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A novel study of using oil refinery plants waste gases for thermal desalination and electric power generation: Energy, exergy & cost evaluations

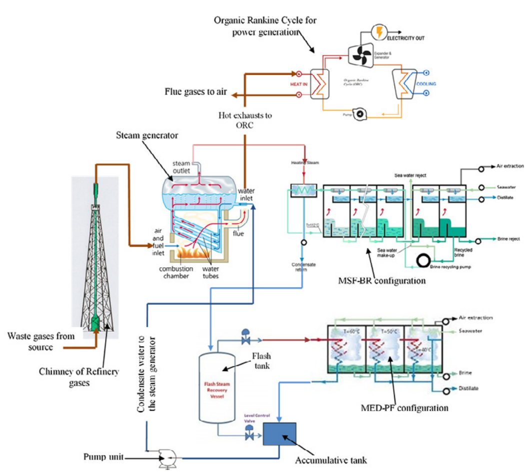
Mohamed A. Sharaf Eldean [a,*], A.M. Soliman [b]

[a] Department of Engineering Sciences, Faculty of Petroleum and Mining Engineering, Suez University, 43721 Suez, Egypt
[b] Mechanical Engineering Department, Faculty of Engineering, Al Jouf University, Sakaka, Saudi Arabia

HIGHLIGHTS

- Waste gases from oil refinery plants are burned for thermal desalination and electric power generation.
- Heat from waste gases powered on ORC and gas turbine cycle.
- Hybrid MSF and MED is used in the performed work.

GRAPHICAL ABSTRACT



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ABSTRACT

Thermal desalination plants need large amounts of fuel to desalinate large quantities of seawater. At the same time, burning non-beneficial gases in the oil refineries is considered a huge waste of energy instead of using it. In this paper, a novel study on the possibility of operating the thermal desalination plants by waste gases that emerged from oil refineries rather than burning these gases in the air is performed. Hybrid MSF-MED thermal desalination processes are utilized in this study to produce a total range of 100–40,000 m^3/day. Three scenarios are performed utilizing the waste gases with MSF-MED. The comparison brings out that using waste gases would save roughly 1136 \$/h (UHC-unit hourly costs, \$/h) while comparing against the conventional natural gas operation. Moreover; 5 m^3/h of waste gases would produce an amount of 58–60 MW of electric power combined with a production of 100 m^3/d of fresh water (gas turbine cycle scenario) and 4.5–5 MW combined with a production of 40,000 m^3/d in case of organic Rankine cycle operation. Based on energy and exergy balances, the 3rd scenario gives remarkable results.

* Corresponding author.
E-mail addresses: mohammed.eldeen@suezuniv.edu.eg, mwahab31@yahoo.com (M.A. Sharaf Eldean), ahsoliman69@hotmail.com (A.M. Soliman).

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Nomenclature

A	cross-sectional area, m^2
BH	brine heater unit
BPR	boiling point ratio, °C
Cp	specific heat capacity, kJ/kg°C
CC	combustion chamber unit
$Comp$	compressor unit
CV	calorific value, kJ/kg
D	diameter, m
Ex	exergy rate, kW
GR	gain ratio = M_d/M_s
H_L	head losses, m
$H.P.T$	high pressure turbine unit
h	enthalpy, kJ/kg
Iex	exergy destruction rate, kW
L	latent heat of vaporization, kJ/kg
$L.P.T$	low pressure turbine unit
M	mass flow rate, kg/s
MED	multi effect distillation
MSF	multi stage flash
N	number of stages, or effects
NEA	non-equilibrium allowance, °C
ORC	Organic Rankine cycle
P	pressure, bar
PR	performance ratio
Pr	pressure ratio, bar
Q	thermal power, kW
Re	reynolds number
S	salinity ratio, g/kg
SFC	specific fuel consumption, kg/h/kW
T	temperature, °C
TBT	top brine temperature, °C
U	overall heat transfer coefficient, kW/m^2 °C
UHC	unit hourly costs, $Z_{unit}^{IC\&OM}$, \$/h
V	velocity, m/s
W	work, kW
WGC	Waste Gas Chimney
Z	level, m

Subscripts

a	air
b	brine
cc	combustion chamber
$comp$	compressor
$cond$	condenser unit
cw	cooling water
d	distillate
ex	exhaust, exergy
evp	evaporator unit
f	feed
fsh	flash tank
g	generator
$H.P.T$	high pressure turbine
i	inlet
$L.P.T$	low pressure turbine
m	mechanical
MED	multi effect distillation
MSF	multi stage flash
N_{stg}	number of stages in MSF
N_{eff}	number of effects in MED
n	last stage
o	out
orc	Organic Rankine cycle
p	pump
r	recycle stream
rec	recuperator
s	steam, isentropic
sea	tend to sea
sg	steam generator
t	turbine, tube
v	vapor
w	exergy work, kW
wg	waste gases

Greek

ρ	density, kg/m^3
ε	effectiveness
η	efficiency
γ	isentropic index

1. Introduction

Water shortfalls in the Arab countries are one of the problems that hamper growth in these, especially in the states of North Africa countries. Desalination of saline (sea/brackish) water is one of the most promising techniques to overcome water shortages in a considerable number of states. Multi stage flash (MSF) and multi effect distillation (MED) are considered a vital option to solve the water shortage problem from the perspective of thermal power. MSF-BR and MED-PF configurations have a gain ratio ranged as 11.5 and 20 respectively with a share capacity around 95,000 m^3/d with specific power consumption ranged between 1_{med}-4_{msf} kW h/m^3 [1]. To produce such large quantities of fresh water, large amounts of thermal power that conventionally represented by the fossil fuel are urgently needed which are already available in abundance. Entirely the same; with the volatility of fossil fuel prices and the continued high prices, fossil fuels remains a problem for thermal desalination plants. At the same time, the oil refining plants produces large amounts of waste gas which is burning around the clock in the air so enormous thermal energy waste. It is calculated that an amount of 960 m^3/d of waste gases would be fired in the air (Egypt case study, 5 m^3/h per each plant [2]). Burning waste gases cause many severe problems, including, for example:

- They contain large quantities of sulfur.
- Air pollution is constantly, especially when flame failure generated hydrogen sulfide.
- Produce large amounts of carbon monoxide and nitrate compounds.
- Produce large amounts of heat energy when incinerated.

Thus, it gets more urgent for the use and recycling of these waste gases in the propagation of thermal energy that will run thermal desalination plants and create electricity. The average flow rate of flue gases is about 5 m^3/h [2] which contains hydrogen sulfide (H_2S), Hydrogen (H_2), Methane (CH_4), Ethane (C_2H_6), Ethylene (C_2H_4), Propane (C_3H_8), Propene (C_3H_6), Butane (C_4H_{10}), Butene (C_4H_8), Pentane (C_5H_{12}), and Pentene (C_5H_{10}). Other than the gas hydrogen sulfate, the rest of the gas has a high calorific value. The norm of the calorific value is about 45,000 kJ/kg, meaning of this a huge amount of thermal energy is completely misplaced and in addition increasing the pollutant contents to the surroundings. Regaining of the waste gas is reasonably economic and has close to environmental benefits. Most of recent researches are focused on the utilization of waste heat not waste gases in power generation, such as a gas turbine cycle, steam cycle and organic Rankin cycle [3–10]. It also practiced as a heat source for absorption refrigeration cycles [11,12]. The retrieval of waste heat for water desalination was investigated via [12–18]. Dexin Wang [19] developed a water recovery technology extract a portion of the water vapor and its latent heat from flue gases based on a nano-porous ceramic membrane capillary con-condensation separation mechanism. However; the permeate wasn't exceeded over 4.5 kg/m^2 h. Dexin [19] technology was particularly beneficial for coal-fired power plants that use high-moisture coals for flue gas cleanup. Lu Zisheng [20] showed a three types of sorption refrigerators which are driven by waste gases from diesel engine and cooled by seawater. Zisheng [20] configurations used the gases waste heat (via heat exchanging) to increase the COP of the addressed cycle. Another application related to the waste heat that emerged from waste gases was presented by Wei He [21]. Wei's work focused on the thermoelectric generator performance by the use of waste heat that emerged from waste gases. Chengyu Li [22] investigated how to use waste heat recovery for different types of power cycles (e.g., Rankine cycle, trans critical cycle and combined cycle). Li's work [22] focused on working fluid selection and heat exchanging from waste gases. The power produced wasn't exceeded over 150 kW [22]. Naijun [23] performed an experimental work to produce 400 W from ORC by the aid of waste heat emerged from waste gases heat exchanging. It is obvious from the literatures that:

- Most of the previous works are produced small amounts of fresh water, energy and power.
- The previous works are focused on some specific cases.
- All literatures are shared about how to use waste heat from waste gases for ORC, thermos-electric generator, cooling, heat exchanging, etc. None of them mentioned how to burn waste gases to produce massive power and/or fresh water.
- Regarding to the desalination matter, the published researches have shown how to use thermal energy waste to power on thermal desalination plants, but none of them progressed to no mention of the role of the oil refinery waste gases as a specific case.

This work wasn't investigated before. The novelty of the current work is depending on why and how to utilize and burn the waste gases from the oil refineries plants to produce large amounts of fresh water and electric power. The main contribution of the work is to present a new applicable idea to solve the power and water shortage by the use of wasted gases. The primary idea of this study is grounded on the use of oil refinery plant waste gases in order to power on thermal MSF-MED. In this study, three scenarios are performed and simulated for the purpose of waste gases from oil refinery plants. The validated REDS [24–26] is utilized to simulate the proposed scenarios. The first scenario is combining MSF-MED with steam generator unit which is fired on by waste gases instead of conventional heat source (natural gas saving). A flash cyclone tank is the primary link between the MSF and MED. The second scenario is regarded the same as the first, however; the outlet exhausts from the steam generator is utilized to run the organic Rankine cycle (ORC). Waste heat from burning operation would operate the ORC to generate power for the pumping system. The third scenario is to operate the MSF by the outlet exhausts from gas turbine cycle (GTC). Energy and exergy analyses are performed for the selection of the most reliable scenario. The estimate of this study may be pinpointed as follows:

- Study of using oil refinery plants waste gases for power generation and seawater desalination instead of burning it into the air (case study: Suez Oil Refinery Co. [27]).
- Energy, mass, exergy and cost analyses are performed.
- A comparison between the three scenarios and then the best choice in terms of saving energy, exergy and water production would be considered.
- Comparison between the proposed scenarios and the conventional operation of MSF-MED under the same operating conditions.
- Measuring the performance of MSF-MED is implemented in this work.
- A Visual simulator program is established to simulates the three scenarios in order to facilitate the process of calculating the energy.
- The operating conditions data points are fixed at reasonable values to ensure optimized results.
- The effect of many different factors on productivity and generate electricity is performed.
- The most alternative solution based on the best scenario is elected in this study.

2. Waste gases problem and analysis

The environmental impact of crude has been often negative because it is toxic to virtually all sorts of animation and its extraction fuels causes hazards and the climate change. Oil refineries cause smog and air contamination. It pollutes at unacceptable, unhealthy levels. Moreover; oil refineries emit many gases like sulfur dioxide (SO_2), nitrogen oxide (NO_2), carbon dioxide, carbon monoxide, methane, dioxins, hydrogen fluoride, chlorine, benzene and others. The problem has been raised when officials in the Suez company for manufacturing and petroleum refining [27] wanted to find out an alternative solution to get rid of the waste gas. Sometimes it blew out the flame, causing leakage of large amounts in the air with an unpleasant smell. Therefore; we have been cooperating to discover a resolution to this problem. Since the gas is burned in the air without any benefit, a huge energy is considered wasted in the air every day (43,800 ton/year). Proposals have focused on the following:

Installation of gas filters to gather up the maximum quantity of sulfur then to be used for other useful industries.
Burning gas to hot up the steam to rotate adjusted a desalination plant instead of burning in the breeze.
Consider the possibility of benefiting from the outside exhaust from burning in order to run the electric power station.

Table 1 shows the gas analysis that been extracted from the gas chimney. The gas mass flow rate is about 5 m^3/h (120 m^3/day), and the discharge pressure is in the range of 1.2–1.5 bar.

3. Scenarios and systems descriptions

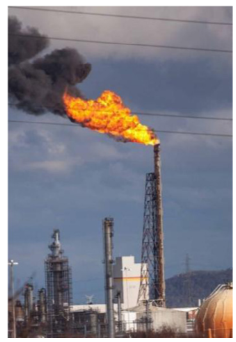
3.1. The 1st & 2nd scenarios (WGC-MSF-MED & WGC-MSF-MED-ORC)

The 1st proposed system is modeled and designed in order to use the waste gases to go off along the steam generator unit which would generate the sufficient steam to the brine heater unit. The system elements are listed as follows:

- Waste gas chimney (WGC): would deliver an amount of 5 m^3/h of waste gases to the steam generator.
- Steam generator (SG): would generate the needed steam to the brine heater unit.
- Brine heater (BH): the generated steam by the steam generator would transfer its latent heat of vaporization through the brine heater unit which would transfer the thermal energy to the preheated seawater comes from the MSF-BR plant.

Table 1
Compositional analysis of waste gases of Suez Oil Refinery plant [27].

Component, vol.%	Coker gases	Coker distillate unifining gases
H_2S	9.85	38.2
Hydrogen	4.05	10.88
Methane	21.17	4.5
Ethane	17.23	14.79
Ethylene	1.85	0.0
Propane	14.51	17.03
Propene	5.65	0.0
i-Butane	1.95	5.62
n-Butane	7.11	4.85
1-Butene	3.68	0.05
Neo-Pentane	1.83	0.01
i-Pentane	1.86	1.78
n-Pentane	2.77	1.02
1-Pentene	2.07	0.0
C_6+	4.41	1.27
CV, kJ/kg	45,000	
Cp, kJ/kg°C	2	
A/F	12.5	
Density, kg/m^3	1.5	
Mass flow rate, m^3/h	5	
Pressure, bar	1.2–1.5	
Dynamic viscosity, Pa s	2e−5	



– Multi stage flash brine recycle plant (MSF-BR): the plant productivity, top brine temperature, number of stages and seawater salinity ratio are assigned. The condensed steam from the brine heater would be delivered to the flashing tank.
– Flash tank (FSH): The condensed steam from the brine heater would be flashed at lower pressure in the flash tank unit. The flashing steam side would power on the MED-PF unit where the rest of water side would accumulate at the bottom of the tank.
– Multi effect distillation parallel feed plant (MED-PF): The productivity, seawater temperature, feed salinity, and number of effects are assigned. The condensed steam in the first effect would be mixed with the accumulated water from the flashing tank bottom then pumped to the steam generator unit.
– Circulating pump (P): the pump unit is to deliver the accumulating water to the steam generator unit.

Fig. 1 shows the schematic diagram of the 1st scenario of the proposed system. The 2nd scenario of the proposed system is considered the same the first however; organic Rankine cycle (ORC) is used in order to increase the benefit from the thermal energy from combustion gases emerged from the steam generator. The exhaust gases from the steam generator would power on the ORC. Toluene working fluid is recommended and used as a working fluid through the ORC [24–26,28]. Fig. 2 shows a schematic diagram of the 2nd proposed system. The ORC components are listed as follows:

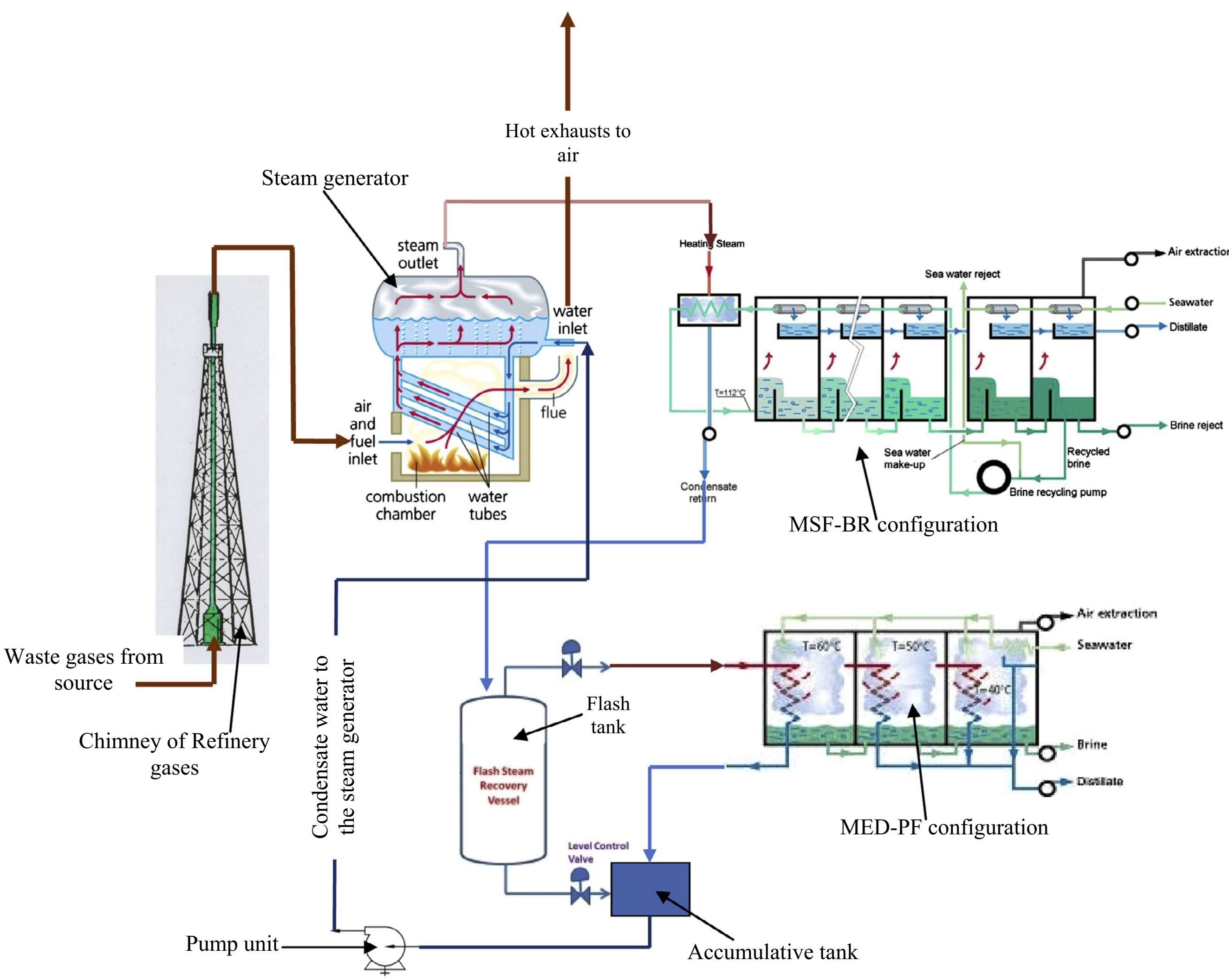


Fig. 1. The schematic diagram of the 1st scenario.

