are equal.

$$\begin{aligned} T_W = {} & T\, a\tan\left[0.151977\left(\mathrm{RH\%}+8.313659\right)^{1/2}\right] \\ & + a\tan\left(T+\mathrm{RH\%}\right) - a\tan\left(\mathrm{RH\%}-1.676331\right) \\ & + 0.00391838\left(\mathrm{RH\%}\right)^{3/2} a\tan\left(0.023101\mathrm{RH\%}\right) \\ & - 4.686.35 \end{aligned} \tag{1}$$

Cooling tower efficiency is given by Eq. (2):

$$\text{Cooling Tower Efficiency} = \frac{\text{Range}}{\left(\text{Range}+\text{Approach}\right)} \times 100 \tag{2}$$

Ataei et al. [11] evaluated the performance of counter-flow wet cooling towers using exergetic analysis. Afshari and Dehghanpour [12] performed a review on cooling towers and simulated those in ANSYS Fluent. Qureshi and Zubair [13] developed a complete model of wet cooling towers with fouling in fills.

At the end of the introduction, what is the novelty of the work and where does it go beyond previous efforts in the literature? What is missing (i.e., research gaps)? What needs to be done? Cooling Water Options for the New Generation of Nuclear Power Stations in the UK (Turnpenny et al. [14]; Environment Agency). Give a comparison of cooling options considering direct cooling, wet cooling and dry cooling for power plants, with direct cooling providing the best efficiency. An attempt has been made to cool hot distillate from thermal a desalination plant by considering the aforementioned techniques. However, the results obtained showed different behaviour. Stochastic bi-objective optimization for closed wet cooling tower systems based on a simplified analytical model (Wu et al., 2022).

To the best knowledge of the authors, most of the solutions that have been designed are for cooling applications in green field power projects ranging from coal-fired power plants to concentrated solar power (CSP) plants. These have been constrained in terms of the availability of space and the projected cost, with limited impact from the weather, the seawater temperature, and seawater availability [14–17]. However, the applications considered in this manuscript are for a brown field desalination project subjected to the limited availability of seawater, high seawater temperatures, wide ambient temperature range and diurnal changes in relative humidity. Most of the cooling required in existing desalination plants is met through seawater cooling and it is extremely rare to consider cooling towers.

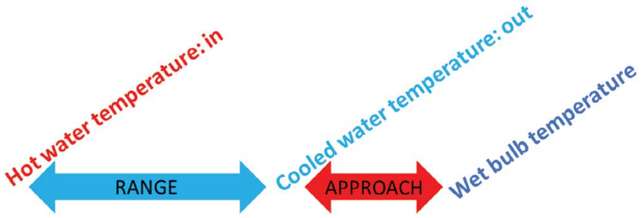

Fig. 1. Relationship between approach, range and wet-bulb temperature.

In this paper, several different solutions were evaluated to address the problem of the high temperature (~46°C) of the water pumped into Saudi cities during peak summers. Different feasibility studies were conducted to figure out the most suitable techno-economically viable solution. The technologies considered included the following, (1) using the existing brine cooler heat exchanger, which is normally used to preheat seawater feed to desalination unit in winter time only, to cool the distillate, (2) distillate flashing and condenser cooling, and (3) the installation of a cooling tower with a heat exchanger. Results are discussed in terms of the ability of the considered solution to lower the temperature despite the challenging climatology during summer.

2. Context of the problem

This project was started to address the problem of high temperatures in the water pumped into Saudi cities in the Eastern Province, without making any changes in the water production quantity and process. The main constraints were considered during the initial study to determine the proper cooling technique in order to keep the performance of the multi effect distillation (MED) unit at design conditions, while lowering the distillate (water) temperature. Different solutions were studied carefully to figure out the most suitable techno-economically viable option to reduce water temperature. Some of the solutions tested were as follows: (a) using the existing brine cooler heat exchanger, which is normally used to preheat seawater feed to the desalination unit in winter time only, (b) distillate flashing and condenser cooling and (c) installation of a cooling tower with a heat exchanger. However, the aforementioned solutions took into consideration some of the design limitations encountered at the site (i.e., the built MED plant), namely, (a) the limited design pressure of the existing heat exchanger, (b) the pump's low available net pressure suction head (NPSH), which will not be sufficient for the main desalination process which is multi effect distillation (MED) on its own, and (c) the limited seawater intake flow, which would limit the cooling water to the cooling system of MED. After a detailed study of the operation and design criteria of the MED plant, it was concluded that cooling towers were the most cost-effective solution to the problem. Proper care was taken to ensure the prevention of cross-contamination of the water while designing the solutions.

2.1. Limitations of the existing desalination plant system

2.1.1. Distillate production

The design point of the MED unit is approximately 12.43% distillate at 44°C, when 93.65% seawater is supplied at a temperature of 35°C. An analysis of the evaporator design and inspection of the actual evaporator condition may allow us to evaluate if there is any potential to increase the performance ratio, which could reduce the heat load on the cooling cycle and help keep the distillate temperature at a low level. However, detailed analysis shows limited room for enhancement if cleaning of fouled tubes in the final condenser is undertaken. However, this was found to be very limited and would not be sufficient to mitigate the situation.

2.1.2. Distillate pumps and available pressure head

Distillate pumps are installed after the MED and are capable of moving the targeted amount of distillate. The net pressure suction head (NPSH) is 3.4 m.

2.1.3. Heat load from distillate of MED

The distillate from the MED unit needs to be cooled to at least 38°C from 45°C to 46°C during periods when the seawater temperature exceeds 28°C. The heat load can be calculated as:

$$\text{Heat load} = mC_p\Delta T = Q\rho C_p\Delta T \quad (3)$$

where Q is the flowrate of distillate (m^3/h), ρ is the density of distillate (kg/m^3), C_p is the specific heat capacity of distillate, and ΔT is the difference in temperature between the hot and cold distillate. The calculated heat load of the MED unit is 8,602 kW.

