THERMOECONOMIC ANALYSIS OF MULTI STAGE FLASH-THERMAL VAPOR COMPRESSSION (MSF-TVC) DESALINATION PROCESS

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ABSTRACT

This work presents the design and thermoeconomic analysis of a proposed multi stage flash-thermal vapor compression MSF-TVC system. The proposed MSF-TVC system is analyzed and investigated under different operating conditions by using the thermoeconomic methodology. The optimum operating conditions of the proposed system is achieved at 6 kPa entrained pressure, top brine temperature110ºC, splitter ratio equal to 0.28. Under these conditions the unit product cost is calculated by 2.15 $/m^3$. The comparison between the proposed MSF-TVC system and the conventional MSF system showed that the gain ratio of the MSF-TVC system is 96% higher than that of the conventional MSF brine circulation plant. The heat transfer area of the MSF-TVC is 52 % higher than the conventional MSF. The exergetic efficiency of the MSF-TVC system is 46 % higher than that of the MSF system. The unit product cost of the MSF-TVC system is 19 % lower than that of the conventional brine circulation multi stage flash (MSF-BR) system.

Keywords: Thermoeconomics, Exergy, Desalination, MSF, TVC, Unit product cost.

INTRODUCTION

One of the goals of the sustainable development in the desalination field is improving thermal efficiency at low product cost. Our previous work [1-2] in the modified multi stage flash MSF process showed that, the improvement through the configuration changes is recommended. The merits of using vapor compression upon combination with multi effect evaporation motivated to the development of the MSF brine circulation with thermal vapor compression (MSF-TVC).

El-Dessouky *et al*. [3] proposed and evaluated a combination of MSF with thermal vapor compression TVC. The energy analysis of the proposed MSF-TVC showed that the gain ratio (GR) decreased when the vapor is entrained from the heat rejection sections while the specific cooling flow rate also decreased and no change in the heating surface area [3].

A design improvement journey for fuel-driven sea water distillation systems was conducted by El-Sayed [4]. The journey started with a simple boiler –MSF distiller of 10 migd of desalted water, the top brine temperature and reject temperature are 100ºC and 38ºC respectively. The system producing its needs for steam in a gas fired boiler and importing its needs for electricity. The simple system flowsheet is improved by substituting the throttle valve by a steam ejector to upgrade a small fraction of low temperature vapor of the heat rejection section to a heating steam. The results showed that 10 % increase of the system exergetic efficiency and a reduction in the unit product cost by 6.6 % are obtained due to use MSF-TVC. The reduction cost in MSF-TVC system results from enhancement in the gain ratio (GR) and a decrease in capital cost of MSF (heating surface area) more than the cost of added effect [4].

The contradicted results about the gain ratio and the heating surface which are illustrated in [3] and that illustrated in [4] showed that the thermal vapor compression with MSF still needs more investigations. So, this work deals with the design and thermoeconomic analysis of a proposed multistage flashing desalination combined with thermal vapor compression (MSF-TVC) configuration. A system of a 5000 m^3/d desalinated water capacity is investigated. The system extracts its heating steam and electricity from a steam power generation plant. The suction pressure of the vent chamber, the temperature of brine directed to the vent chamber, the top brine temperature TBT, splitter ratio of the make up flow rate, ejector compression ratio are considered as decision variables. The MSF-TVC performance ratio, the ejector efficiency, and the unit product cost are considered as the dependent variables. The decision variables are varied with its limits range to reach the optimum operating conditions. The developed VDS package is used as a tool for design and thermoeconomic analysis [5]. The present work is organized to study the effect the entrained vapor pressure then considered the effect of top brine temperature. Thermoeconomic variables are calculated for process units from which the deficient unit is focused. The effect of the cooling water splitter ratio α_1 is investigated. The comparison between the proposed system and the conventional MSF is presented.