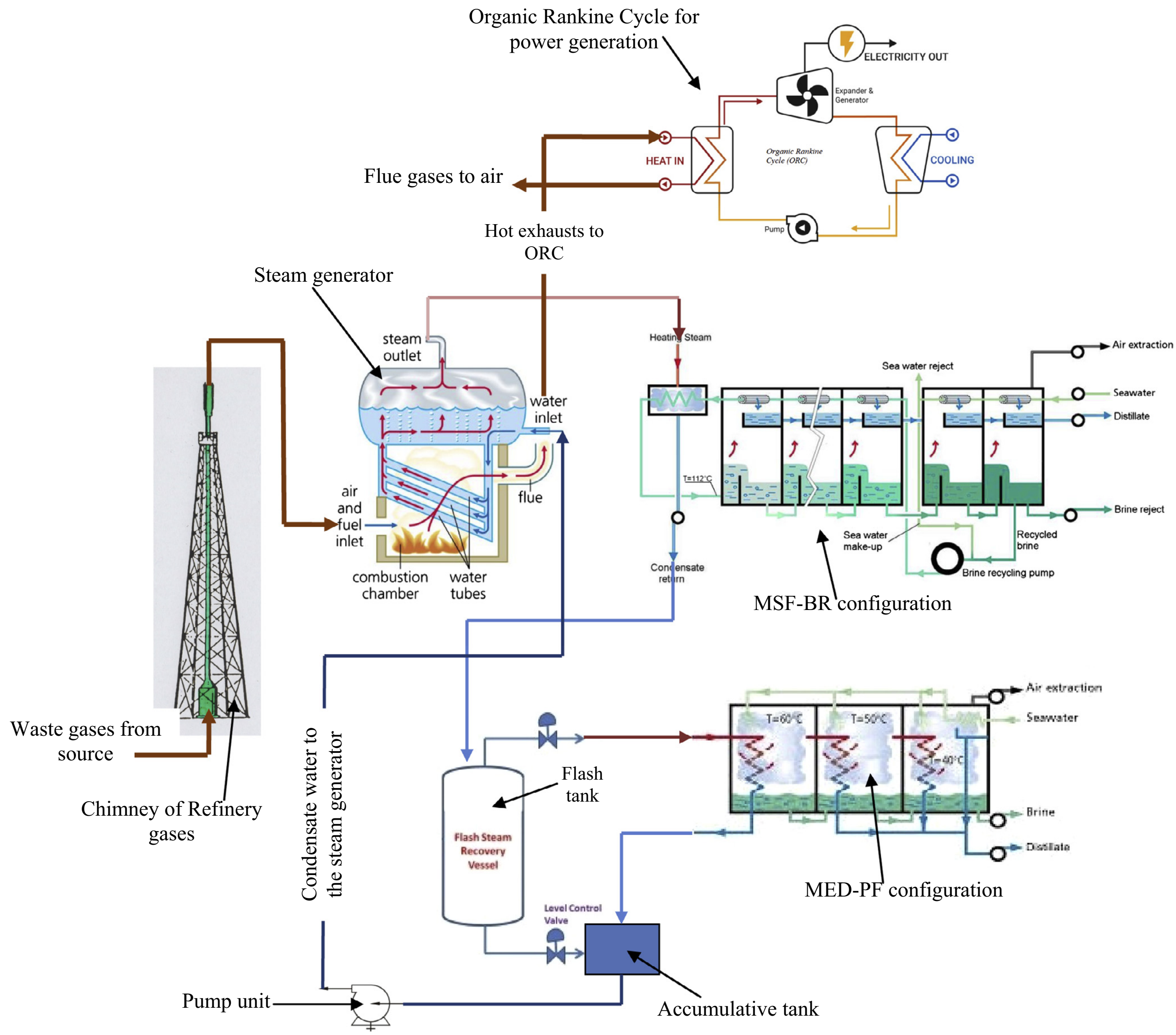


Fig. 2. The schematic diagram of the 2nd scenario.

– Heat exchanger evaporator unit (Hevp): the exhaust gases emerged from the steam generator would transfer its thermal power and passion to the toluene working fluid through the evaporator unit.
– Turbine (T): organic Rankine turbine would power on all pumping systems in MSF and MED. The needed power, kW is considered as the warhead on the organic turbine unit.
– Recuperator (REC): the outage steam (toluene) from the turbine is still in the super heat region, according to the positive side on the T-S diagram [28,29]. Therefore; recuperator unit would transfer the thermal power to pumped liquid stream from pump unit that goes to the evaporator heat exchanger unit.
– Condenser (Cond): condenser will condensate the steam emerged from the recuperator unit before pumping operation to the evaporator.
– Pump (P): the pump would deliver the liquid stream for the recuperator unit before going to the evaporator unit.

The governing equations for the 1st and 2nd scenarios are listed in Table 2. All physical properties of Toluene working fluid (ORC) are presented in the Appendix A.

3.2. The 3rd scenario (WGC-MSF-GTC)

The 3rd proposed scenario is differing from the 1st and the 2nd. It is anticipated to produce the electric power and water by the aid of gas turbine cycle (GTC). The waste gases from the refinery plant would fire on the combustion chamber adding by this more thermal power (heat addition) to the stream goes to the high-pressure gas turbine. The exhausts from the H.P.T would power on the L.P.T then the rest of exhausts would power on the brine heater for the MSF-BR plant. The system components are listed as following:

– Waste gas chimney (WGC): to deliver the required waste gases to the combustion chamber unit.
– Gas compressor (G_{Comp}): to compress the air and increasing the pressure stream that going to the combustion chamber.
– Combustion chamber (CC): increasing the thermal power of the gases (heat addition) up to the H.P.T operation.
– High pressure gas turbine (H.P.T): to deliver the required power to the gas compressor.

Table 2
1st & 2nd scenarios equations.

Waste gases chimney	
Chimney tube cross sectional area m^2, waste gases flow velocity and pressure are calculated as follows: $A_t = \frac{\pi}{4} \times D_t^2$, m^2, and $V_{wg} = M_{wg}/(\rho_{wg} \times A_t)$, m/s, $P_{wg} = (\frac{V_{wg}^2}{2g} + H_L + dZ) \times g \times \rho_{wg}$, bar, and $H_L = 64/Re$	1
Steam generator	
The waste gases mass flow rate kg/s, $M_{wg} = \frac{M_{total} \times \Delta h}{CV \times \eta_{sg}}$	2
Air mass flow rate kg/s: $M_a = \frac{A}{F} \times M_{wg}$, where *A/F* is the air to fuel (gases) ratio	3
Exit exhausts temperature °C: $T_{exh} = \frac{(M_{wg} \times CV) - (M_{total} \times \Delta h)}{(Mwg + Ma) \times Cp_{wg}}$, where *CV* is the gases calorific value, kJ/kg, and Cp_{wg} is the specific heat capacity of the waste gases, kJ/kg°C	4
The waste gases thermal power kW: $Q_{wg} = M_{wg} \times CV \times \varepsilon_{sg}$	5
The specific gas consumption, kg/h/kW: $SFC = 3600 \times M_{wg}/Q_{wg}$	6
Brine heater	
Heat rejection kW: $Q_{bh} = M_{cw-MSF} \times \Delta h_{cw-MSF}$	7
Inlet steam temperature °C: $T_{si} = \frac{T_{cwo} - T_{cwi}}{\eta_{bh}} + T_{cwi}$	8
Steam mass flow rate, kg/s: $M_{total} = \frac{Q_{bh}}{\eta_{bh} \times \Delta h_s}$	9
Multi Stage Flash brine recycle configuration	
For known distillate product M_d, feed stream M_f to the mixer unit is obtained, kg/s: $M_f = (S_b/(S_b - S_f)) \times M_d$	10
Total needed feed (M_{ft}) based on 1st splitter ratio: $M_{f_t} = M_f/SPL1$, Therefore the rest of feed loss: $M_{fl} = M_{f_t} - M_f$, and the brine blow-down loss: $M_b = M_f - M_d$	11
Stage temp drop based on top brine temperature (TBT), last stage brine temperature (T_n) and number of stages (N): $T_{stg} = TBT - T_n/N$	12
The recycle brine flow rate M_r, kg/s and latent heat L, kJ/kg is then calculated: $Y = (C_p \times T_{stg})/L, M_r = M_d/(1 - (1 - Y)^N)$	13
Then the salinity ratio g/kg of the recycle stream is calculated S_r: $S_r = \frac{(S_f \times M_f + (M_r - M_d) \times S_b - M_b \times S_b)}{M_r}$	14
The outlet temperature of the distillate product T_d could be calculated based on brine blow down temperature T_n, non equilibrium allowance *NEA*, and boiling point ratio *BPR*: $T_d = T_n - NEA - BPR$, the non-equilibrium allowance *NEA* and *BPR* are calculated by the following equations: $NEA = A + B \times T_n + C \times T_n^2 + D \times T_n^3$, Where $A = 2.556$, $B = -0.203 \times 10^{-1}$, $C = -0.129 \times 10^{-1}$, $D = 0.1123 \times 10^{-5}$ $BPR = (B + C \times S) \times S$, Where S is the stream salinity and, $10^3 \times B = 6.71 + 6.43 \times 10^{-2} \times T_n + 9.74 \times 10^{-5} \times T_n^2$ $10^5 \times C = 2.38 + 9.59 \times 10^{-3} \times T_n + 9.42 \times 10^{-5} \times T_n^2$	15
For the heat recovery and rejection sections, the overall heat transfer coefficient based on vapor temperature T_v: $U = 1.7194 + 3.2063E - 3 \times T_v + 1.5971E - 5 \times T_v^2 - 1.9918E - 7 \times T_v^3$ [23]	16
Flash tank	
The flashing dryness fraction: $X_{fsh} = \frac{M_{s-med}}{M_{total}}$	17
Unvaporized water kg/s: $M_w = (1 - X_{fsh}) \times M_{total}$	18
The flashing enthalpy for the MED side, kJ/kg: $h_x = h_{f-med} + X_{fsh} \times (h_{g-med} - h_{f-med})$	19
Multi Effect Distillation parallel feed configuration	
Energy balance for the condenser unit based on the specified effectiveness ε: $T_f = \varepsilon \times (T_v - T_{sea}) + T_{sea}$, where T_v is the vapor temperature; and the distillate temperature is obtained from the same equation: $T_d = T_v - (\varepsilon \times (T_v - T_{sea})$	20
Mass and material balances: $M_f = M_d \times S_b/(S_b - S_f)$, $M_b = M_d \times S_f/(S_b - S_f)$	21
And steam flow rate M_s could be obtained from the following relation: $M_s = M_d/PR$ where *PR* is the performance ratio which is also obtained as following: $PR = \frac{L(T_s)}{L(T_v) + C_p(T_{av}, S_f) \times (T_v - T_f) \times \frac{S_b}{(S_b - S_f)} + \frac{S_f}{S_b - S_f} \times C_p(T_{av}, S_f) BPE(T_b, S_b)}$	22
The *BPE* is the boiling point elevation as a function of brine temperature and salinity percent; cooling water blow down from the condenser unit is obtained from the following energy balance relation: $M_{cw} = \frac{M_d \times L(T_v)}{C_p(T_{av}, S_f) \times (T_f - T_{sea})} - M_f$, Where the T_{av} is the average temperature, °C, for the feed seawater across the condenser unit ($T_{av} = \frac{T_f + T_{sea}}{2}$)	23
Therefore; the total mass flow rate is then calculated: $M_{ft} = M_{cw} + M_f$	24
Pump	
Pump work kW: $W_p = M_{total} \times \Delta P/\rho \times \eta_p$	25
Enthalpy of saturated liquid water, kJ/kg: $h_{po} = (W_p/M_{total}) + h_{pi}$	26
ORC heat exchanger evaporator	
Evaporator thermal power Q_{hev} based on enthalpy difference (Toluene working fluid): $Q_{hev} = M_{orc} \times \Delta h$	27
Effectiveness of heat exchanger evaporator unit: $\varepsilon_{hev} = \frac{T_{vo} - T_{vi}}{T_{exi} - T_{vi}}$	28
Outlet temp of Waste Gases (exhausts), °C: $T_{exo} = T_{exi} - \varepsilon_{hev} \times T_{exi} - T_{vi}$	29
ORC turbine	
Outlet enthalpy of the turbine outlet, kJ/kg: $h_{to} = h_{ti} - \eta_t \times \eta_g (h_{ti} - h_{tos})$, the turbine power is depending on the desalination part pumping system	30
Calculate ORC mass flow rate kg/s: $M_{orc} = W_t/(h_{ti} - h_{to})$	31
ORC recuperator	
Outlet recuperator temperature to the condenser unit, °C: $T_{ro} = T_{ri} - \varepsilon_{rec} \times (T_{ri} - T_{ri-p})$	32
The recuperator thermal power, kW: $Q_r = M_{orc} \times \Delta h$, then the outlet recuperator enthalpy is calculated: $h_{ro-p} = \frac{Q_r}{M_{orc}} + h_{ri-p}$	33
ORC condenser	
Heat rejection kW: $Q_{cond} = M_{orc} \times \Delta h$	34
Outlet cooling water temperature, °C: $T_{cwo} = \varepsilon_{cond} \times (T_{condi} - T_{cwi}) + T_{cwi}$	35
Cooling water mass flow rate kg/s: $M_{cw} = Q_{cond}/\Delta h_{cw}$	36

– Low pressure gas turbine (L.P.T): to deliver the required power for the main grid or to the production sector. The L.P.T exhausts would power on the brine heater of the MSF plant.
– Brine heater (BH): would receive the L.P.T exhausts to exchange the thermal heat with the seawater side for the MSF-BR plant (heat rejection process).
– Multi stage flash (MSF-BR): would produce the required fresh water via brine heater unit as a source of heat.

Fig. 3 shows a schematic diagram of the 3rd proposed system. Fig. 4. represents a schematic diagram of the gas turbine cycle on

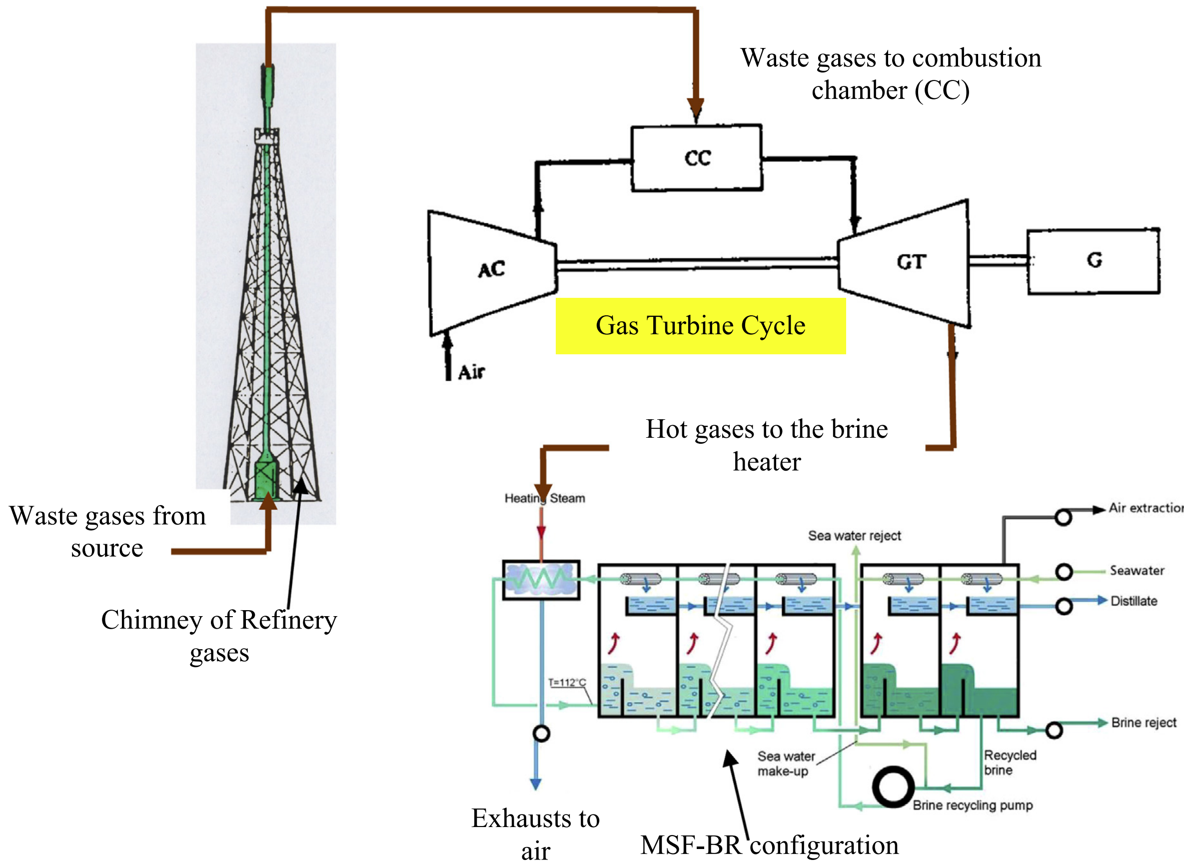


Fig. 3. The schematic diagram of the 3rd scenario configuration.