2.1.4. Seawater supply pumps (SSP) and cooling water availability

The MED unit is supplied by a SSP with a margin of 6.35% higher flow, which implies an additional availability of seawater supply. There are no standby pumps which could potentially provide additional seawater supply when needed. The maximum seawater temperature crosses 35°C and can reach 37°C during summers (Fig. 2a), which indirectly impacts the cooling capacity. At duty point, the SSP efficiency is around 87%, with an available head of 30.5 m.

2.1.5. Preheater fouling, pressure, and flow velocities

Flow velocities less than 0.3 m/s will lead to fouling of heat exchanger tubes. Hence, the velocity of flow needs to be taken into consideration while determining the viable flow rate. The preheater can handle pressures below 3 bar.

2.1.6. Space for piping and other miscellaneous civil works

There is no available built infrastructure to install additional SSPs and to undertake large amount of pipework.

2.2. Climatic factors

The main climatic factor that influences seawater cooling is the seawater temperature. The seawater temperature in the Eastern Province of Saudi Arabia varies from 14°C during the winter to 37°C during the summer (Fig. 2a).

On the other hand, the performance of cooling towers is mainly influenced by the ambient air temperature and the relative humidity. Air temperature crosses 45°C during summers and can reach values up to 47°C (Fig. 2b).

2.3. Percent occurrence of different climate parameters

From Fig. 3 it can be seen that seawater temperatures exceed 35°C almost 10% of the time in a year (i.e., during August and September). Seawater temperatures are below

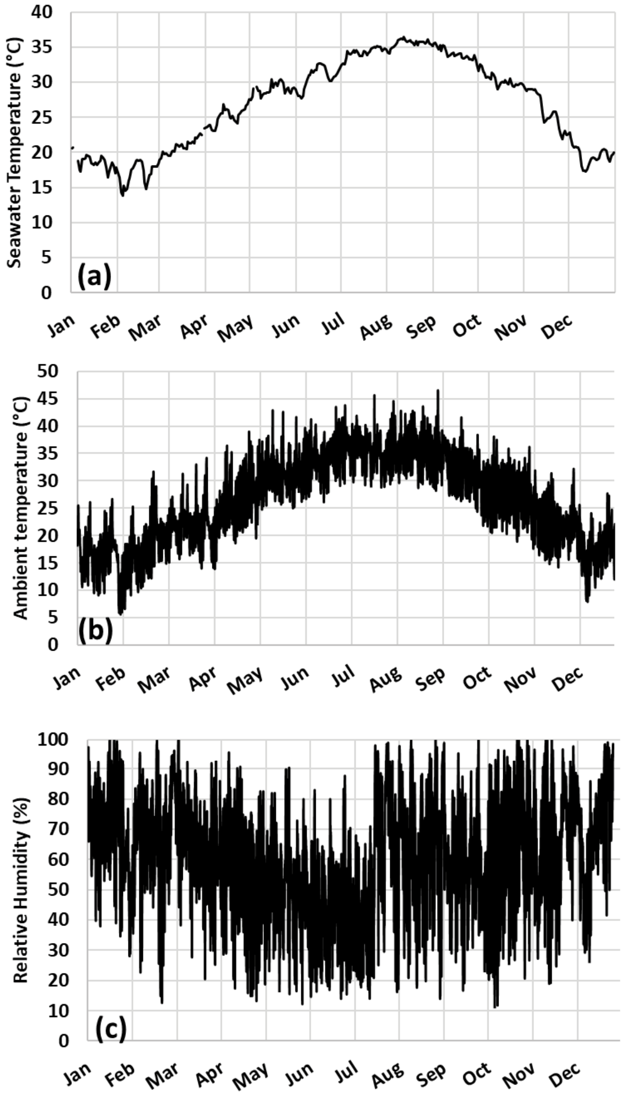

Fig. 2. Variation of seawater temperature, ambient temperature and relative humidity in Jubail in the Eastern Province of Saudi Arabia.

27.5°C about 46% of the time in a year. Ambient temperatures above 32.5°C occur almost 31% of the time in a year (Fig. 3) and those exceeding 37.5°C occur 16% of the time.

The ambient temperature range is 30°C–35°C with relative humidity exceeding 70% about 5.9% of the time (Table 1). Table 1 was compiled using the ambient temperature and relative humidity data for Jubail. 6.9% of the time in a year, the ambient temperature ranges between 35°C and 40°C and the relative humidity exceeds 50%. Similarly, only 0.6% of the time in a year, the ambient temperature ranges from 40°C to 45°C and relative humidity exceeds 30%.

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3. Concept and solutions tried

Locations away from the coastal areas have limited seawater available and need to adopt dry cooling or wet cooling towers depending on the availability of water, whereas those on the seacoast can adopt once through seawater cooling. The solutions tested for cooling the hot distillate are briefly discussed in the following sub-sections.

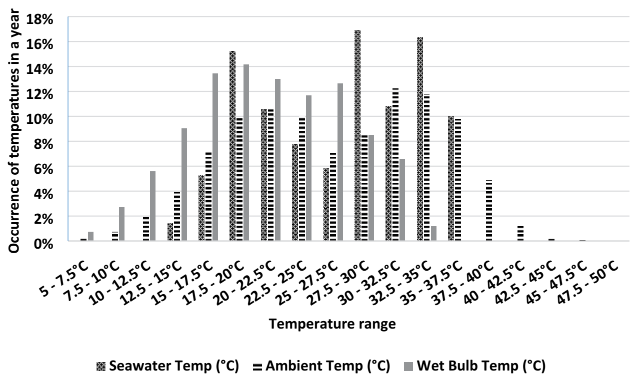

Fig. 3. Percent occurrence of different temperature ranges for the seawater, ambient temperature and the wet bulb temperature.