PROCESS DESCRIPTION

Flow sheet diagram for MSF-TVC is shown in Fig. (1). The system constitutes the brine heater, the flashing stages, and the steam jet ejectors. The flashing stages divided among heat recovery and heat rejection sections. Each flashing stage includes brine pool, submerged orifice, vapor space, demister, distilled collection tray, and condenser tubes. The brine stream absorbs the latent heat of condensing steam and its temperature increase to its maximum design value known as the top brine temperature. The hot brine enters the flashing stages in the heat recovery section and then in the heat rejection section, where a small amount of fresh water vapor is formed by brine flashing in each stage. The flashed off vapors condenses on the outside surface of the condenser tubes. The condensed water vapor accumulates across the stages and forms the distillate product stream. The intake seawater stream is introduced into the condenser tubes of the heat rejection section, where its temperature is increased by absorption of the latent heat of the condensing of fresh water vapor. The warm stream of intake sea water is divided into two parts: the first is the cooling seawater which is rejected back to the sea, and the second is the feed seawater, which is de-aerated, chemically treated and then mixed in the brine pool of the flashing stages in the heat rejection section. The brine recycle stream is extracted from the brine pool of the last stage in the heat rejection section and is introduced into the condenser tubes of the last stage of the heat recovery section. The remaining brine in the heat rejection section known as the brine blow down is rejected to the sea. The steam jet ejector entrains a specified portion of the vapor formed in the flashing stage in the heat rejection section. The motive steam compresses the entrained vapor to the desired temperature and pressure. The compressed vapor is then used to heat brine recycle stream in the brine heater. Two stages of ejector in series are required to compress vapor from 32 to 113ºC as the design operation of steam ejector puts maximum limit of 5 on the compression ratio [3].