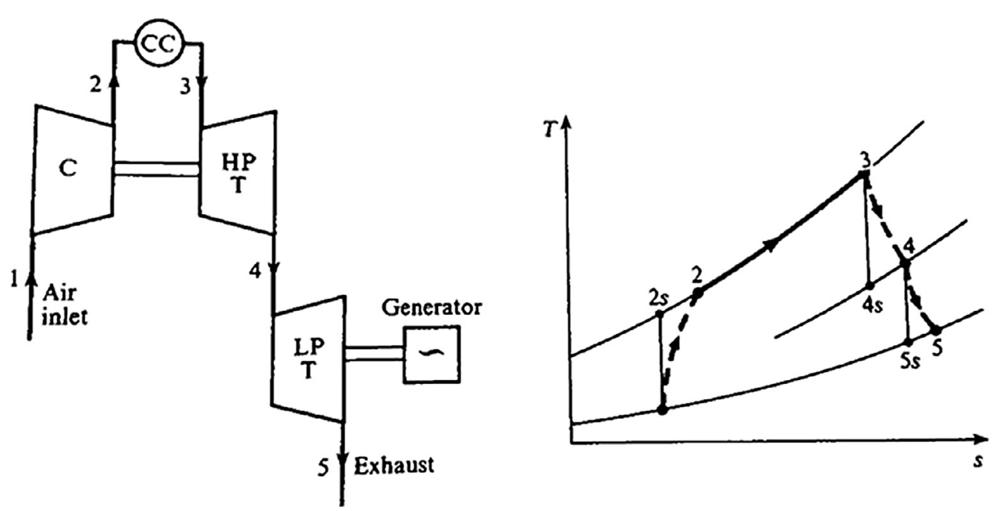


Fig. 4. The gas turbine cycle for the 3rd scenario on the T-S diagram.

the T-S diagram. Table 3 shows the governing equations that represents the 3rd scenario.

4. Modeling, simulation and assumptions

The proposed scenarios in this work require an iterative program in order to work out the complicated streams (recycle and backward streams). Hence; the authors used REDS software library [25,28] in order to make the projected scenarios. Three models are built according to the proposed configurations. The models are built according to design calculation method. The system border streams (outlet temperature, ambient temperature, inlet cooling water temperature, etc.) are assigned by the user than the entire design data (area, length, volume, mass flow rate, etc.) Will then be calculated. Therefore; a user would assign the amount of needed fresh water from the desalination plant then all possible or required data for all the system units would be calculated in sequence. Specifying the system productivity would calculate the required thermal load (in case of MSF). Besides; the required design limits and performance calculations would be pass out

Table 3
The 3rd scenario equations.

Waste gases chimney	
Chimney tube cross sectional area m^2, waste gases flow velocity and pressure are calculated as follows:$A_t = \frac{\pi}{4} \times D_t^2$, m^2, and $V_{wg} = M_{wg}/(\rho_{wg} \times A_t)$, m/s, $P_{wg} = (\frac{V_{wg}^2}{2g} + H_L + dZ) \times g \times \rho_{wg}$, bar, and $H_L = 64/Re$	37
Gas compressor	
Outlet isentropic temperature, K: $T_{os} = T_i \times Pr^{\frac{\gamma-1}{\gamma}}$, the compressor power is calculated by the meaning of isentropic and mechanical efficiencies, kW: $W_{comp} = \frac{M_{total} \times Cp \times (T_{os} - T_i)}{\eta_{comp} \times \eta_m}$, the outlet compressor temperature K is then calculated: $T_o = \frac{W_{comp}}{Cp \times M_{total} + T_i}$	38
Combustion chamber	
The waste gases mass flow rate kg/s: $M_{wg} = \frac{M_{total} \times Cp \times \Delta T}{CV \times \varepsilon_{cc}}$, where *CV* is the calorific value of the waste gases and air mass flow rate kg/s is then calculated: $M_a = \frac{A}{F} \times M_{wg}$	39
The combustion chamber power kW: $Q_{cc} = M_{wg} \times CV \times \varepsilon_{cc}$	40
The specific fuel (waste gases) consumption kg/h/kW: $SFC = 3600 \times M_{wg}/Q_{cc}$	41
High Pressure Turbine (H.P.T)	
H.P.T top cycle temp, K: $T_{ti} = \frac{W_{H.P.T}}{M_{total} \times Cp \times T_{to}}$, the H.P.T power is assigned by the MSF pumps and the compressor power.	42
Isentropic outlet temperature, K: $T_{tos} = T_{ti} - \frac{W_{H.P.T}}{\eta_t \times \eta_g \times M_{total} \times Cp}$	43
Outlet turbine pressure, bar: $P_{to} = \frac{Pr}{(\frac{T_{ti}}{T_{tos}})^{\frac{\gamma}{\gamma-1}}}$	44
Low Pressure Turbine (L.P.T)	
Inlet turbine temperature, K: $T_{ti} = T_{to} + \frac{W_{H.P.T}}{M_{total} \times Cp}$, where the turbine power is assigned by the user or the load demand.	45

instantly. For thermal configuration, the thermal load would calculate the mass flow rate and the considered physical properties. Figs. B.1–B.3 in Appendix B show the model interface of the proposed scenarios. The modeling assumptions are listed in Table 4. Physical properties and operating conditions are stored in lookup tables. The argument functions would call the data stored in the lookup tables. Saturated liquid and vapor phases of pressure, temperature, enthalpy, specific volume, and specific entropy are stored behind the modeled blocks. The source of physical properties is obtained from NIST [30] web chemistry book. The n-D Lookup Table block evaluates a sampled representation of a function in n variables $y = f(x1, x2, x3, \ldots, xn)$ where the function f can be empirical. The block maps inputs to an output value by looking up or interpolating a table of values you define with block parameters. The example figure of Lookup Tables is presented in the Appendix B (Fig. B.4). Generally, optimization process has been done in order to bring down the costs and techno-economic solutions. Desalination plants (MSF and MED) were optimized in order to lower the thermal loads on the brine heater for MSF and first effect for MED. Optimization for the 3rd scenario would reduce the mass flow rate of gases through the gas turbine cycle, thence; reducing the power of the gas compressor unit.

5. Results and discussions

5.1. Results of 1st & 2nd scenarios

It has to any designer to take into account the cost of raw materials used. Then it must be the work of modeling and optimization processes in order to reduce raw materials, especially to increase productivity while handling thermal power plants. In this study, if any effect and improvement of the operational conditions of the desalination plants would affect positively to what followed from operations and units. It is so important to know the impact of the number of stages and/or effects on productivity for both MSF and MED because it would reduce or increase the percentage of steam. Moreover; Increasing the thermal load is followed effecting by this on the areas, dimensions, and the percentage of fuel consumption. The gain ratio ($GR = M_d/M_s$) is an important term to decide how to optimize the desalination process. Therefore; higher values of *GR* would be considered. Fig. 5 shows the effect of increasing number of stages$_{msf}$ or effects$_{med}$ on the *GR*. The *GR* for MED-PF is noticed at 11.3 for 12 effect against 5.7 for 6 effects.

Furthermore; it achieves 9.7 at 40 MSF-BR stages against 2.43 for 10 stages of MSF-BR. Therefore; it depends on the designers'

Table 4
Data assumptions for the proposed scenarios.

Unit process	Assigned data	Calculated data
MSF-BR	• TBT, °C = 90–140 (∼)! • Seawater temperature, °C = 27 • Last stage temperature, °C = 40 • Seawater salinity ratio, ppm = 45,000 • Brine blow down salinity ratio, ppm = 70,000 • Plant productivity, m^3/day = 18,000 (∼)! • Feed seawater splitter ratio = 0.482 • Total number of stages = 40 (heat recovery = 37/heat rejection = 3) (∼)! • Pumping system efficiency, % = 75 • Brine heater effectiveness, % = 80	– Total feed, make up feed, kg/s – Brine loss, kg/s – Total brine loss profile, kg/s – Recycle stream, kg/s – Salinity profile, ppm – Distillate profile, kg/s – Brine temperature profile, °C – Feed and recycle temperature profiles, °C – Vapor temperature profile, °C – stages dimensions – Tubes heat transfer area, m^2 – Pumping system power load, kW – Pumping system pressure, bar – Brine heater thermal power, kW – Total cycle steam flow rate, kg/s – Top steam temperature, °C – Brine heater area, m^2 – Performance ratio

Table 4 (*continued*)

Unit process	Assigned data	Calculated data
Steam Generator	• Effectiveness, % = 80 • A/F = 12.5 • Waste gases CV, kJ/kg = 45,000	– Total gases flow rate, kg/s – Air mass flow rate, kg/s – Thermal power, kW – Exhaust gases temperature, °C – Outlet steam temperature, °C
Waste Gas Chimney	• Waste gases density, kg/m^3 = 1.5 • Dynamic viscosity, Pa s = 2e−5 • Funnel tube diameter, m = 0.31	– Chimney cross sectional area, m^2 – Pressure drop, bar – Gases velocity, m/s – Pressure head losses, bar
Flash Tank	• Flashing pressure for MED, bar = 0.35 (0.12–0.4)! • Steam velocity, m/s = 2.5–45	– Steam inlet/outlet tube diameter, m – Height & width, m – Total tank volume, m^3 – MED flashing enthalpy, kJ/kg – Flashing dryness fraction, % – Tank bottom mass flow rate, kg/s
MED-PF	• Seawater temperature, °C = 27 • Brine blow down temperature, °C = 40 • Seawater salinity, ppm = 45,000 • Brine blow down salinity, ppm = 70,000 • Productivity, m^3/day = 20,000 (∼)! • Number of effects = 1:12 (∼)! • End condenser effectiveness, % = 80 • Pumping system efficiency, % = 75	– MED steam flow rate, kg/s – Feed profile flow rate, kg/s – Distillate profile, kg/s – Brine flow rate, kg/s – Salinity profile, ppm – Feed temperature profile, °C – Brine temperature profile, °C – Vapor temperature profile, °C – Distillate temperature profile, °C – Gain ratio – Effect heat transfer area, m^2 – Total effects heat transfer area, m^2 – End condenser area, m^2 – Pumping systems power, kW – Pumping system pressure, bar
Pump	• Steam cycle pump efficiency,% = 75	– Pump power, kW – Outlet stream temperature, °C – Outlet stream enthalpy, kJ/kg
Heat Evaporator (ORC)	• Hot side temperature difference, °C = 20–50 • Specific heat capacity of waste gases, kJ/kg°C = 2	– Effectiveness,% – Thermal power across tubes, kW – Outlet exhausts temperature, °C – Outlet vapor temperature, °C and enthalpy, kJ/kg – Outlet vapor pressure, bar – Heat transfer area, m^2
Turbine (ORC)	• Demanded load by the grid + Pumping system load, kW • Generator and turbine efficiencies, % = 95, 85	– ORC mass flow rate, kg/s – Outlet temperature and enthalpy
Recuperator (ORC)	• Effectiveness, % = 80	– Outlet stream temperature and enthalpy to the condenser – Outlet stream temperature and enthalpy to the evaporator – Thermal power, kW
Condenser	• Condensation temperature, °C = 35 • Inlet cooling water temperature, °C = 27 • Effectiveness, % = 80	– Cooling water mass flow rate, kg/s – Outlet temperature and enthalpy – Condenser pressure, bar – Thermal power, kW
Pump (ORC)	• ORC pump efficiency, % = 75	– Pump power, kW – Outlet stream temperature, °C – Outlet stream enthalpy, kJ/kg
Gas turbine cycle		
Gas Compressor	• Pressure ratio, bar = 5–10 • Efficiency, % = 85 • Mechanical efficiency, % = 95	– Adiabatic index – Compressor power, kW – Outlet temperature and enthalpy
Combustion Chamber	• Effectiveness, % = 80 • A/F = 12.5 • Waste gases CV, kJ/kg = 45,000	– Waste gases flow rate, kg/s – Air mass flow rate, kg/s – Thermal power, kW – Specific waste gases consumption, kg/h/kW
High Pressure Turbine	• Pressure ratio, bar = 5–10 • Pumping system power, kW • Efficiency, % = 85 • Generator efficiency, % = 95	– Adiabatic index – Outlet turbine pressure, bar – Top cycle temperature, K – Outlet isentropic temperature, K
Low pressure Turbine	• Ambient pressure, bar = 1.03 • Demanded power, kW • Efficiency, % = 85 • Generator efficiency, % = 95	– Adiabatic index – Pressure ratio, bar – Inlet turbine temperature, K – Outlet isentropic temperature, K
Notes	(∼)!: Variable input according to the optimized selection Data are run out based on steady state operating conditions Ambient temperature is fixed as 25 °C for all process runs Seawater temperature is fixed as 27 °C for all process runs Waste gases mass flow rate is fixed at 5 m^3/h The elevation of the waste gases chimney is fixed at zero level (no elevation)	

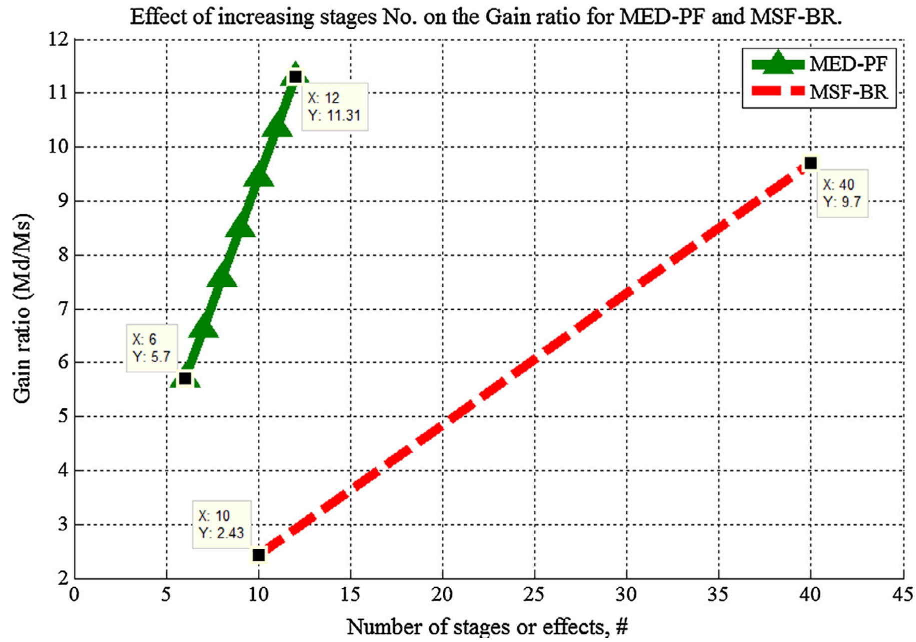


Fig. 5. Effect of number of stages or effects on the gain ratio ($GR = M_d/M_s$) for MED-PF and MSF-BR.

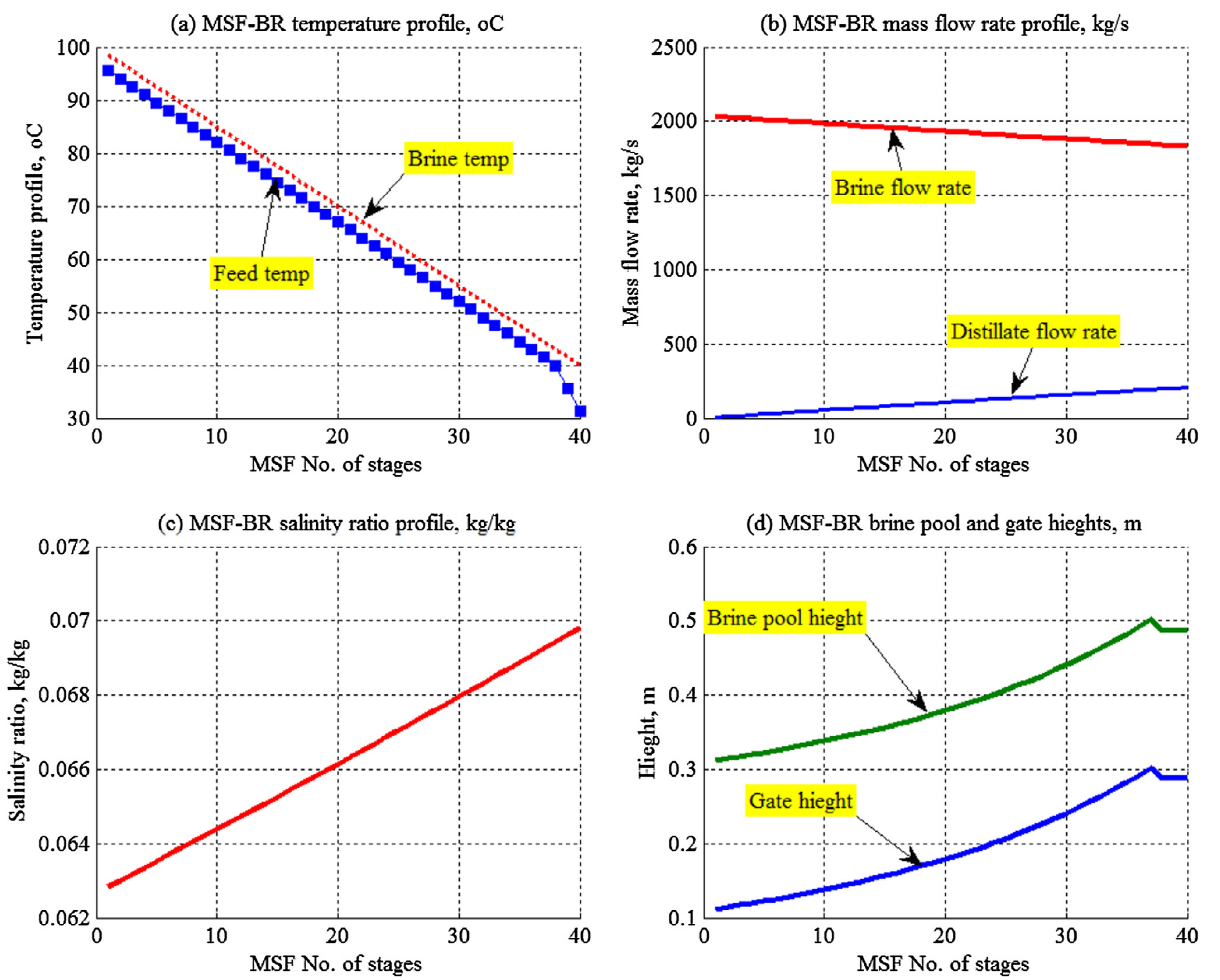


Fig. 6. The variations of different operating conditions according to the number of stages of the MSF-BR: (a) temperature profiles, (b) mass flow rate profile, (c) salinity profile, and (d) gate and brine pool heights.

decision about the reliable operating conditions, areas, and cost. However; increasing the number of effects gives an advantage to the desalination plant by reducing the total water price by increasing the *GR*. Fig. 5 shows that by increasing the number of effects, the GR will increase. Thence; the number of stages and effects for the MSF-BR and MED-PF are fixed at 40 and 12 respectively. The variations of some operating and design conditions against the number of stages of the MSF-BR is shown in Fig. 6. Fig. 7 shows the same variations per effect for the MED-PF where the number of effects was 12 effect. The profiles data are taken at 18,000 m^3/d and 20,000 m^3/d for MSF-BR and MED-PF respectively.