Table 1
Occurrence of combinations of ambient temperature and relative humidity in a year

		Ambient temperature								
		5°C–10°C	10°C–15°C	15°C–20°C	20°C–25°C	25°C–30°C	30°C–35°C	35°C–40°C	40°C–45°C	45°C–50°C
Relative humidity	0%–10%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%
	10%–20%	0.0%	0.0%	0.0%	0.0%	0.1%	0.4%	0.5%	0.2%	0.0%
	20%–30%	0.0%	0.0%	0.0%	0.3%	0.7%	1.7%	1.7%	0.4%	0.0%
	30%–40%	0.2%	0.2%	0.3%	0.8%	1.8%	2.3%	2.5%	0.4%	0.0%
	40%–50%	0.1%	0.4%	0.8%	2.0%	2.3%	4.4%	3.2%	0.2%	0.0%
	50%–60%	0.2%	0.7%	2.1%	3.4%	2.9%	5.2%	3.1%	0.0%	0.0%
	60%–70%	0.1%	1.1%	4.7%	4.7%	2.9%	4.2%	2.1%	0.0%	0.0%
	70%–80%	0.1%	1.5%	4.3%	5.1%	2.7%	2.5%	1.5%	0.0%	0.0%
	80%–90%	0.0%	1.2%	2.9%	3.2%	1.6%	2.2%	0.2%	0.0%	0.0%
	90%–100%	0.2%	0.7%	1.9%	1.3%	0.6%	1.2%	0.0%	0.0%	0.0%

3.1. Solution 1: once through seawater cooling (OTSC)

3.1.1. Using the existing seawater preheater

The maximum allowable seawater cooling discharge temperature rise is 10°C in the Royal Commission areas of Jubail and Yanbu [18]. However, the Saline Water Conversion Corporation (SWCC) limits the maximum temperature rise of discharge to 7°C in most of its thermal desalination plants. The amount of distillate cooling achievable using the existing seawater preheater in the MED unit and when using a new plate heat exchanger are discussed here. The seawater preheater is being used to cool the brine being blown down and heat the feed during winter time only. Therefore, an attempt was made to use it to cool the distillate during summer time. Fig. 4 shows a schematic of the system with the existing seawater preheater (bounded by grey dashed line) and the new plate heat exchanger considered (bounded by black dashed line). Some of the seawater is rejected at 42°C and is also used as make-up in the MED at 42°C. There is about 6.35% seawater supply margin available. If this seawater is dumped into the sea without mixing it with the seawater going to the distillate condenser, then the performance characteristics of the MED will not change (Fig. 5). However, if it is mixed with the seawater going to the distillate condenser, then the make-up temperature will change resulting in a change in the performance of the MED unit, which is out of scope of the current study.

HTRI® software was used to evaluate the cooling capabilities of the existing seawater preheater (brine cooler) at different seawater cooling flow rates. Table 2 shows the distillate cooling achieved at different seawater flowrates. The distillate can be cooled from 44°C to 40.7°C (3.3°C). By installing a pressure reducing valve, the distillate pressure could be reduced to less than 3 bar which is safe for the preheater heat exchanger.

Table 3 shows the process parameters used while evaluating the existing seawater pre-heater and the new plate heat exchanger. Table 4 shows the performance of the existing seawater pre-heater (brine cooler), when hot distillate is used instead of brine. When the seawater temperature is 35°C, only 3.57°C cooling is achieved, whereas 4.11°C cooling is seen when the seawater flow is 38.22%, which is more than the allowable margin of seawater flow available. The maximum seawater discharge temperature is 40.18°C,

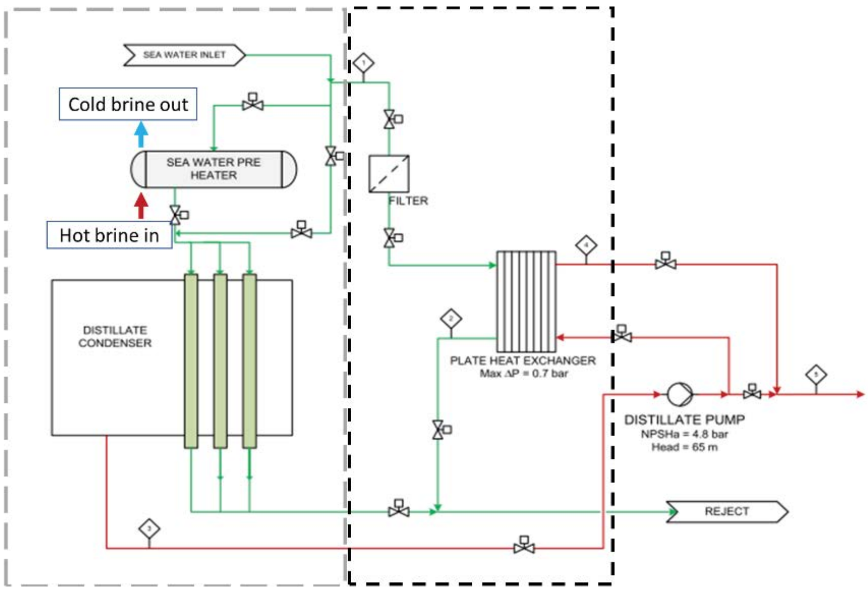

Fig. 4. Direct cooling of water (MED distillate) using seawater with plate type heat exchanger.