It is obvious from Figs. 5–7 that increasing the number of stages or effects has a massive effect on the process optimization such as *GR* parameter. However; top brine temperature (TBT, °C) has also a massive effect on valuable parameters such as waste gases flow rate, specific wastes consumption, specific steam consumption, and dryness fraction percentage. Fig. 8 shows the effect of both terms (N_{stg}, and TBT, °C) on the last-mentioned parameters. It is noticed in Fig. 8a that the effect of increasing the N_{stg} is massive comparing against the TBT effect on the M_{wg}. Increasing the N_{stg} was found decreasing the percentage of M_{wg} thence; the chance of increasing the productivity is increased too. On the contrary, increasing the TBT leads to an increase in the proportion of fuel waste gases and thus reduce productivity or increase the cost. This was made clear in the form of Fig. 8b where increasing gas consumption rate significantly, especially under the influence of TBT. Either in the form of Fig. 8c has been the effect of TBT is very clear to the rate of steam consumption, while the effect of the N_{stg} almost constant. For Fig. 8d, the effect of N_{stg} on the dryness fraction is remarkable where increasing the N_{stg} would increase the percentage of vapor over the percentage of mixture (from 30% @ N_{stg} = 10–90% @ N_{stg} = 40) thence; decreasing the costs over the flash tank related to the design considerations.

For MED-PF, Fig. 9a–d shows the effect of steam pressure, bar and N_{eff} on the waste gases flow rates, kg/s, specific fuel (wastes) consumption, kg/h/kW, specific steam consumption, kg/h/kW, and dryness fraction. In Fig. 9a:c, it is found that the effect of steam pressure on M_{wg}, SFC, SSC is remarkable. However; N_{eff} is highly effective on the flashing dryness fraction parameter. In Fig. 9a:c, increasing the steam pressure inlet to the MED-PF unit would decrease the M_{wg}, SFC, and SSC, meaning by this a reduction in technical cost by increasing the plant productivity. The value of 0.35 bar (steam temperature = 72.66 °C) is found more optimized for the all plant operation putting in consideration the upper operating conditions of the MSF-BR plant. The effect of N_{eff} on the related parameters is slightly remarkable however; N_{eff} is quite

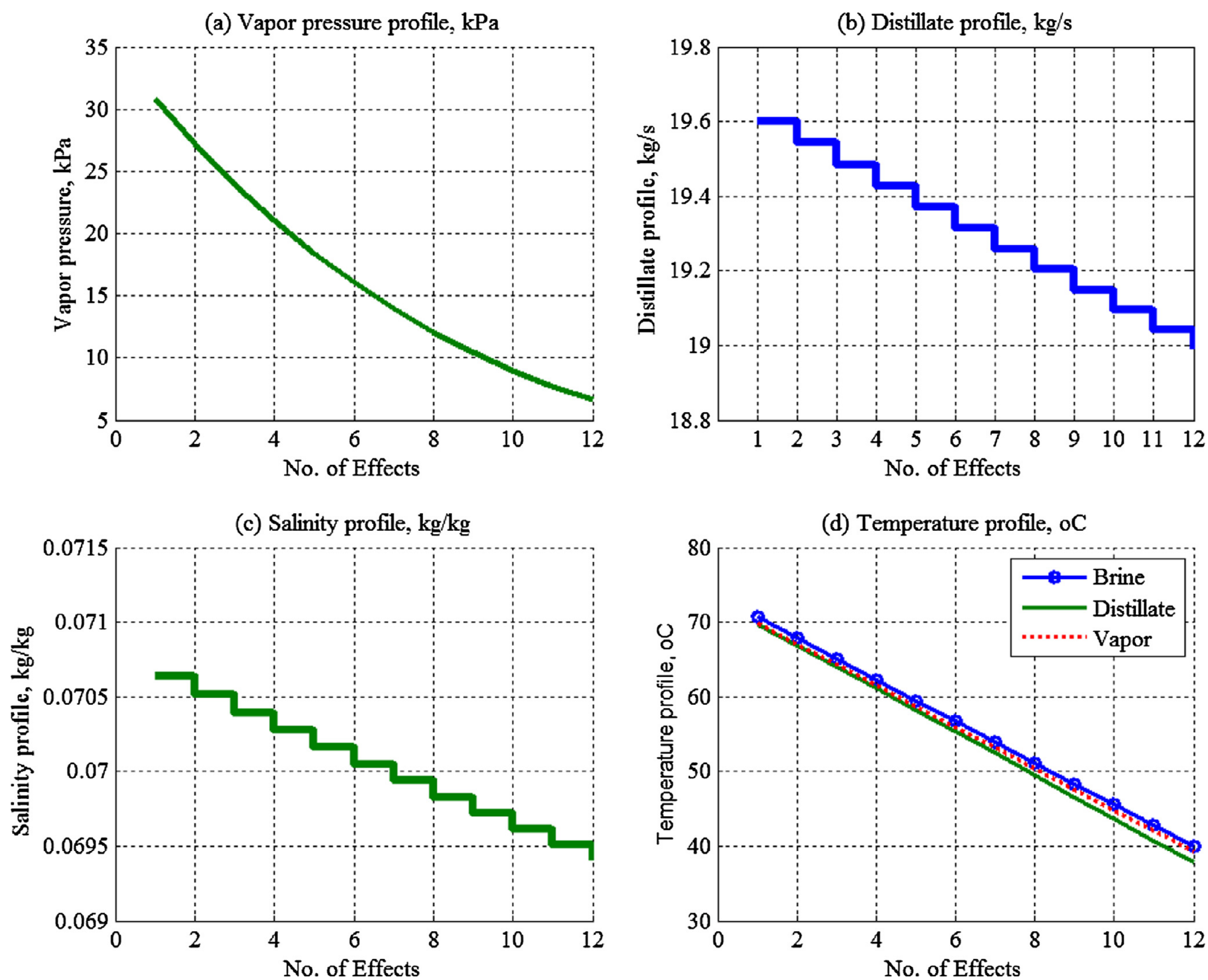


Fig. 7. The variations of some different operating conditions against the number of effects for the MED-PF: (a) vapor pressure, (b) Distillate profile, and (c) salinity profile, temperature profile.

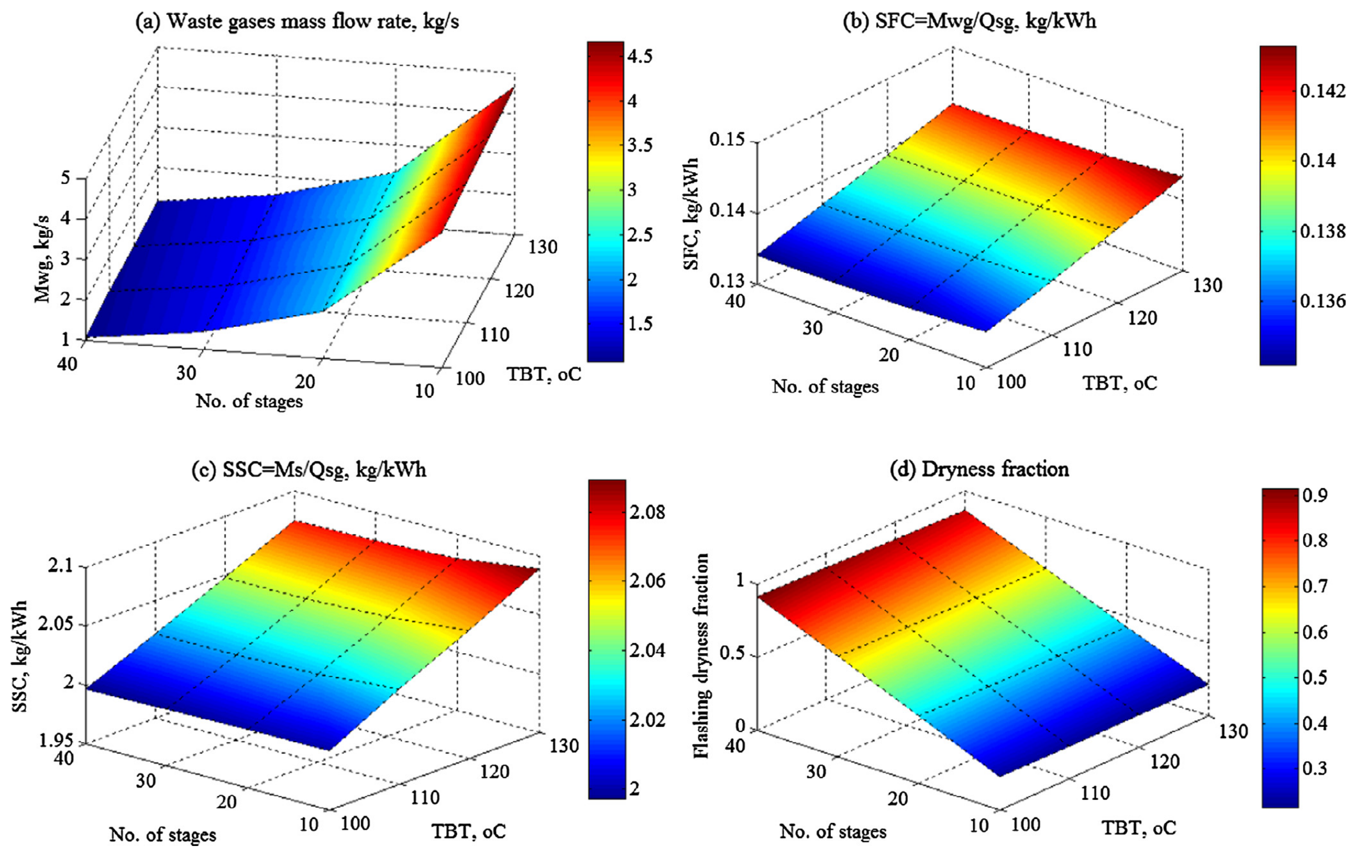


Fig. 8. The effect of No. of stages and TBT, °C on: (a) waste gases mass flow rate, kg/s, (b) specific fuel (gases) consumption, kg/h/kW, and (c) specific steam consumption, kg/h/kW, flashing dryness fraction.

important to increase the GR. Flashing dryness fraction is quite important parameter because it simply means the percentage of steam to power on the MED-PF. It is obvious in Fig. 9d. that by a slightly decrease in N_{eff} would increase the dryness fraction percentage production thence; remarkable results from the side of techno economic results. The flashing dryness fraction is ranged between 70% up to 95% at 22 and 16 MED effects respectively. Moreover; increasing the steam pressure up to 0.35 bar would increase the flashing dryness fraction. The optimized values for steam pressure and N_{eff} is fixed at 0.35 bar and 12 effects respectively.

Fig. 10a –d. represents the effect of operating conditions (steam pressure, TBT) on the same parameters the last been mentioned. It is noticed that the effect of TBT is remarkable on such parameters (M_{wg}, SFC, SSC, flashing dryness fraction). In Fig. 10a, the increasing of TBT would be followed by an increasing in the mentioned parameters. At the same time, any increase in steam pressure would decrease the values of the same parameters. Therefore; it is a matter of optimization in order to achieve a remarkable result. However; in Fig. 10b, the increasing of TBT caused a decreasing in flashing dryness fraction with opposite effect of steam pressure term. The effect of steam pressure would increase the value of the flashing fraction from 80% up to 95%. Fig. 11a and b. shows the effect of productivity of MSF and MED on the mass flow rate of waste gases and dryness fraction.

Clearly evident that the impact of the productivity of MSF-BR over M_{wg} significant largely because the thermal correlation between the brine heater and the steam generator. It is found that the M_{wg} increased from 0.8 kg/s up to 1.4 kg/s at 1e4 m^3/d and 1.8e4 m^3/d respectively. The MED-PF productivity hasn't any significant change on the M_{wg}. On the contrary; Fig. 11b. presents a remarkable effect from MED-PF productivity on the dryness fraction percentage. It is noticed that the dryness fraction percentage is increased from 80% up to 95% at 1e4 m^3/d and 1.5e4 m^3/d respectively. However; the increasing in the MSF-BR productivity would decrease the dryness fraction percentage.

From the previous figures the following notes should be considered:

- The productivity of the MSF-BR has a quite effect on the waste gases however; it may decrease the dryness fraction percentage and this considered not favorable.
- The effect of TBT on M_{wg}, SFC, and SSC is quite noticed because of the thermal connection between brine heater and the steam generator.
- Number of stages has also an effect on the *GR* and M_{wg}.
- For MED-PF, the effect was directly proportional to the flashing dryness fraction parameter. Increasing the productivity and steam pressure would increase the flashing dryness fraction percentage, thence; reducing the techno-economic consideration.

For the ORC, Fig. 12 shows the effect of evaporator hot side temperature ($dT = T_{exi} - T_{evp}$) and ORC turbine power on the M_{orc}, kg/s and thermal power, kW for heat rejection units. The massive effect is noticed for the ORC turbine load power. Increasing the turbine power would increase the M_{orc} and surly the thermal power loads on the heat rejection units. The effect of dT is slightly notable by increasing the dT a slight increase for both M_{orc} and thermal energy would happen. Decreasing the dT would increase the evaporator effectiveness, thence; little bit reducing the M_{orc}. Data results for the 1st and 2nd scenarios are illustrated in Tables 5 and 6.

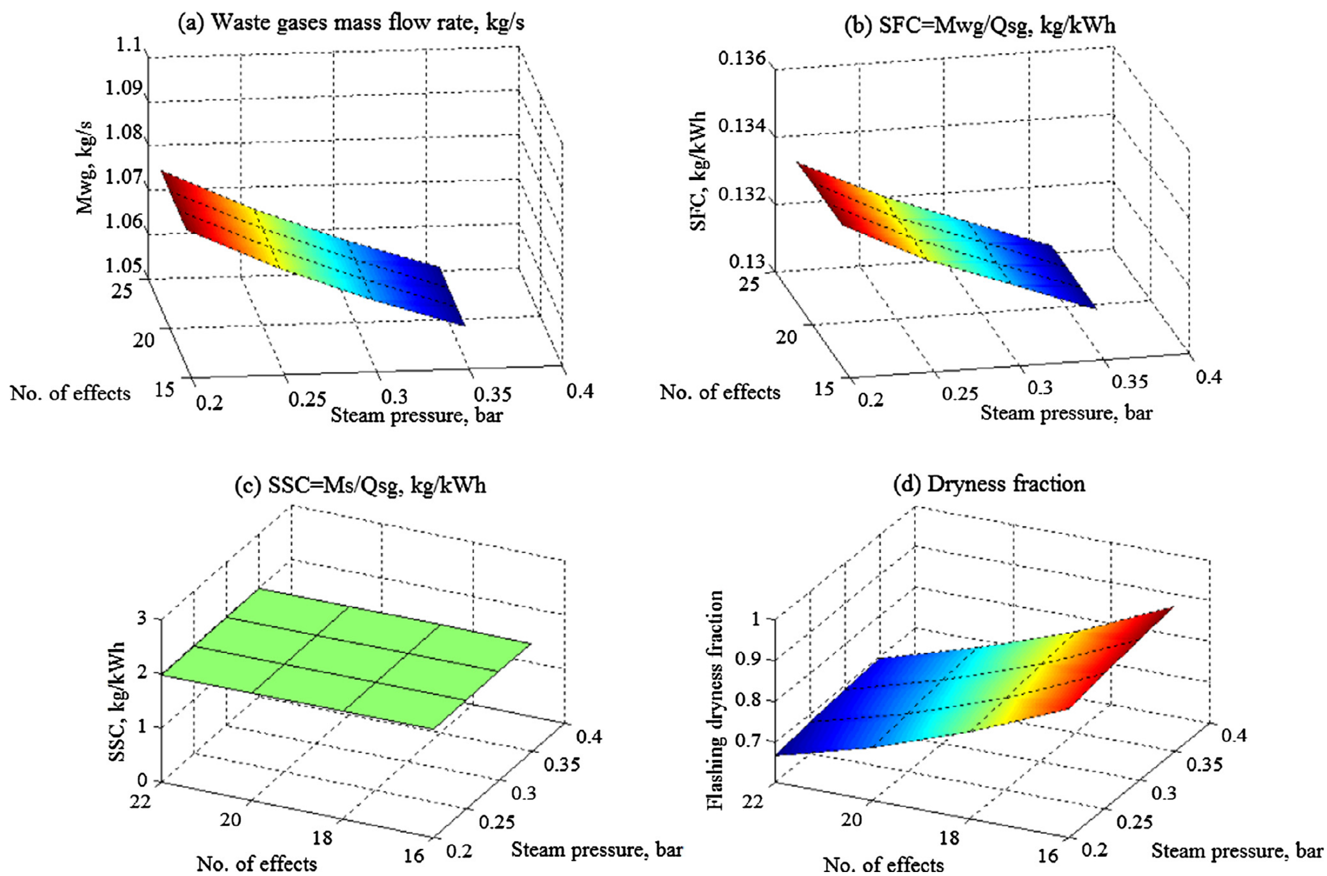


Fig. 9. Effect of Neff and steam pressure on: (a) waste gases flow rate, kg/s, (b) specific fuel (wastes) consumption, kg/h/kW, (c) specific steam consumption, kg/h/kW, and (d) dryness fraction.