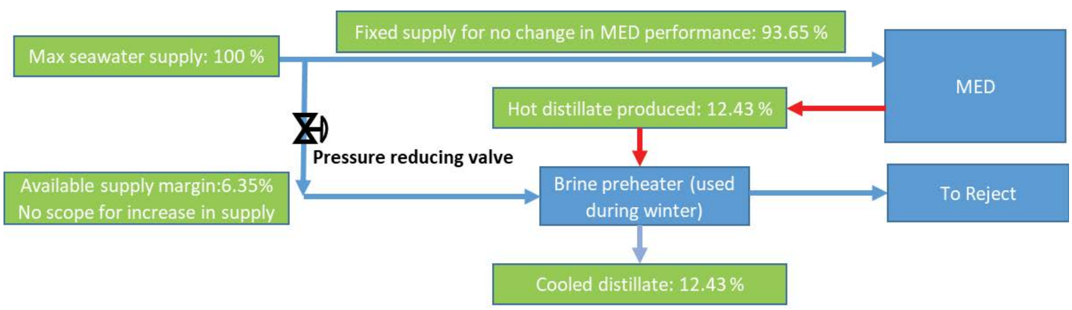

Fig. 5. Schematic representation showing the usage of available margin of seawater supply to reduce the distillate temperature in the preheater (brine cooler).

Table 2
Performance of the seawater preheater

Distillate				Seawater			
Distillate %	Shell vel. (m/s)	T_{in} (°C)	T_{out} (°C)	Seawater supply %	Tube vel. (m/s)	T_{in} (°C)	T_{out} (°C)
12.43%	0.35	44	37	40%	2	35	37.2
12.43%	0.35	44	37.1	35%	1.8	35	37.4
12.43%	0.35	44	37.4	30%	1.5	35	37.8
12.43%	0.35	44	37.6	27%	1.4	35	38
12.43%	0.35	44	37.6	26%	1.3	35	38.1
12.43%	0.35	44	37.8	23%	1.2	35	38.3
12.43%	0.35	44	38.1	20%	1	35	38.7
12.43%	0.35	44	38.4	17%	0.9	35	39.1
12.43%	0.35	44	38.8	14%	0.7	35	39.6
12.43%	0.35	44	39.6	10%	0.5	35	40.4
12.43%	0.35	44	40.71	6.35%	0.32	35	41.44
12.43%	0.35	44	40.84	6%	0.30	35	41.55

which meets the effluent standards. The pressure drops on the shell side (hot distillate) and the tube side (cooling seawater) are 72 and 35 kPa, respectively.

Table 4 shows the cooling that can be achieved by adding a new plate type heat exchanger to the existing system. Table 4 shows variation of seawater cooling effect on distillate temperature, through preheater.

3.1.2. Reversing the flow direction of distillate in the preheater

The flow direction of the distillate which flows on the shell side was reversed in order to evaluate its impact on cooling efficiency. It was found that there was a minor improvement in cooling on the order of 0.1°C. Table 5 shows the results when the flow direction was reversed.

3.1.3. Using the rejected seawater to cool the distillate

The rejected seawater which goes through the distillate condenser reaches a temperature of 42°C. If this is used to cool the hot distillate which is at 44°C, the cooling achieved is insignificant as the ΔT between the hot and cold streams is almost equal to the pinch point, which usually ranges between 1°C and 2°C. Lowering the pinch point entails the usage of very large heat transfer area.

3.2. Solution 2: distillate flashing and condensing

The distillate is flashed in a flash vessel which is maintained at the saturation pressure corresponding to a temperature 5°C above the cooling seawater temperature. The pressure in the flash vessel is maintained due to the condensation of vapours in the condenser. The condensation of the vapours is ensured by the additional available cooling water and periodic venting of non-condensable gases accumulated in the flash vessel. According to the seawater temperature and flow available, the vapour flow rate that needs to be condensed is determined.

The heat and mass balance equations of the flashing and condensing system neglecting losses.

Table 3
Cold and hot fluid process fluid parameters for the existing pre-heater and new plate heat exchanger

	Existing pre-heater	New plate heat exchanger
Outside diameter, mm	19.05	19.05
Wall thickness, mm	0.50	0.50
Tube length (1)*, mm	8,580	24,000
Number of tubes per pass	2,200 × 2	1,600
Number of passes (1)*	2	1
Tube material	Titanium	Titanium
FF tube side, $m^2 \cdot K/W$	0.000300	0.000300
FF shell side, $m^2 \cdot K/W$	0.000200	0.000000
HT area, m^2		2,238
Distillate location	Shell side	Shell side
Direction	Mixed cross	Counter current
Direction	Mixed	Cross flow
Support plate "e" mm	1,200	3,000
Support plate "e 1" (2)*, mm		750
Shell diameter, mm	2,680	1,600
Tube pitch factor	1.36	1.33333333
Tube configuration	Δ60°	Δ60°

Total number of tubes to be input = Total number of passes × Total number of tubes.

Flashing system:

$$m_{hd} = m_{fd} + m_{cd} \tag{4}$$

$$m_{hd} h_{hd} = m_{cd} h_{cd} + m_{fd} h_{fd} \tag{5}$$

Condenser system:

$$m_{fd} h_{fd} = m_{SW} C_p \left(T_{SW,out} - T_{SW,in} \right) + m_{fd} h_{cond,fv} \tag{6}$$

Table 4
Performance of the existing seawater preheater and the new plate type seawater preheater

	Existing preheater				New preheater			
	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
Shell diameter, mm	2,680	2,680	2,680	2,680	1,600	1,600	1,600	1,600
Support plate "e", mm	1,200	1,200	1,200	1,200	3,000	3,000	3,000	3,000
Tube pitch factor	1.36	1.36	1.36	1.36	1.3	1.3	1.3	1.3
Seawater supply, %	38.22%	25.48%	38.22%	25.48%	38.22%	25.48%	38.22%	25.48%
Temperature—in, °C	35	35	34	34	35	35	34	34
Temperature—out, °C	38.98	40.18	38.86	40.33	40.33	41.86	40.52	42.39
Distillate flow (3)*, %	12.43%	12.43%	12.43%	12.43%	12.43%	12.43%	12.43%	12.43%
Temperature—in, °C	44	44	45	45	44	44	45	45
Temperature—out, °C	39.89	40.43	39.98	40.64	38.49	39.28	38.27	39.22
Tube side velocity, m/s	1.84	1.23	1.84	1.23	2.53	1.69	2.53	1.69
Tube side pressure drop, kPa	35	16	35	16	72	32	72	32
Shell side velocity, m/s	0.77	0.77	0.77	0.77	0.66	0.66	0.66	0.66
Shell side pressure drop, kPa	72	72	72	72	72	65	65	65
Heat transfer coefficient, W/m²·K	1,756	1,745	1,756	1,754	2,646	2,724	2,746	2,724

Table 5
Comparison of the difference in distillate temperature achieved by reversing flow direction (shell side)

Case	Distillate %	T_{out} (°C)	T_{out} reversed (°C)	Difference in T_{out} (°C)
1	12.43%	36.95	36.85	–0.10
2	12.43%	37.13	37.02	–0.11
3	12.43%	37.4	37.25	–0.12
4	12.43%	37.6	37.42	–0.13
5	12.43%	37.6	37.48	–0.13
6	12.43%	37.8	37.70	–0.13
7	12.43%	38.1	37.96	–0.15
8	12.43%	38.4	38.29	–0.15
9	12.43%	38.8	38.72	–0.12
10	12.43%	39.6	39.51	–0.12

where m_{hd} = mass of hot distillate from one MED, m_{fd} = mass of flashed distillate (t/h), m_{cd} = mass of distillate remaining after flashing (*t/h*), h_{hd} = enthalpy of hot distillate (kJ/kg), h_{cd} = enthalpy of distillate remaining after flashing (kJ/kg), h_{fd} = enthalpy of flashed distillate (kJ/kg), $h_{cond,fv}$ = enthalpy of condensed vapor in condenser (kJ/kg), C_p = Specific heat capacity of seawater at constant pressure (kJ/kg·°C), $T_{SW,in}$ = temperature of seawater going in (°C), $T_{SW,out}$ = temperature of seawater coming out (°C).

During the summers, the seawater temperature can exceed 37°C, corresponding to which the condenser can achieve a temperature of 42°C (saturation pressure equals 0.0821 bar). Highest cooling is achieved when the seawater temperatures are lower (Fig. 7a). However, the amount of flashed vapor is also higher (Fig. 7b), which would result in very large flash vessels. Less than 3°C distillate cooling is achieved when the seawater temperatures exceed 35°C. This is the only cooling possible, as only 6.35%

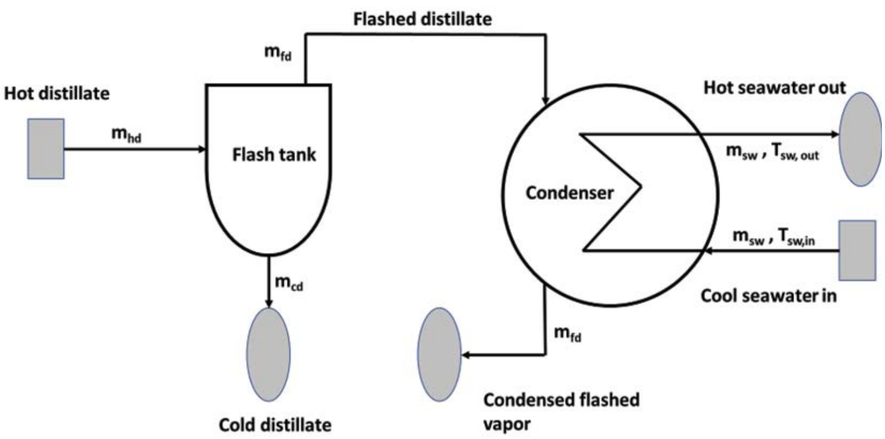

Fig. 6. Schematic representation showing the usage of available margin of seawater supply to reduce the distillate temperature in the preheater (brine cooler).

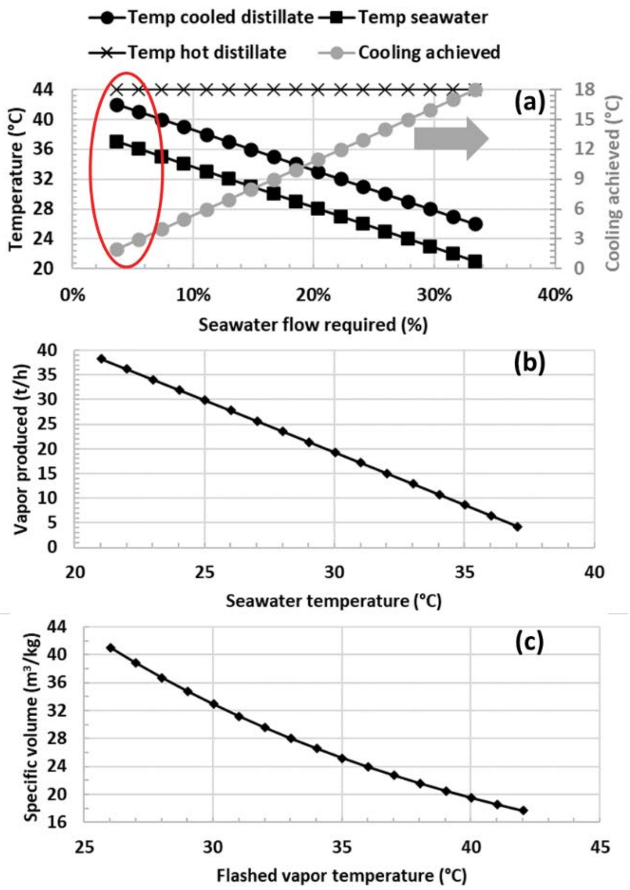

Fig. 7. (a) Temperature of seawater and cooled distillate at various seawater flows. Regimes covered under the available seawater flows are highlighted by the red ellipse. (b) Vapor produced in the flash vessel at various seawater temperatures. (c) Specific volume of the vapor produced in the flash vessel at various flashing temperatures.

extra seawater is available for cooling during summers for the MED (see the highlighted area in Fig. 7a). The amount of vapor produced with varying seawater temperature is shown in Fig. 7b. The sizing of flash vessels and the techno-economics are a topic for another manuscript and details are not discussed here.