5.2. Results of 3rd scenario

This scenario configuration is quite different. It's an example about electric power production-mainly-and freshwater production. As presented before, user can decide the productivity and the power demanded, thence; calculating the design aspects. The operating conditions of the MSF-BR would assign the results data of the L.P.T outlet stream. The energy balance through the BH would calculate the outlet L.P.T temperature. Thence; the calculation procedure would calculate the inlet L.P.T conditions which would calculate the H.P.T conditions (temperature, enthalpy, pressure, entropy, etc.). This scenario is producing much less freshwater (100 m^3/d) however; it can produce more than 60 MW of electric power (12 MW_{aux} + 48 MW_{grid}). Its low productivity tends to the design limits of quantities of waste gases from the refinery plant (shouldn't exceed over 5 m^3/d).

Moreover; increasing the productivity would increase the massively the mass flow rate across the cycle causing by this increasing the power of the gas compressor. Table 7 illustrates the data results of the 3rd scenario. It has seen from Table 7 that pressure ratio wasn't exceeded over 5, the top cycle temperature reached 1065.7 °C, cycle mass flow rate was 59 kg/s, and the outlet gases temperature (cycle bottom) was about 90 °C. The bottoming cycle gases are still high enough to power on MED-PF, however; it's out of enthalpy for the demanded steam. The cycle last line exhausts could be used for drying, water heating, building heating, and/or process heating. Fig. 13a shows the effect of M_{wg} on the scenarios water production.

It is shown that the 1st and 2nd scenarios produce much higher quantities of water production against the 3rd scenario. This is because the lower enthalpy of the mass flow rate across the BH unit in the 3rd one (3.86e4 kW_{th} for the 1st and 2nd vs. 215 kW_{th} for the 3rd). Fig. 13b shows the effect of increasing the M_{wg} on the power production. Increasing the M_{wg} rates would increase the power production rate. It is depending on the case of refinery plant for the wastes production flow rate, thence; more wastes equal more power. Fig. 13c shows the effect of power increasing on the pressure ratio (*Pr*) of the L.P.T. Increasing the power load on the L.P.T would decrease the pressure ratio. Therefore; the optimized value for the pressure ratio was at 5–8 to produce about 48 MW power. Fig. 13d shows the effect of production power on the CC heat addition rate, kW. Increasing the power load would increase the heat addition hence; the waste gases rate.

5.3. Exergy and cost results

In this part, exergy and cost analyses are performed in order to judge the most reliable scenario that should be applied. It has seen from the previous part that the 2nd scenario is in competitiveness result against the 3rd scenario based on energy analysis. For exergy analysis, there are two main parameters to judge the system performance. The first is exergy destruction, and the second is the exergy efficiency. Unlike energy, which is conserved in any process according to the first law of thermodynamics, exergy is destroyed due to irreversibility taking place in any process, which manifest itself in entropy creation or entropy increase. The availability equation for an open system in a uniform-state, uniform flow process can be developed with the first and second law of thermodynamics. The steady state general form of the exergy balance equation is defined as following [29], $0 = Ex_q + Ex_w + Ex_{fi} - Ex_{fo} - I_{ex}$. Where

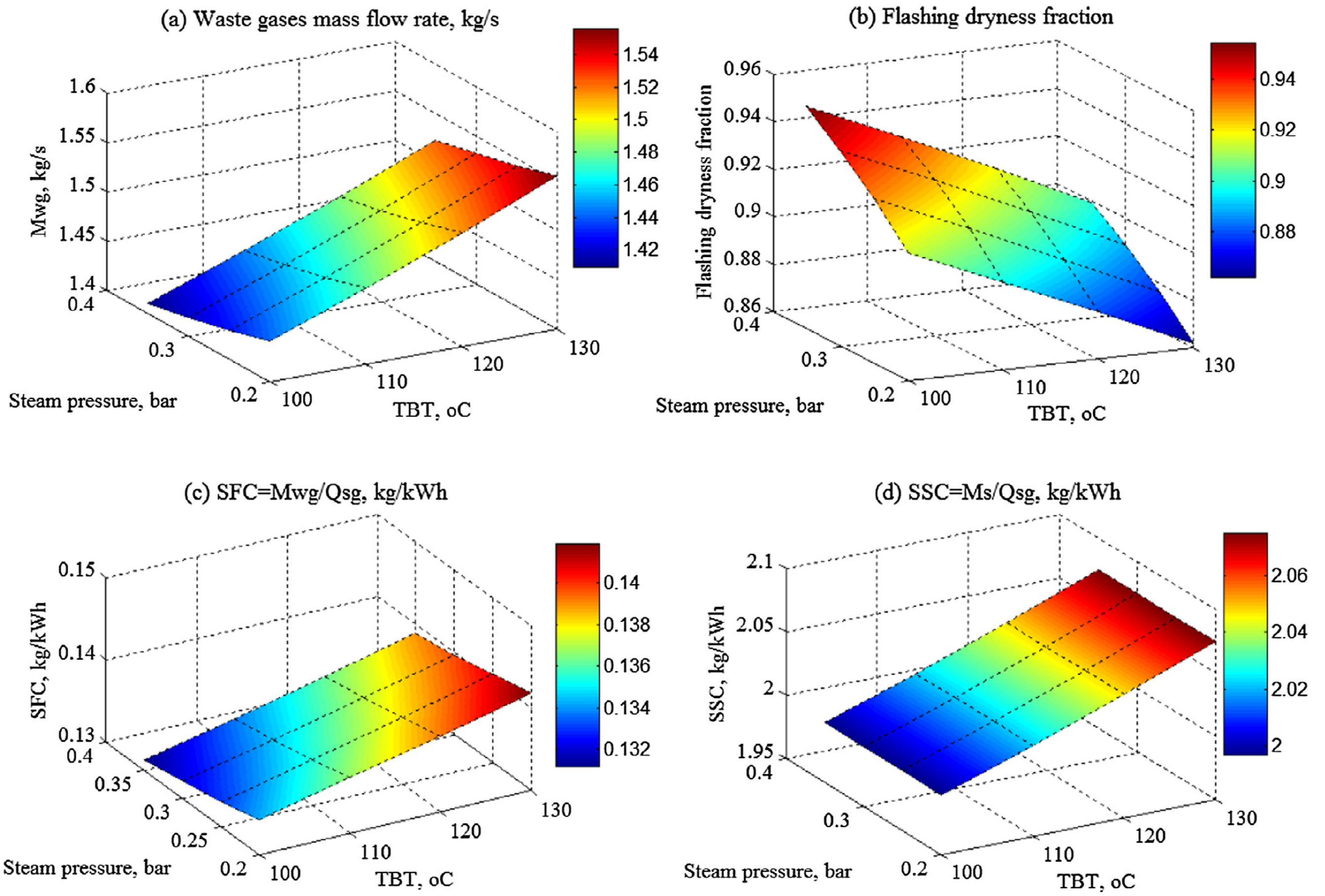


Fig. 10. Effect of MED steam pressure and MSF TBT on: (a) Waste gases flow rate, kg/s, (b) Flashing dryness fraction, (c) specific fuel (wastes) consumption, kg/h/kW, and (d) specific steam consumption, kg/h/kW.

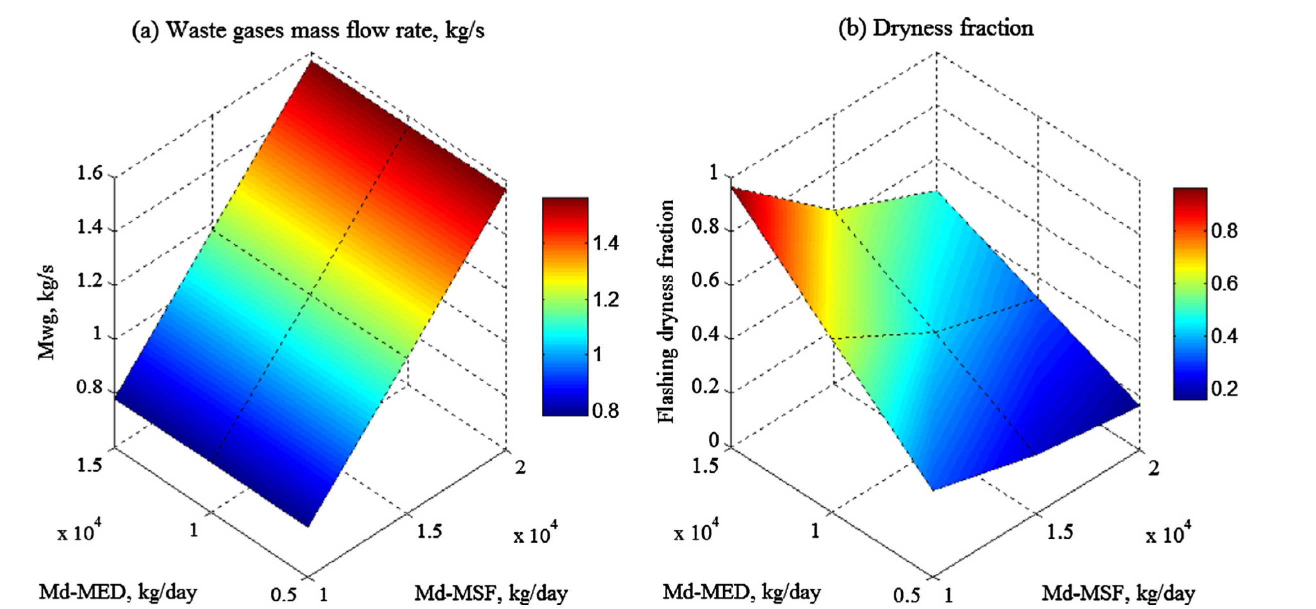


Fig. 11. Effect of productivity of MSF + MED on the waste gases flow rate and dryness fraction.

Ex_q is the exergy transfer due to the heat transfer between the control volume and its surroundings, and Ex_w is the value of the work produced by the control volume but in most cases the control volume has a constant volume, therefore $Ex_w = W$ can be further simplified. Ex_{fi} and Ex_{fo} is the flow exergy inlet and outlet from the control volume. I_{ex} is the total system exergy destruction rate, kW. Exergy efficiency can be measured as a relationship between ingoing and outgoing exergy flows or the ratio of net exergy output

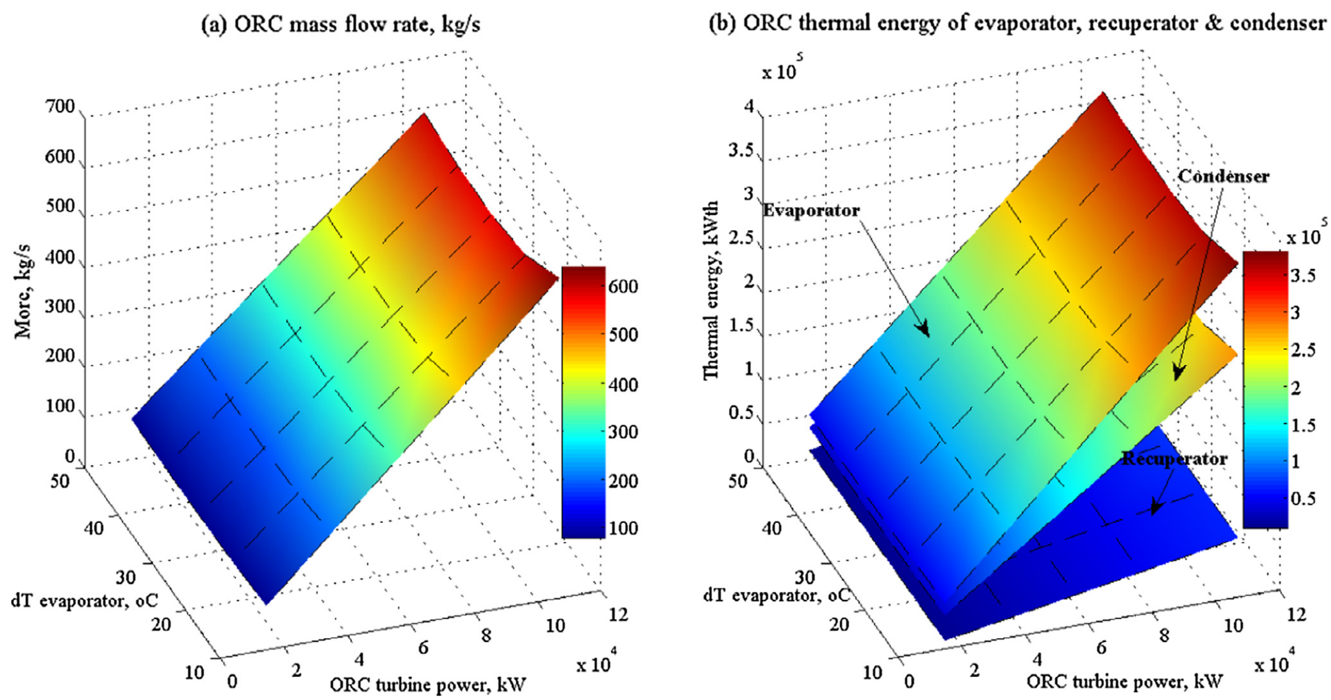


Fig. 12. (a) The variation of ORC flow rate, kg/s and (b) ORC thermal loads for evaporator, recuperator and condenser units.

Table 5
Data results for the 1st scenario (WGC-MSF-MED).

Waste gases chimney (WGC) unit	
Waste gases flow rate, kg/s	1.41
Waste gases flow velocity, m/s	12.45
Waste gases pressure drop, bar	1.163
Chimney tube cross sectional area, m^2	0.07548
Chimney tube diameter, m	0.31
Steam generator (SG) unit	
Air mass flow rate, kg/s	17.62
Waste gases thermal energy, kW_{th}	5.075e+4
Specific fuel consumption ($SFC = M_{wg}/Q_{sg}$), kg/kWh	0.1312
Specific steam consumption ($SSC = M_s/Q_{sg}$), kg/kWh	1.997
Outlet temperature, °C	333.3
Outlet enthalpy, kJ/kg	666.7
Brine heater (BH) unit	
BH tube diameter, m	0.03
Flow velocity, m/s	0.5
BH pressure, bar	1.056
Steam enthalpy inlet/outlet, kJ/kg	2677/423.9
Steam temperature inlet/outlet, °C	101.1/101.1
Total steam mass flow rate, kg/s	21.45
Seawater temperature inlet/outlet, °C	95.5/100
Brine recycle mass flow rate, kg/s	2039
BH thermal energy, kW_{th}	3.867e+4
BH area, m^2	6912
Multi stage flash brine recycle (MSF-BR) unit	
Distillate flow rate, kg/s	208 = 18,000 m^3/day
Total inlet feed flow rate, kg/s	1210
Make up feed flow rate, kg/s	583.3
Feed loss flow rate, kg/s	626.9
Brine loss flow rate, kg/s	375
Recycle feed flow rate, kg/s	2039
Total brine blow down, kg/s	1831
Feed salinity ratio, ppm	45,000
Brine salinity ratio, ppm	70,000
1st splitter ratio	0.482
2nd splitter ratio	0.7952
Top brine temperature (TBT), °C	100
Top feed temperature, °C	95.5
Seawater temperature, °C	27
Brine blow down temperature, °C	40

(continued on next page)

Table 5 (*continued*)

Stage area of heat recovery section, m^2	2731
Stage area of heat rejection section, m^2	1429
Pumps efficiency	75%
MSF-BR total pumping power, kW	3228
No. of stages = heat recovery/heat rejection	40 = 37/3
Flash cyclone separation tank	
Operating steam pressure, bar	0.35
Inlet flash temperature, °C	101.1
Inlet flash enthalpy, kJ/kg	423.9
Steam temperature, °C	72.66
Flashing enthalpy, kJ/kg	2524
Dryness fraction	0.954
Liquid flow rate @ tank bottom, kg/s	0.954
Steam mass flow rate, kg/s	20.47
Tank volume, m^3	62
Inlet/outlet tube cross sectional area, m^2	0.7321/0.7321
Steam tube diameter, m	0.9
Flash tank height/width, m	7/3
Multi effect distillation parallel feed (MED-PF)	
MED-PF distillate productivity, kg/s	231.5 = 20,000 m^3/day
Steam flow rate, kg/s	20.47
Feed flow rate/effect, kg/s	54.01
Seawater salinity/brine blow down salinity, ppm	45,000/70,000
Top evaporation pressure, kPa	30.77
Total feed/cooling water, kg/s	1117/469.2
Top steam temperature, °C	72.66
Top brine temperature, °C	70.65
Top vapor/distillate temperature, °C	69.88/69.71
Seawater temperature, °C	27
End condenser outlet feed temperature, °C	36.78
End condenser outlet distillate temperature, °C	29.45
End condenser effectiveness	80%
Total pumping power, kW	1126
Pumps efficiency	75%
No. of effects	12
Total heat transfer area, m^2	121,082
End condenser heat transfer area, m^2	4050
Effect area, m^2	1.015e4
Pump unit	
Cycle pump power, kW	2.07
Inlet/outlet pump temperature, °C	72.66/73
Inlet/outlet pump enthalpy, kJ/kg	304.2/304.3

to the actual exergy input for any given system when achieving the required task, $\eta_{ex} = \frac{Ex_o}{Ex_i}$. Regarding to the proposed scenarios, the following relations show the exergetic inlet and outlet streams which are considered in this study. To measure the exergy destruction for the 1st scenario, there are four inputs and two outputs which are the following:

$$\dot{I}_{ex} = (Ex_{wg} + Ex_{wp} + Ex_{fi\_msf} + Ex_{fi\_med})_i - (Ex_{d\_msf} + Ex_{d\_med})_o$$

And the exergy efficiency is obtained as, $\eta_{ex} = \frac{(Ex_{d\_msf}+Ex_{d\_med})_o}{(Ex_{wg}+Ex_{wp}+Ex_{fi\_msf}+Ex_{fi\_med})_i}$ Where; Ex_{d_msf} and Ex_{d_med} are the exergy gain from the plant (distillate flow exergy). Ex_{fi_msf}, Ex_{fi_med}, Ex_{wg} and Ex_{wp} are the inlet exergy of feed streams (MSF and MED), waste gases exergy and power exergy for pumps respectively. The exergetic power for the pumping system in the plant are considered a power cost loaded by the main grid. Outlet brine flow rate exergy (MSF and MED), thermal leakages and exhaust exergy are neglected for all scenarios because they are a loss from the system.

According to the 2nd scenario, the power from the ORC cycle would serve the plant pumping system. Therefore; the total exergy destruction rate is obtained as;

$$\dot{I}_{ex} = (Ex_{wg} + Ex_{wp} + Ex_{fi\_msf} + Ex_{fi\_med})_i - (Ex_{w\_ORC} + Ex_{d\_msf} + Ex_{d\_med})_o \text{ and } Ex_{w\_ORC} = Ex_{wp}.$$

The 2nd scenario exergetic efficiency is then obtained as $\eta_{ex} = \frac{(Ex_{d\_msf}+Ex_{d\_med})_o}{(Ex_{wg}+Ex_{fi\_msf}+Ex_{fi\_med})_i}$.

According to the 3rd scenario, the power from the H.P.T would power on the gas compressor and the L.P.T would deliver the main power for pumping system and for the main grid as a main exergy gain beside the distillate productivity. Therefore; the total exergy destruction rate would become;

$$\dot{I}_{ex} = (Ex_{wg} + Ex_{wp} + Ex_{w\_Comp} + Ex_{fi\_Comp} + Ex_{fi\_msf})_i - (Ex_{w\_H.P.T} + Ex_{w\_L.P.T} + Ex_{d\_msf})_o$$

where $Ex_{w\_H.P.T} = Ex_{w\_Comp}$. The 3rd scenario exergetic efficiency is then obtained as;

$$\eta_{ex} = \frac{(Ex_{d\_msf} + Ex_{w\_L.P.T})_o}{(Ex_{wg} + Ex_{fi\_msf} + Ex_{wp} + Ex_{fi\_Comp})_i}.$$

Figs. 14–16 show the data obtained for the three scenarios related to the exergy efficiency and the exergy destruction rate. The results show that the 3rd scenario gives remarkable results compared against the remaining scenarios by achieving 62.73% of exergy efficiency. The 2nd scenario is considered the same as the 1st with a slightly increase in exergy efficiency due to powering the pumping system based on the generated power from the ORC. The same behavior is noticed while comparing based on the exergy destruction rate which the 3rd scenario gives the minimum rate of exergy destruction (2.85e4 kW vs 1.84e6 kW for the 1st and the 2nd). Table 8 shows the inlet and outlet exergy streams related to each scenario. Exergy balances for the proposed scenarios are shown in detail in Appendix C.

Table 6
Data results of the 2nd plant scenario (WGC-MSF-MED-ORC).

Heat exchanger evaporator (HEX) unit	
M_{ORC} mass flow rate, kg/s	24.17
Exhausts flow rate, kg/s	1.41
Inlet/outlet exhausts temperature, °C	333/141.3
Inlet/outlet vapor temperature, °C	101.3/293.3
Inlet/outlet enthalpy, kJ/kg	244.5/859.3
HEX thermal power, kW_{th}	1.486e4
Outlet high pressure to the turbine, bar	30.1
HEX area, m^2	40
No. of tubes	1550
ORC turbine unit	
M_{ORC} mass flow rate, kg/s	24.17
Power, kW (all plant pumping power)	4470.64
Inlet/outlet enthalpy, kJ/kg	859.3/674.3
Inlet/outlet temperature, °C	293.3/138.7
Recuperator unit	
M_{ORC} mass flow rate, kg/s	24.17
Inlet/outlet temperature-steam side, °C	138.7/57.85
Inlet/outlet enthalpy-steam side, kJ/kg	674.3/556.2
Inlet/outlet temperature-liquid side, °C	37.65/101.3
Inlet/outlet enthalpy-liquid side, kJ/kg	126.3/244.5
Thermal power, kW_{th}	2855
Heat transfer area, m^2	30
Condenser unit	
M_{ORC} mass flow rate, kg/s	24.17
M_{cw} flow rate, kg/s	101.8
Inlet/outlet temperature-steam side, °C	57.85/35
Inlet/outlet enthalpy-steam side, kJ/kg	556.2/121.6
Inlet/outlet temperature-cooling side, °C	27/51.68
Condensation pressure, bar	0.06215
Thermal power, kW_{th}	1.05e4
Heat transfer area, m^2	537
Pump unit	
Cycle pump power, kW	113.4
Inlet/outlet pump temperature, °C	35/37.65
Inlet/outlet pump enthalpy, kJ/kg	121.6/126.3

It is clear that there is a relation between the desalination plant productivity and the inlet exergy to the system. Therefore; the 3rd scenario gives lower exergy destruction rate compared against the other scenarios. The 3rd scenario proves that increasing the power generation is much better than producing large amount of fresh water. Among all units/scenario, steam generator, MSF, MED and combustion chamber recorded a massive rates of exergy destruction (see Fig. 16). However; the 3rd scenario is in the lowest rates according to the low thermal loads on the MSF plant connected with the low rate productivity. Moreover; the high-power production is considered as a useful gain to the system. Fig. 17 shows the data comparison related to the unit hourly costs parameter (UHC, $/h-total operating and maintenance costs). Cost analyses are shown in details in Appendix D. Fig. 17 shows that 2nd and 3rd scenarios are most favorable compared against the 1st scenario and the conventional operation. It is related to the designer and/or decision makers in Suez Company for Oil Refinery [27] to construct the most reliable scenario (2nd or 3rd), however; the 2nd scenario is most favorable because of a massive production of freshwater. The 3rd scenario is favorable while using much greater amounts of refinery waste gases more than 5 m^3/d. Therefore; and due the limitations of quantity, the 2nd scenario is much remarkable based on the massive productivity while the 3rd scenario is remarkable based on the massive power generation with high rates of exergy efficiency as an advantage. The massive cost results are recorded by the MSF and MED. It was recorded lower in the 3rd scenario according to lower water production and the operation of MSF without MED. The 2nd scenario gives minimum results for the steam generator because the ORC generated the required

Table 7
Data results of the 3rd plant scenario (WGC-MSF-GTC).

Waste gases chimney (WGC) unit	
Waste gases flow rate, kg/s	1.412 (5 m^3/d)
Waste gases flow velocity, m/s	12.47
Waste gases pressure drop, bar	1.27
Chimney tube cross sectional area, m^2	0.07548
Chimney tube diameter, m	0.31
Gas compressor unit	
Pressure ratio	5–10
Inlet/outlet temperature, K	298/513.5
Adiabatic index	1.4
Power, kW	1.278e4
Mass flow rate, kg/s	59.06
Combustion chamber (CC) unit	
Air/waste gases ratio	12.5
Mass flow rate, kg/s	59.06
Air flow rate, kg/s	17.65
Thermal power heat addition, kW	5.083e4
Inlet/outlet temperature, K	513.5/1346.87
H.P.T unit	
Total demanded power for the electric grid, kW	Compressor power
Mass flow rate, kg/s	59.06
Inlet/outlet temperature, K	1346.87/1161.76
Inlet/outlet pressure, bar	5/2.36
Adiabatic index	1.321
L.P.T unit	
Total demanded power for the electric grid, kW	47,000–50,000
Mass flow rate, kg/s	59.06
Inlet/outlet temperature, K	1161.76/374.13
Inlet/outlet pressure, bar	2.36/1
Adiabatic index	1.372
Brine heater (BH) unit	
BH tube diameter, m	0.03
Flow velocity, m/s	0.5
Gases temperature inlet/outlet, K	374.13/369.63
Total mass flow rate, kg/s	59.06
Seawater temperature inlet/outlet, °C	95.5/100
Brine recycle mass flow rate, kg/s	11.33
BH thermal energy, kW_{th}	214.8
BH area, m^2	40
Multi stage flash brine recycle (MSF-BR) unit	
Distillate flow rate, m^3/d	100
Total inlet feed flow rate, kg/s	6.724
Make up feed flow rate, kg/s	3.241
Feed loss flow rate, kg/s	3.483
Brine loss flow rate, kg/s	2.083
Recycle feed flow rate, kg/s	11.33
Total brine blow down, kg/s	10.17
1st splitter ratio	0.482
2nd splitter ratio	0.7952
Top brine temperature (TBT), °C	100
Top feed temperature, °C	95.5
Seawater temperature, °C	27
Brine blow down temperature, °C	40
MSF-BR total pumping power, kW	18
No. of stages = heat recovery/heat rejection	40 = 37/3

power for the pumping system. Fig. 18 shows the results for the three scenarios based on the total plant cost, $/y. It is anticipated by Fig. 18. That the 3rd scenario gives minimum results while comparing against the 1st and the 2nd scenarios.

6. Conclusion

In the present study, three novel scenarios were introduced to resolve the oil refinery waste gas problem. The work is given a new three ideas in order to utilize the waste gases burning for desalination and electric power generation rather than burning it as a waste into air. Due to some decision difficulties, three models were simulated and introduced in this study. The 1st scenario is to

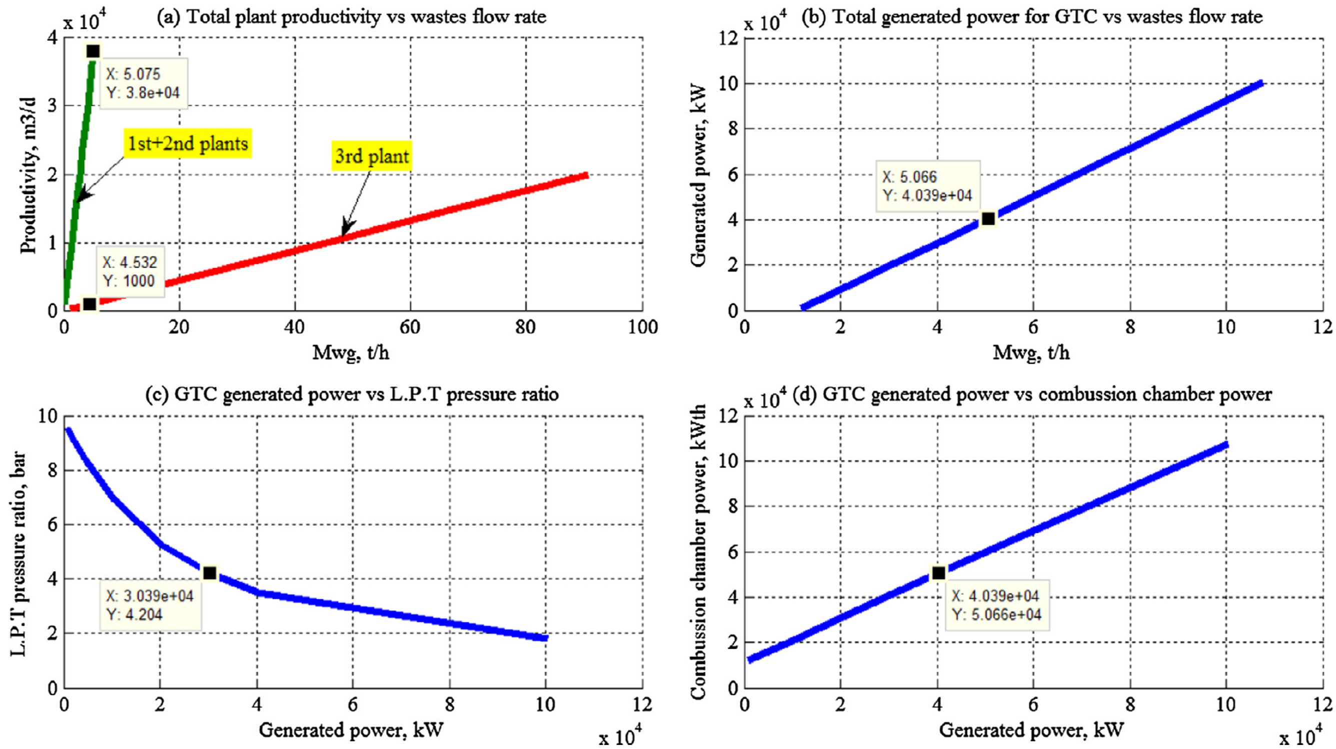


Fig. 13. (a) Effect of waste gases on the productivity, m^3/d, (b) Effect of waste gases on the power generation, kW, (c) Effect of generated power on the L.P.T pressure ratio, and (d) Effect of generated power on the combustion chamber thermal power, kW.

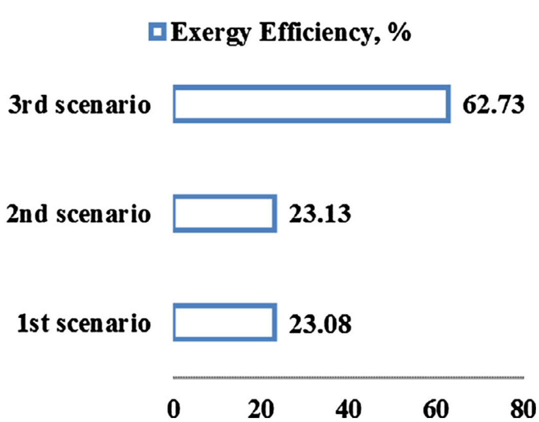


Fig. 14. The exergetic efficiency for the three scenarios.

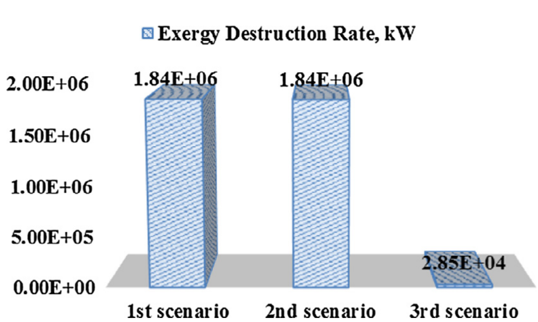


Fig. 15. The exergy destruction rate, kW for the three scenarios.

use the waste gases for MSF-BR and MED-PF desalination plants via brine heater and flash tank separation units. The 2nd scenario is considered the same as the 1st however; ORC is added in order to generate the sufficient power for the whole cycle pumping system and to the main grid. Toluene is recommended and used for the ORC operation. The exerted exhausts from the steam generator is considered the primary source of power for the ORC via the evaporator heat exchanger unit. The 3rd scenario is quite different because it depends on the gas turbine cycle to generate power and heat for the MSF part. The gas turbine cycle exhausts would power on the optimized MSF-BR plant via the brine heater unit. Due to some design limits, such as compressor load and operating temperatures, the 3rd scenario would desalinate an amount of 100 m^3/d related to 5 m^3/h of waste gases. The 3rd scenario could desalinate more amounts of freshwater in case of much greater amounts of waste gases are needed (more than 5 m^3/h). Nevertheless; the 3rd scenario generated much larger power (60 MW) with unit hourly costs (UHC, \$/h) almost zero. The UHC, \$/h of the 1st scenario was about the range of 435 \$/h (nearly the same as the conventional desalination plants) without any cost advantage compared against the 2nd and the 3rd. It is referred to the designer and/or decision makers to confirm the most reliable scenario (2nd or 3rd), however; the 2nd scenario is more favorable because of a massive production of freshwater (38,000 m^3/d). The 3rd scenario is also favorable while using much larger amounts of refinery waste gases more than 5 m^3/d. The 3rd scenario is more applicable for desert oil plants that far away from the sea. According to the exergy analysis, the 3rd configuration gives minimum exergy destruction rate against the remaining scenarios with much high exergetic efficiency (62.73% vs 23% for the 2nd and vs 23% for the 1st). There are a lot of benefits should be harvested from the scenarios proposal. Location landscape, seaside, environmental impact, water supply, etc. It is quite favorable for oil refinery plants

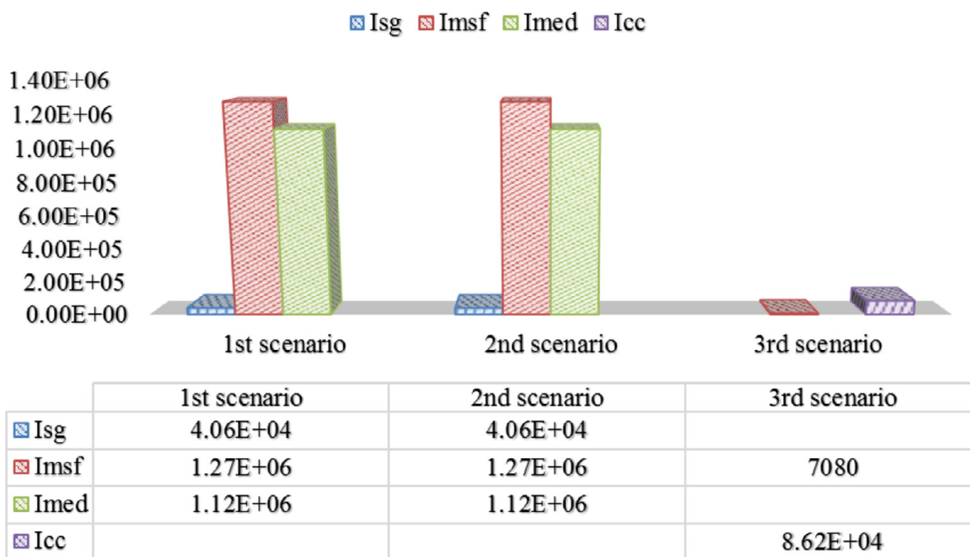


Fig. 16. Exergy destruction rate, kW for the proposed scenarios based on steam generator, MSF, MED, and combustion chamber units.