During winters, when the seawater temperatures are low, the flashing temperature is also low. Due to the low temperatures, the flashed vapor has very high specific volume, and hence velocity of vapor is very high which results in very large flashing vessel size (Fig. 7c). The maximum allowable vapor velocity in the vessel is calculated using the Souders–Brown equation [19,20]:

$$v = k\sqrt{\frac{\rho_L - \rho_v}{\rho_v}} \tag{7}$$

where v = Maximum allowable vapor velocity, m/s; ρ_L = Liquid density (kg/m^3); ρ_v = Vapour density (kg/m^3); k = 0.107 (m/s), when the drum includes a de-entraining mesh pad.

The cross-sectional area of the drum can be found from:

$$A = \frac{\dot{V}}{v} \tag{8}$$

where $\dot{V}$ = Volumetric flow rate of vapour (m^3/s); A = Cross-sectional area of the drum.

The diameter of the drum is given as:

$$D = \sqrt{\frac{4A}{\pi}} \tag{9}$$

Another constraint which limits the flashing vessel size is the maximum metal sheet length that can be manufactured, which is only 25 m. This observation was made by Mahmoud et al. [21] during the design of long tube (LT) multistage flash (MSF) systems. As per calculations made to design the evaporator of MSF (not shown), flashing and condensing will not be a suitable option for seawater temperatures below 27°C.

3.3. Solution 3: using cooling towers

3.3.1. Dry cooling towers

Dry cooling towers cool the fluid through convective heat transfer, unlike wet cooling towers where evaporation of water takes place. The amount of heat rejected is dependent on the ambient temperature. Even if an approach to the ambient temperature is considered as 5°C, the dry cooling system will fail to cool the hot distillate to 38°C on most of the days of June, July, and August. This is because of the ambient temperatures exceeding 30°C most of the time during the summer months (Fig. 2b).

3.3.2. Wet cooling towers

A wet cooling tower is used to cool water which is passed through a heat exchanger where the hot distillate rejects heat, thus preventing contamination of the distillate (Fig. 8). The water being recirculated in the cooling tower is obtained from the distillate stream. 5°C was considered as the approach temperature in the cooling tower with 3°C being taken as the pinch in the heat exchanger. The heat and mass balance equations of a cooling tower system neglecting losses under steady state conditions. Reduction in the temperature of the hot distillate is governed by the ambient temperature and the relative humidity.

$$m_{a,\text{in}} = m_{a,\text{out}} \tag{10}$$

$$m_{w,\text{in}} = m_{w,\text{out}} \tag{11}$$

$$m_{p,\text{out}} = m_{s,\text{in}} + m_{\text{evap}} \tag{12}$$

$$m_{s,\text{in}} + m_{a,\text{in}}\text{RH}_{a,\text{in}} = m_{p,\text{out}} + m_{a,\text{out}}\text{RH}_{a,\text{out}} \tag{13}$$

$$m_{a,\text{in}}h_{a,\text{in}} + m_{s,\text{in}}h_{s,\text{in}} = m_{a,\text{out}}h_{a,\text{out}} + m_{p,\text{out}}h_{p,\text{out}} \tag{14}$$

where $m_{a,\text{in}}$ is the mass flow rate of dry air coming in, $m_{a,\text{out}}$ is the mass flow rate of dry air going out, m_{evap} is the mass flow rate of water lost to the atmosphere, $m_{w,\text{hot}}$ is the mass flow rate of hot distillate coming in, $m_{w,cool}$ is the mass flow rate of cooled distillate going out, $m_{s,in}$ is the mass flow rate of water being sprayed, $m_{p,out}$ is the mass flow rate of water being pumped, $RH_{a,in}$ is the relative humidity of air coming in (%), $RH_{a,out}$ is the relative humidity of air going out (%, almost 98%), $h_{a,in}$ is the enthalpy of air coming in (kJ/kg), $h_{a,out}$ is the enthalpy of air going out of the cooling tower (kJ/kg), $h_{s,in}$ is the enthalpy of the water being sprayed (kJ/kg), $h_{p,out}$ is the enthalpy of the water being sprayed (kJ/kg).

Enthalpy of saturated air (h_a) can be calculated as [22]:

$$h_a = 4.7926 + 2.568T_{wb} - 0.029834T_{wb}^2 + 0.0016657T_{wb}^3 \quad (15)$$

The months of June, July and August experience high day time temperatures (>40°C) with moderate humidity and moderate (>30°C) night temperatures with high humidity. The temperatures can exceed 47°C during the day with humidity reaching values as high as 80% in the night. Fig. 9a shows the variation of ambient temperature and relative humidity in the month of August and the inverse relationship between ambient temperature and relative humidity is highlighted in Fig. 9b for some days from July and August when the temperature and relative humidity are high.

A study was performed by varying the ambient temperature and the relative humidity. The temperatures considered were 30°C, 35°C, 40°C, 45°C, and 47°C, which are observed during the months of June, July, and August. Relative humidity exceeding 45% is mostly seen when the temperatures are below 40°C (Fig. 9b). Therefore, for each temperature mentioned above, relative humidity was varied between 10% and 80% in increments of 10%. The results are presented in Fig. 10.