Table 8
The exergetic data streams for the proposed scenarios.

Stream	1st scenario	2nd scenario	3rd scenario
$Ex_{q\ loss}$, kW	---	---	---
$Ex_{b\ msf}$, kW	---	---	---
$Ex_{b\ med}$, kW	---	---	---
$Ex_{fi\ msf}$, kW	1.522e6	1.522e6	8454
$Ex_{fi\ med}$, kW	8.147e5	8.147e5	---
$Ex_{d\ msf}$, kW	2.615e5	2.615e5	1453
$Ex_{d\ med}$, kW	2.906e5	2.906e5	---
Ex_{wp}, kW	4354	4469.61	18
Ex_{wg}, kW	5.075e4	5.075e4	5.075e4
$Ex_{w\ ORC}$, kW	---	4469.61	---
$Ex_{w\ Comp}$, kW	---	---	1.278e4
$Ex_{fi\ Comp}$, kW	---	---	1.767e4
$Ex_{w\ H.P.T}$, kW	---	---	1.278e4
$Ex_{w\ L.P.T}$, kW	---	---	4.65e4
Total Ex_i, kW	2.392e6	2.387e6	7.647e4
Total Ex_o, kW	5.521e5	5.521e5	4.797e4
Total exergy efficiency, %	23.08	23.13	62.73
Total I_{ex}, kW	1.84e6	1.835e6	2.85e4
Productivity, m^3/day	38000	38000	100
Net power generation, kW	---	---	47000

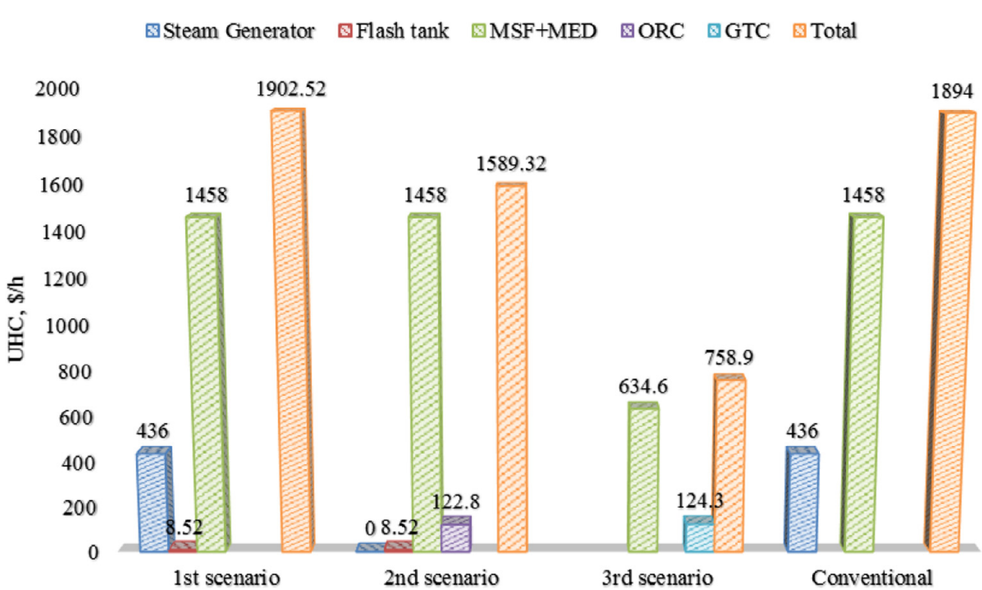


Fig. 17. Units hourly costs and total for all scenarios units.

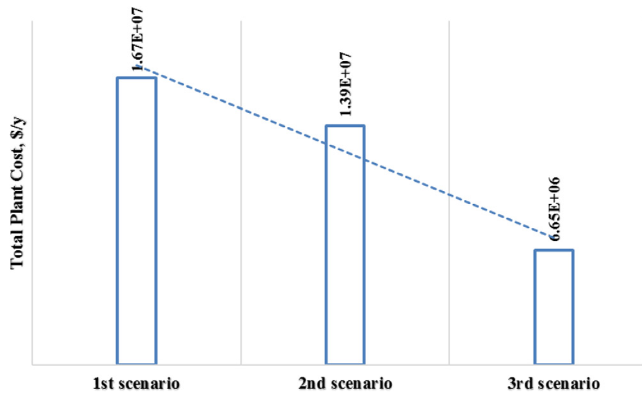


Fig. 18. Data results for the proposed scenarios based on total plant cost, $/y.

that are located beside the sea shore. Therefore; desalination matter is considered a vital role in order to payback some benefits to the environment against the hazards that been emitted from the refinery plant itself. The three scenarios are applicable for all types of the oil refinery plants with greater advantage to the second and third scenarios according to the exergy and cost analyses. Being far away from sea means that the 3rd scenario is favorable without desalination part (only power generation). The following benefits should be considered:

- Sulfur extraction should be considered. The extracted sulfur could be used in many industries.
- Power generation is the main gain for such plants in order to recover some benefits related to environmental issues.
- Desalination is the second main gain in order to overcome the water shortage problem based on the location situation. The desalinated water is also essential for the power plant operation.
- The proposed scenarios would generate electricity and fresh water instead of burning waste gases without any gain or benefits add to this the hazards to the environment.
- Water and electricity would be useful for the establishment operations of new hostels and tourist districts.

Appendix A. Toluene thermos-physical properties

A.1. Density for liquid and vapor phases kg/m³

$$\rho_{tl} = -7.981^{-19} \times T^9 + 7.002^{-16} \times T^8 - 2.087^{-13} \times T^7 + 1.821^{-11} \times T^6 + 1.971^{-9} \times T^5 - 3.474^{-7} \ldots \times T^4 - 3.29^{-6} \times T^3 + 0.001316 \times T^2 - 0.9326 \times T + 884.5$$

$$\rho_{tv} = 7.873^{15} \times \exp^{-((Tco-868.2)/97.11)^2)} + 1898 \times \exp^{-((Tco-666.7)/219.2)^2)}$$

A.2. Dynamic viscosity for liquid and vapor phases kg/m³

Toluene physical properties for liquid and vapor phases are obtained from the following correlations;

$$\mu_{tl} = 10^{-6} \times (3.262729 \times 10^{-5} \times T^3 + 5.14015 \times 10^{-2} \times T^2 - 27.89675 \times T + 5.305598 \times 10^3)$$

$$\mu_{tv} = 10^{-6} \times (6.338982 \times 10^{-8} \times T^4 - 1.602562 \times 10^{-4} \times T^3 + 1.519286 \times 10^{-1} \times T^2 \ldots - 63.99838 \times T + 1.011961 \times 10^4)$$

A.3. Specific enthalpy of dry saturated vapor kJ/kg

$$h_v = 2.323\mathrm{e} - 019 \times \mathrm{T}^9 + 2.638\mathrm{e} - 16 \times \mathrm{T}^8 - 7.835\mathrm{e} - 14 \times \mathrm{T}^7 + 6.784\mathrm{e} - 12 \times \mathrm{T}^6 + 7.627\mathrm{e} - 10 \times \mathrm{T}^5 - 1.392\mathrm{e} - 7 \times \mathrm{T}^4 - 1.443\mathrm{e} - 6 \times \mathrm{T}^3 + 0.002331 \times \mathrm{T}^2 + 1.019 \times \mathrm{T} + 490.4$$

A.4. Specific enthalpy of saturated liquid kJ/kg

$$h_l = -3.023\mathrm{e} - 19 \times \mathrm{T}^9 - 2.041\mathrm{e} - 16 \times \mathrm{T}^8 + 6.098\mathrm{e} - 14 \times \mathrm{T}^7 - 5.372\mathrm{e} - 12 \times \mathrm{T}^6 - 5.526\mathrm{e} - 10 \times \mathrm{T}^5 + 9.276\mathrm{e} - 8 \times \mathrm{T}^4 + 2.962\mathrm{e} - 6 \times \mathrm{T}^3 + 0.001018 \times \mathrm{T}^2 + 1.628 \times \mathrm{T} + 63.19$$

A.5. Specific entropy of saturated vapor kJ/kg°C

$$s_v = -6.571\mathrm{e} - 16 \times \mathrm{T}^6 - 7.761\mathrm{e} - 14 \times \mathrm{T}^5 + 2.712\mathrm{e} - 10 \times \mathrm{T}^4 - 1.128\mathrm{e} - 7 \times \mathrm{T}^3 + 2.61\mathrm{e} - 5 \times \mathrm{T}^2 - 0.001973 \times \mathrm{T} + 1.813$$

A.6. Specific entropy of saturated liquid kJ/kg°C

$$s_l = 1.038 \times \exp^{(0.002218 \times T)} - 0.7889 \times \exp^{(-0.004717 \times T)}$$

A.7. Saturation pressure bar

$$P_{sat} = 7.025e - 22 \times T^9 - 4.53e - 19 \times T^8 + 1.187e - 16 \times T^7 - 2.775e - 14 \times T^6 + 6.104e - 12 \times T^5 + 2.474e - 9 \times T^4 + 2.434e - 7 \times T^3 + 1.429e - 5 \times T^2 + 0.0005795 \times T + 0.009935$$

Appendix B. The proposed scenarios

B.1. The 1st & 2nd scenarios: WGC-MSF-MED & WGC-MSF-MED-ORC

The system includes waste gas chimney (WGC), steam generator (SG), brine heater (BH), multi stage flash brine recycle (MSF-BR), flash tank (FSH), multi effect distillation parallel feed (MED-PF), and pumping system. User should assign the total plant productivity (m^3/d) and some operating conditions such as ambient temperature, seawater temperature, and salinity. The 2nd scenario is considered the same in configuration as the 1st except adding the organic Rankine cycle units which are heat exchanger evaporator (H_{evp}), turbine, recuperator, condenser and pump. Figs. B.1 and 2 show the model browser-SimuLink environment- that been designed by Sharaf [19–21,23].

B.2. The 3rd scenario: WGC-MSF-GTC

The system includes waste gas chimney (WGC), gas compressor (Comp), high pressure turbine (H.P.T), low pressure turbine (L.P.T), brine heater (BH) and multi stage flash brine recycle (MSF-BR). User should assign the total plant productivity (m^3/d) and some operating conditions such as ambient temperature, seawater temperature, and salinity. Fig. B.3 shows the model browser-SimuLink environment- that been designed by Sharaf [19–21,23].

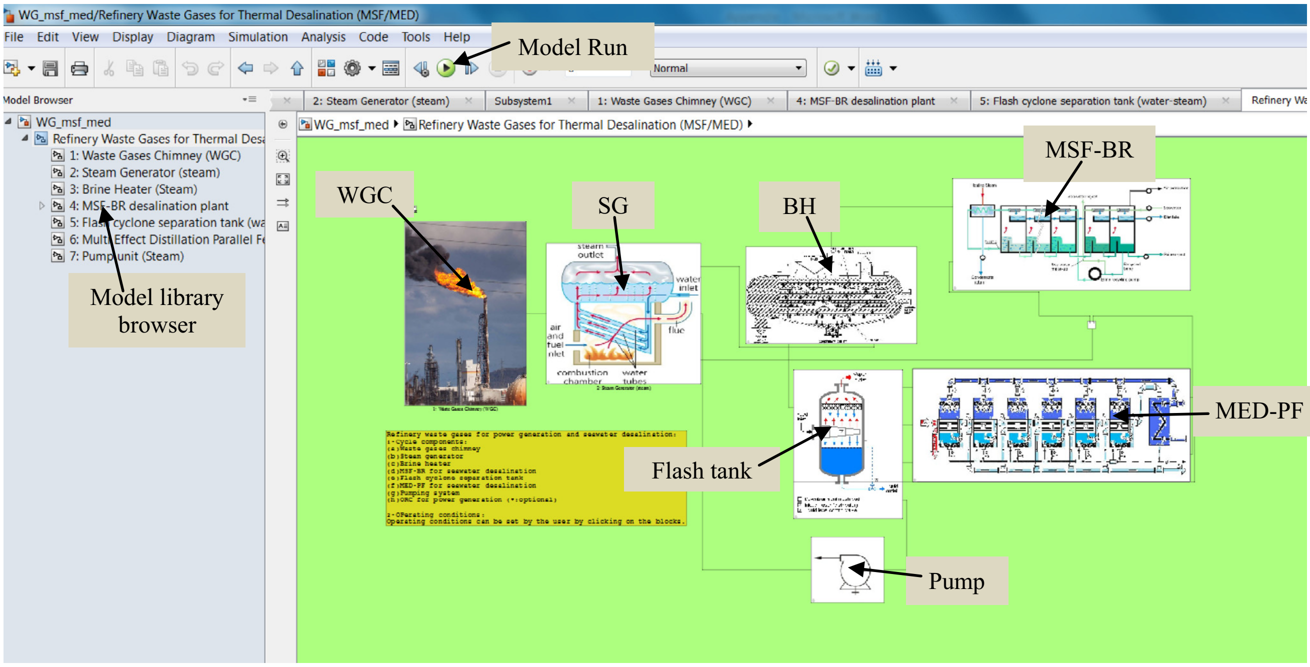


Fig. B1. Model browser of the 1st scenario by the aid of REDS-SimuLink toolbox [24–26,28].

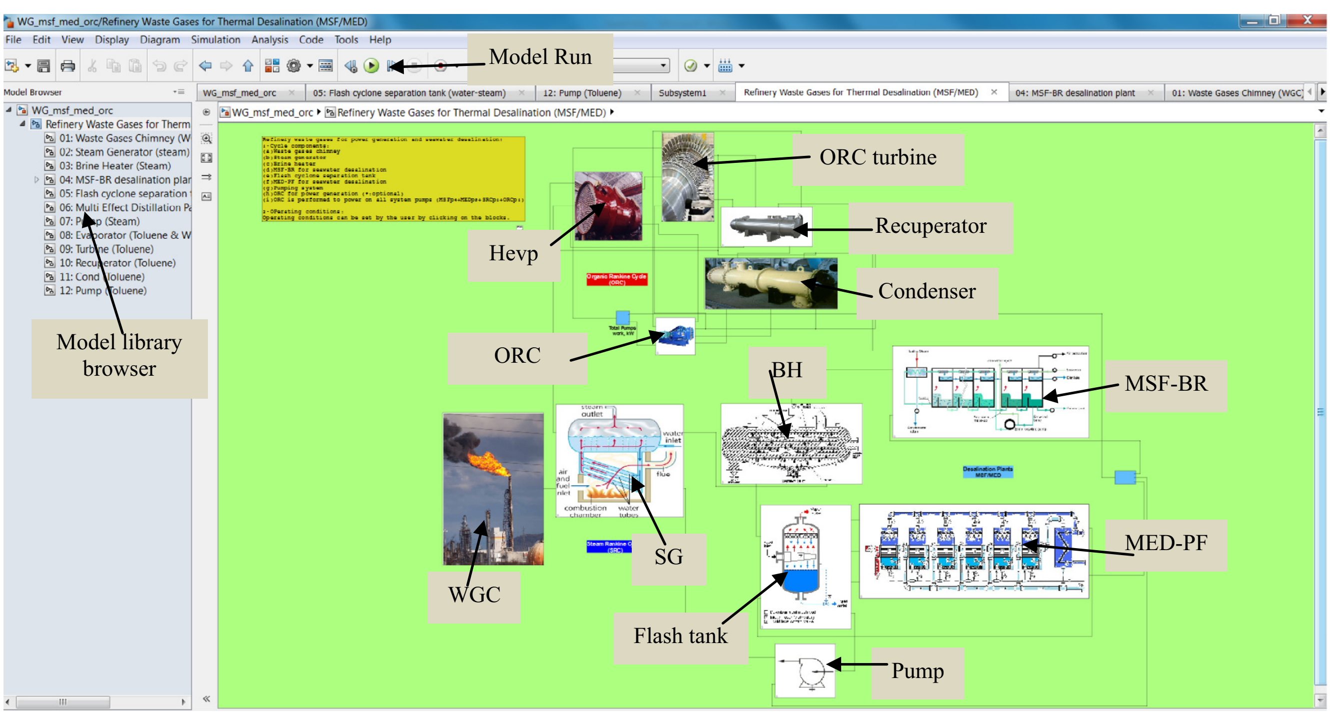


Fig. B2. Model browser of the 2nd scenario by the aid of REDS-SimuLink toolbox [24–26,28].

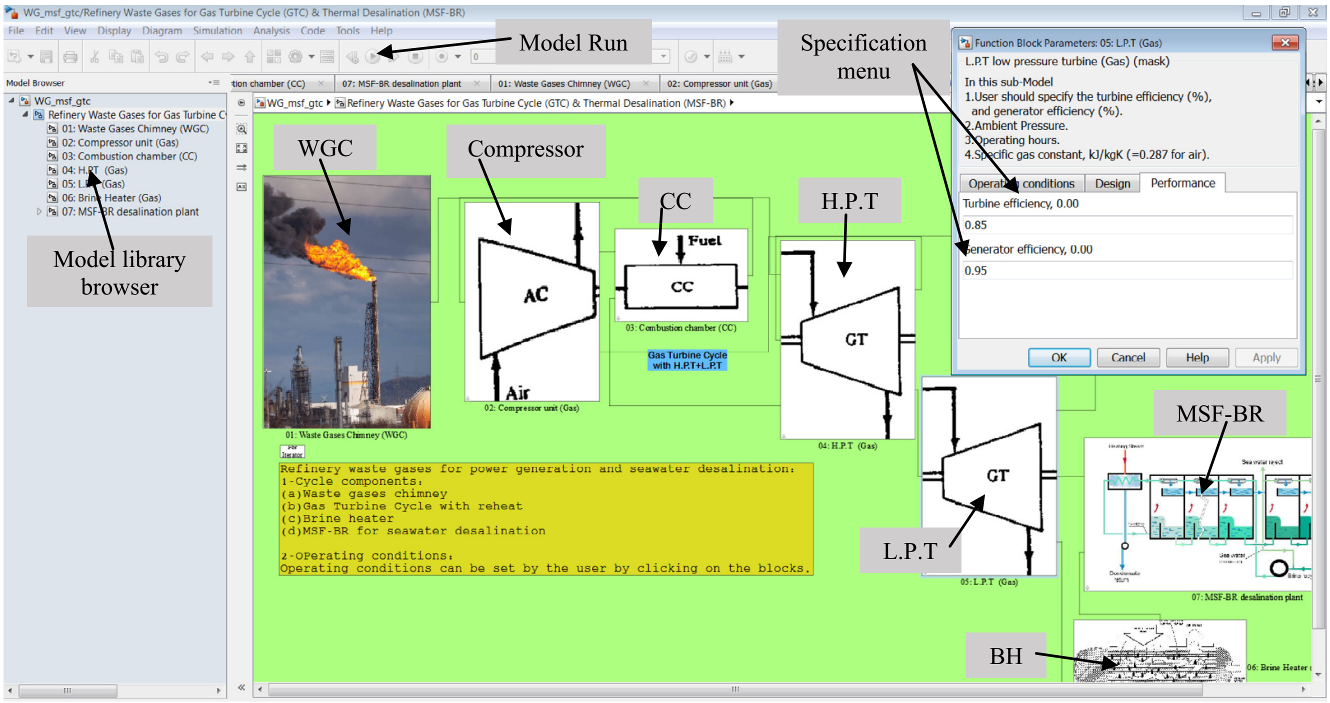


Fig. B3. Model browser of the 3rd scenario by the aid of REDS-SimuLink toolbox [24–26,28].

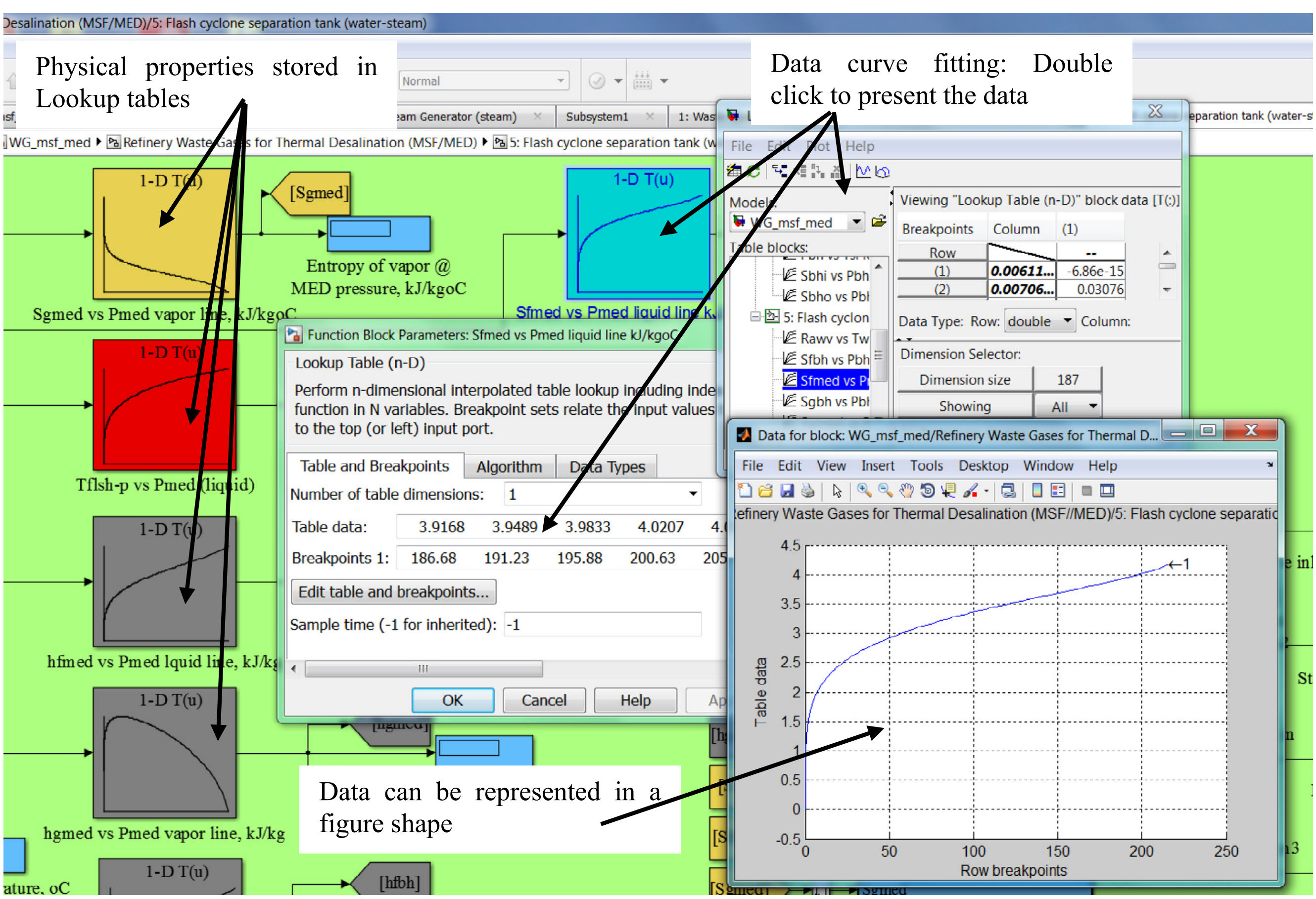


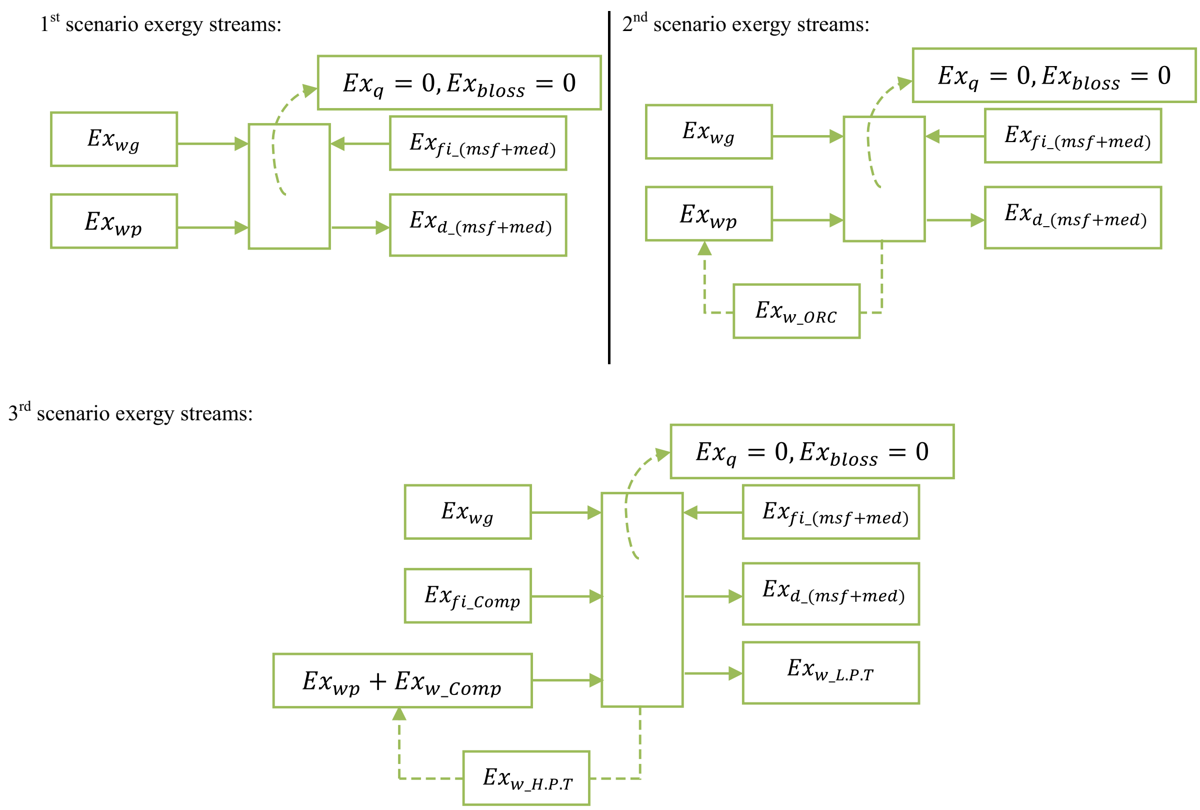
Fig. B4. Lookup Tables for physical properties data: entropy, enthalpy, pressure, temperature, specific volume.

Appendix C

The exergy balances for all proposed scenarios are shown in this part. The following schematics show the exergy streams that been used to calculate the exergy destruction rate and the exergetic efficiency.

$$\dot{Ex}_{ph} = \dot{m}\left(C_p(T,S)\times(T-T_o)\times C_p(T,S)\log\frac{T}{T_o}\right), \quad (T_o = \text{reference temperature})$$

For chemical part; the exergy stream (kg/s) should be calculated according to the following relation:



where Ex_f represents the chemical and physical exergy of seawater feed stream to the MSF and MED effects, Ex_b is the exergy stream associated with brine and neglected as loss stream, while Ex_d is the chemical and physical exergy stream of distillate product. Exergy of saline streams is obtained based on physical and chemical components. In physical part; the exergy streams to feed, brine, and distillate are functions of h_f, h_b, and h_d which are calculated based on seawater specific heat capacity C_p, salinity s, and feed seawater temperature for each stream [22] where;

$$h_{f,d,b} = h_o + (A\times T + B/2\times T^2 + C/3\times T^3 + D/4\times T^4)$$

where $h_o = 9.6296\times s - 0.4312402\times s^2$
And;

$$A = 4206.8 - 6.6197\times S + 1.2288\times 10^{-2}\times S^2$$
$$B = -1.1262 + 5.4178\times 10^{-2}\times S - 2.2719\times 10^{-4}\times S^2$$
$$C = 1.2026 - 5.3566\times 10^{-4}\times S + 1.8906\times 10^{-6}\times S^2$$
$$D = 6.8774\times 10^{-7} + 1.517\times 10^{-6}\times S - 4.4268\times 10^{-9}\times S^2$$

Therefore, the physical exergy equation (kg/s) for any saline stream is obtained as:

$$\dot{Ex}_{ch} = \dot{m}(N_{mol}(S, M_w, M_s)\times 10^{-3}\times 8.314\times T_o\{-X_w\times\log X_w - X_s\times\log X_w\})$$

And the total stream exergy rate is then calculated,

$$\dot{Ex}_{total} = \dot{Ex}_{ph} + \dot{Ex}_{ch}$$

where

$$X_w = N_{pure}(S, M_w)/N_{mol}(S, M_w, M_s),$$
$$X_s = N_{salt}(S, M_w)/N_{mol}(S, M_w, M_s) \text{ and}$$
$$N_{pure} = \frac{1000 - S}{M_w}, \quad Nsalt = S/M_s$$

$N_{mol} = N_{pure} + N_{salt}$ is the number of particles, and X_w, X_s is the fraction of water and salt (mole), and the molar weight $M_{w,s}$ for water and salt is 18 g and 58.5 g respectively. The inlet exergy feed stream for the air compressor is obtained base on the specific enthalpy, temperature and specific entropy as $\dot{Ex}_{i,o} = f(T, s, h = \dot{m}\Delta h - \dot{m}T\Delta s)$.

Appendix D

Cost analysis for all units that been considered in this study.

MED [31]

Interest rate, %	5
Plant life time, y	20
Amortization factor, 1/y	$A_f = \frac{i.(1+i)^{LTp}}{(1+i)^{LTp}-1}$
Direct capital costs, $	$DCC = 9 \times 10^5$
Annual fixed charges, $/y	$AFC = A_f \times DCC$
Annual heating steam costs, $/y	$AHSC = \frac{SHC \times L_s \times LF \times M_d \times 365}{1000 \times PR}, SHC = \frac{1.466\$}{MkJ}$
Annual chemical cost, $/y	$ACC = SCC \times LF \times M_d \times 365, SCC = 0.025\ \$/\mathrm{m}^3$
Annual labor cost, $/y	$ALC = SLC \times LF \times M_d \times 365, SLC = 0.1\ \$/\mathrm{m}^3$
Total annual cost, $/y	$TAC_{MED} = AFC + AHSC + AEPC + ACC + ALC$
Operating and maintenance costs, $	$OMC_{MED} = 0.02 \times DCC$
Hourly operating & maintenance costs (UHC) in $/h	$Z_{MED}^{IC\&OM} = \frac{OMC_{MED} \times A_f + AFC}{8760}$

MSF [31]

For MSF desalination plant, the annual fixed charges in $/y may represented by [31] as following;

$AFC = A_f \times (DCC + IDCC)$, and $IDCC$ is the indirect capital costs and equal to $0.4 \times DCC$ [31]. The operating and maintenance costs are presented in $ as following;

$OMC = 0.02 \times (DCC + IDCC)$ and the annual chemical cost is obtained in $/y as; $ACC = SCC \times LF \times D_p \times 365$

Where, SCC is the specific chemical costs (0.025 $/m^3 [31]), and LF is the plant load factor and is fixed at 0.9, and D_p is the distillate product. The annual labor costs in $/y is given as following; $ALC = SLC \times LF \times D_p \times 365$ Where, SLC is the specific labor costs (0.1 $/m^3). The total annual costs in $/y for MSF is calculated according to the following, $TAC_{msf} = AFC + ACC + ALC + (OMC \times A_f)$.

The operating and maintenance cost (UHC) in $/h for MSF ($Z_{msf}^{IC\&OM}$) is found to be as following; $Z_{msf}^{IC\&OM} = (OMC \times A_f + AFC)/8760$

Steam generator [32,33]

$Z_{SG}^{IC\&OM} = C_f \times M_{wg} \times 3.6$ where C_f is the fuel cost (fuel cost is adjusted as $0.1354\$/M_f^3$ [32,33] (natural gas operation). Then the total capital cost is calculated $TCC_{SG} = Z_{SG}^{IC\&OM} \times OH \times 365$, where, OH is the operating hours and fixed at 24 h

Flash Separation unit [34]

$Z_{fsh}^{IC\&OM} = 8.52$ [34] and the total capital cost for the flash unit is then calculated, $TCC_{fsh} = Z_{fsh}^{IC\&OM} \times OH \times 365$

ORC [35,36]

The indirect cost for ORC is calculated as $IC_{ORC} = 3000 \times W_{ORC}$ (2000–4000 $/kW [35]) and the total capital cost is then obtained, $TCC_{ORC} = IC_{ORC} \times A_f$, and $Z_{fsh}^{IC\&OM} = TCC_{ORC}/(OH \times 365)$

GTC [37]

The gas turbine cycle cost, $/kW is obtained based on the following correlation [37];

$C_{GTC} = 332.6 \times \exp(-4.612e-05 \times W_{net}) + 274.2 \times \exp(-1.756e-06 \times W_{net})$, and the total capital cost is then calculated as, $TCC_{GTC} = W_{net} \times C_{GTC} \times A_f$ and the hourly cost is then calculated as, $Z_{GTC}^{IC\&OM} = TCC_{GTC}/(OH \times 365)$

Total Plant cost

1. WGC-MSF-MED:
   The total hourly costs: $Z_{Total}^{IC\&OM} = Z_{SG}^{IC\&OM} + Z_{fsh}^{IC\&OM} + Z_{Pump}^{IC\&OM} + Z_{msf}^{IC\&OM} + Z_{med}^{IC\&OM}$, $/h, and the total Plant Costs:
   $TPC_{1stscenario} = TCC_{SG} + TCC_{fsh} + TCC_{pump} + TCC_{msf} + TCC_{med}$, $/y and the total water price is calculated as,
   $TWP_{1stscenario} = TPC_{1stscenario}/M_{d_{msf+med}} \times 365 \times LF$, $/m^3
2. WGC-MSF-MED-ORC:
   The total hourly costs: $Z_{Total}^{IC\&OM} = Z_{SG}^{IC\&OM} + Z_{fsh}^{IC\&OM} + Z_{Pump}^{IC\&OM} + Z_{msf}^{IC\&OM} + Z_{med}^{IC\&OM} + Z_{ORC}^{IC\&OM}$, $/h, and the total Plant Costs:
   $TPC_{2ndscenario} = TCC_{SG} + TCC_{fsh} + TCC_{pump} + TCC_{msf} + TCC_{med} + TCC_{ORC}$, $/y and the total water price is calculated as,
   $TWP_{2ndscenario} = TPC_{2ndscenario}/M_{d_{msf+med}} \times 365 \times LF$, $/m^3
3. WGC-MSF-GTC:
   The total hourly costs: $Z_{Total}^{IC\&OM} = Z_{GTC}^{IC\&OM} + Z_{med}^{IC\&OM}$, $/h, and the total Plant Costs: $TPC_{3rdscenario} = TCC_{GTC} + TCC_{msf}$, $/y and the total water price is calculated as, $TWP_{3rdscenario} = TPC_{3rdscenario}/M_{d_{msf}} \times 365 \times LF$, $/m^3

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