The desired cooled distillate temperature (38°C) is achieved for all values of relative humidity considered when the ambient temperature is 30°C (Fig. 10c). For an ambient temperature equal to 35°C, when the relative humidity is 80%, the distillate is cooled to 39.6°C only. For the 40°C, 45°C, and 47°C ambient temperature cases, 38°C can be achieved

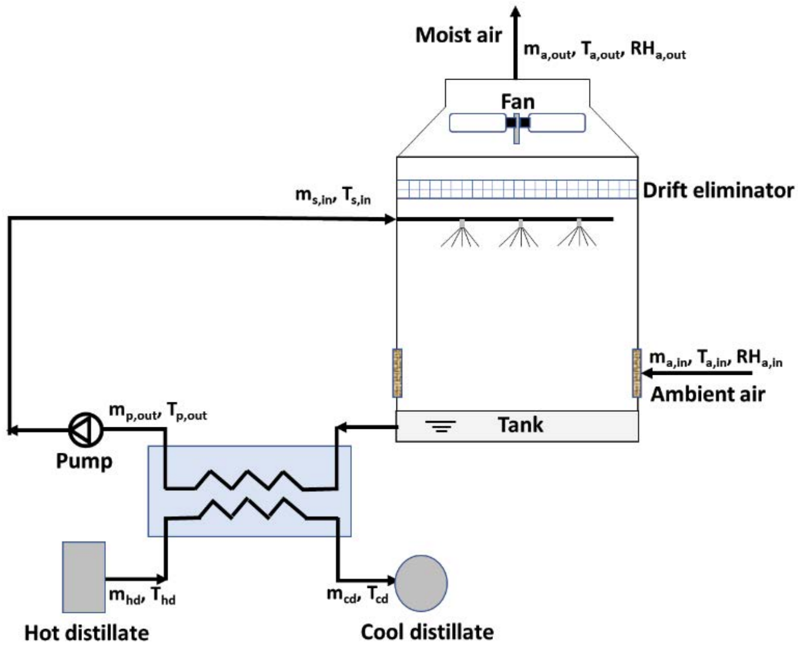

Fig. 8. Cooling tower with a heat exchanger used for cooling the hot distillate.

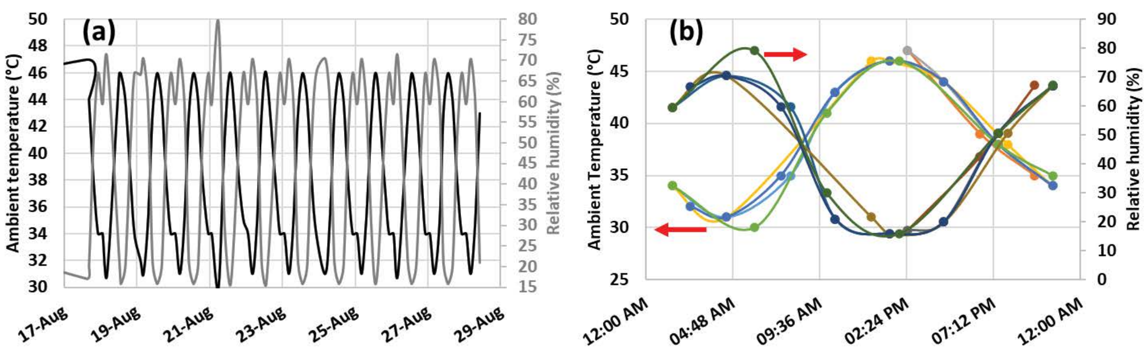

Fig. 9. Variation of ambient temperature and relative humidity in Jubail. (a) Values from the month of August are shown. (b) Relationship between ambient temperature and relative humidity. Values from days experiencing high values are shown.

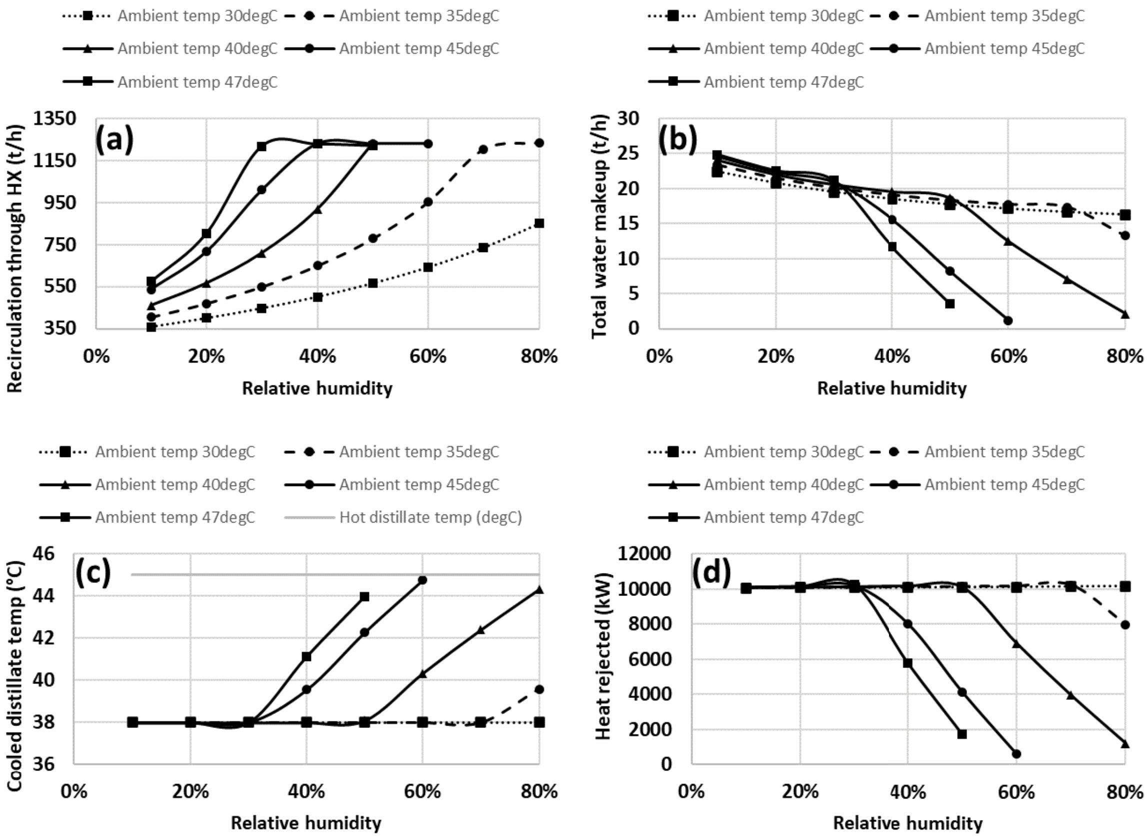

Fig. 10. (a) Variation of flow through the heat exchanger with relative humidity. (b) Variation of total water makeup with relative humidity. (c) Cooled distillate temperature achieved when the relative humidity is varied. Grey line represents the hot distillate temperature (45°C). (d) Heat rejected in the cooling tower when the relative humidity is varied.

up to 50%, 30%, and 30% relative humidities, respectively. The heat rejected and the total water makeup decreases with an increase in the relative humidity indicating the inability of the wet cooling tower in achieving desired cooling.

Actual variation of ambient temperature and wet bulb temperature based on field measurements for a typical summer day (27th August) are shown in Fig. 11. High and low ambient temperatures are observed in conjunction with low and high relative humidity, respectively. This results in very low variation in the wet bulb temperature. Furthermore, slightly higher amount of heat is rejected when the ambient temperature is high because of the lower wet bulb temperature. As a very minor amount of water (<40 t/h) is necessary for making up for lost water, this approach can be considered viable. The sizing of the cooling towers and the techno-economics are a topic for another manuscript and details are not discussed here.

4. Discussion and conclusions

An attempt was made to come up with viable solutions for cooling hot distillate being produced in a MED at a temperature of 45°C–46°C to a temperature of 38°C. The solutions considered were constrained by the availability of seawater, seawater temperature, ambient temperature and relative humidity which resulted in high approach temperatures.

When using once through seawater cooling (OTSC) with the existing seawater preheater, only 3.57°C cooling could be achieved when the seawater temperature was 35°C. This does not cool the distillate to the desired temperature of 38°C. Further cooling would require additional seawater which is beyond the available capacity. Furthermore, from Fig. 3 it can be seen that seawater temperatures exceed 35°C almost 10% of the time in a year (i.e., during August and September), which indicates that OTSC with the usage of the existing preheater is not a viable option. Even with the installation of a new plate type heat exchanger, cooling the distillate to 39°C required almost double the available capacity of seawater supply. Reversing the flow in the preheater improved the cooling by a meagre 0.1°C, which indicated the non-viability of the solution.

The second solution tested was flashing the hot distillate and condensing the vapor in a condenser using seawater as a cooling source. When the seawater temperatures are less than 27°C, the size of the flash vessels becomes very large due to very high specific volume of vapor at lower temperatures. About 46% of the time in a year, seawater temperatures are below 27.5°C (Fig. 3), which makes the solution techno-economically unfeasible.

Dry and wet cooling towers with non-mixing of streams were considered as another option for cooling the hot distillate. Ambient temperatures above 32.5°C occur almost 31% of the time in a year (Fig. 3) and those exceeding 37.5°C

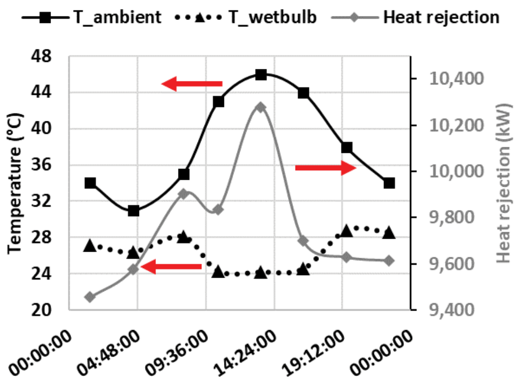

Fig. 11. Variations of ambient temperature and wet bulb temperature are shown on the primary vertical axis. Heat rejected by the wet cooling tower is shown on the secondary vertical axis.

occur 16% of the time. Even with an approach to dry bulb temperature being maintained at 5°C, the targeted cooling temperature of 38°C could not be met easily whenever the ambient temperatures exceed 32.5°C. Therefore, dry cooling was discarded as a cooling option based on the technical aspects itself.

Wet cooling towers are the best cooling option as the cooled distillate is able to achieve 38°C temperature (Fig. 10c) about 86.6% of the time. However, when the ambient temperature range is 30°C–35°C with relative humidity exceeding 70%, the cooled temperature might be between 39°C and 40°C about 5.9% of the time (Table 1 and Fig. 10c). Following the same logic, it can be concluded that about 6.9% of the time in a year, the cooled distillate temperature achieved will be between 40°C and 44°C, when the ambient temperature ranges between 35°C and 40°C and the relative humidity exceeds 50%. Similarly, only 0.6% of the time in a year, when the ambient temperature ranges from 40°C to 45°C and relative humidity exceeds 30%, the cooled distillate temperature will be between 39.55°C and 44.76°C.

Based on the discussions made above, it can be concluded that wet cooling towers with heat exchanger are the most suitable option for cooling the hot distillate based on the climatological conditions of the region. Further work needs to be carried out in terms of techno-economic evaluation and field testing of the proposed solution. If stringent conditions are imposed on the cooled water temperature to be at 38°C, 100% of the time, then nanofluid applications for enhanced heat transfer in cooling towers could potentially be explored. Climate change impacts on the performance of the wet cooling towers need to be considered in future studies. In places with ample seawater supply, seawater cooling tower can be considered.

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