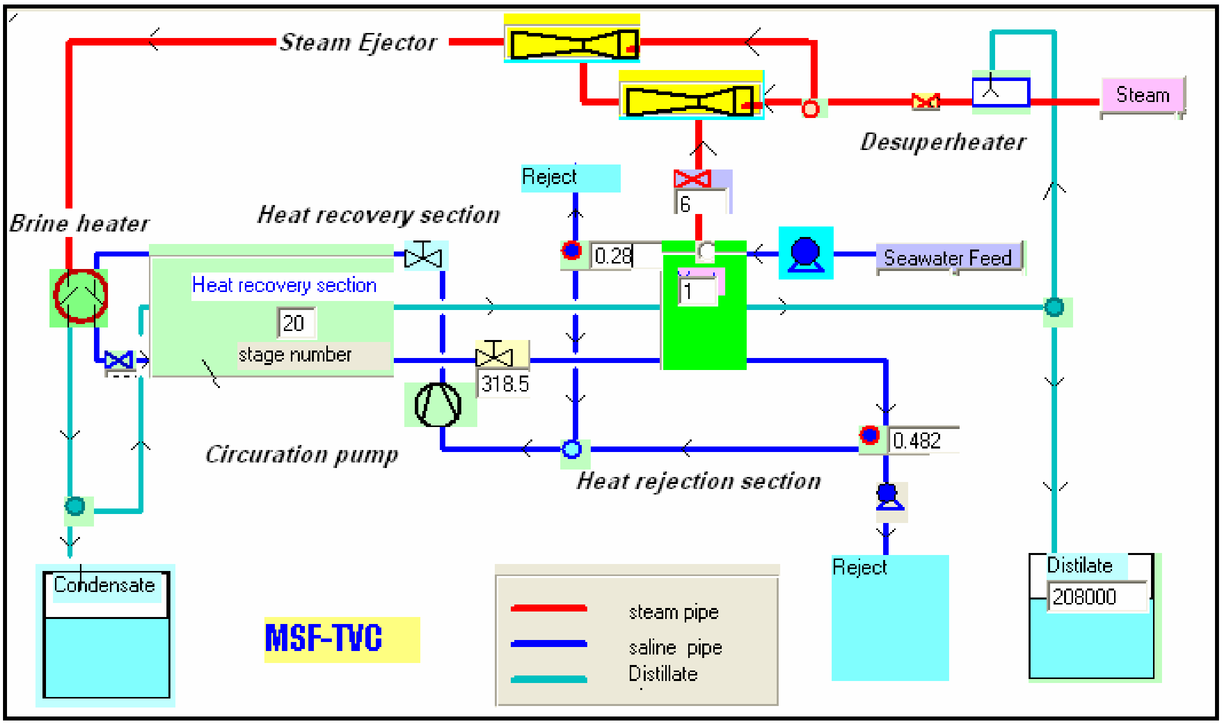


Figure (1): Flow sheet of the proposed MSF-TVC system

MATHEMATICAL MODEL

Using thermoeconomic, energy, exergy and cost balance equations of the plant are considered. The details of the mathematical models of mass and energy equations for the system components such as evaporator, pumps, desuperheater and brine heater are illustrated in [1, 2]. The details of exergy and thermoeconomic mathematical models are illustrated in [5].

The capital investment and operating & maintenance of each component in the MSF-TVC configuration are calculated using the illustrated relations and specified values in table (1).

Table (1): cost data of the process unit

Unit	Equation	Reference
Feed Pump, $	$1000\times 32\times 0.000435\times(\dot{M}_{water})^{0.55}\Delta P^{0.55}\left(\frac{\eta}{1-\eta}\right)^{1.05}$ *M*, kg/s; ΔP, kPa	El-Sayed [12]
Heat exchanger area, $/m^2.hr	0.0096	Nafey *et al*. [1]
Steam Ejector, $	$8\times 989\times \dot{M}_{vapor}\times(T_i / P_i)^{0.05} P_e^{-0.75}$ $\dot{M}$, kg/s; P_i, P_e, MPa ; T_i, k	El-Sayed [4]
Steam cost, $/ton	13.5	Nafey *et al*. [1]
Intake cost, $/hr	15	Nafey *et al*. [1]

The total annual investment cost of each unit is calculated according the following relation:

$$\text{Annual capital investment} = \text{Present value} \times \frac{i\times(1+i)^N}{(1+i)^N - 1} \qquad (1)$$

The annual capital investment includes the purchased cost, transportation, and installation. Using an interest rate, i = 5 % and the amortization year, N = 20 years. Multiplying the value of equation (1) by a cost index of 1.2 [12], the annual capital cost in the time of evaluation is calculated as:

Then, the hourly cost equals to $= \frac{\text{Total annual investment(capital+running)}}{365\times 24\times 0.9}$ $/hr. This cost is generated by the program as a capital &operating and maintenance (CI+OM) cost for each unit.

Chemicals are used for pre-treatment and post-treatment of feed water and distillate respectively in desalination plant. These chemicals are including sulfuric acid for adjusting pH and preventing calcium carbonate $CaCO_3$ scaling. Various anti-scalants are used (e.g. Belgard E.V.) to prevent $CaSO_4$ precipitation. Sodium Bi sulfite Na_2SO_3 is used for de-chlorination of the make up sea water. Table (2) shows the dosing rate and the chemical cost used in the desalination plant [3]. The desalination plant is treating the make up sea water by using the anti-scalant and the sodium sulfite. The hydraulic acid is used for period cleaning. The hourly cost of the used chemicals is burden on the distillate water output.

Table (2): Chemical cost and dosing rate,[4]

Chemical	**Unit cost ($/kg)**	**Dosing rate (g/ton)**
Pre-treatment for make up		
Sulfuric acid, H_2SO_4, (Demineralization)	0.504	24.2
Caustic soda, $NaOH$, (Demineralization)	0.701	14
Belgard EV2030, (Antiscalant)	1.9	5-14.4
Sodium Sulfite, Na_2SO_3, (Dechlorination)	0.1	5
Post-treatment for distillate		
Chlorine	0.482	4

The developed VDS package is utilized to solve mass, pressure, energy equations of the considered iteratively to obtain the state point in the system. Then the exergy flow rate of the streams is calculated. Finally the cost balance equation model is solved to obtain the monetary cost flow rate of the streams. Then the following parameters are calculated as follows:

- Gain ratio (GR) $= \dfrac{distilled \text{ flow rate}}{steam \text{ flow rate}} = \dfrac{\dot{m}_d}{\dot{m}_s}$ (2)
- Specific heat transfer area (SA) $= \dfrac{Area\ (\text{m}^2)}{Distilled\ (\text{T/hr})}$ (3)
- Exergetic efficiency (η_{II})$= \dfrac{Exergy \text{ output}}{Exergy \text{ input}} = \dfrac{\dot{E}_P}{\dot{E}_F}$ (4)
- Unit product cost , \$/m^3 $= \dfrac{\dot{C}_{distilled} + \dot{C}_{loss}\ (\$/\text{hr})}{Distilled \text{ flow rate (T/hr)}}$ (5)
- Brine circulation ratio (BCR) $= \dfrac{brine \text{ recycle flow rate}}{\text{Distiled out put flow rate}}$ (6)
- Entrained ratio (Ra) $= \dfrac{motive \text{ steam}}{entrained \text{ vapor}}$ (7)
- Compression ratio (Cr) $= \dfrac{Compressed \text{ vapor pressure}}{Entrained \text{ vapor pressure}}$ (8)

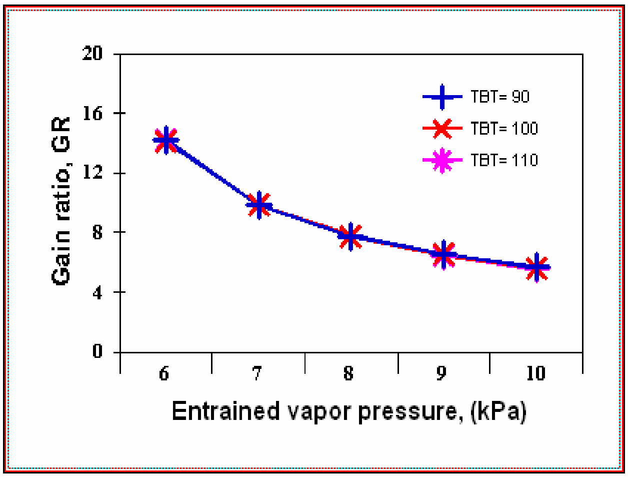
RESULTS AND DISCUSSION

Table (3) shows the steam ejectors analysis with the entrained pressure variations. This table shows that the entrained ratio (Ra) has a minimum value in the two ejectors at entrained pressure of 8 kPa. Table (3) shows also that the cross section area of the nozzle throat increases, with increasing the entrained pressure, however, the nozzle exit area and the diffuser throat decrease. The nozzle throat of the secondary stage is greater than that of the primary ejector. This is because most motive steam is directed to the secondary ejector. The entrained ratio of the secondary stage is generally greater than that of the first stage. The exergy destruction in the first ejector has a minimum value at the entrained suction pressure of 8 kPa while the minimum exergy destruction in the second ejector is obtained at the entrained pressure of 7 kPa. As a result the exergetic efficiency of both steam ejectors has a maximum value at 8 and 7 kPa respectively as shown in Table (3). The numerical results reveal that the exergy destruction will be negative value if the entrained pressure lowered more than 6 kPa. From the second law of thermodynamics view point, the negative value of the exergy destruction means impossible process occurrence. On the other hand the economic visibility limits the higher values of the entrained pressure.

Table (3): Design results of two steam ejectors connected in series.

Entrained pressure, P_E, kPa	6	7	8	9	10
First Stage Ejector					
Entrained ratio, Ra	0.56	0.49	0.48	0.55	0.62
Compression ratio[&], Cr*	5	5	5	5	5
Nozzle area, $A_{th,n}$, cm^2	9	13	17	20	23
$A_{e,n}/A_{th,n}$	32	28	25	23	21
$A_{th,d}/A_{th,n}$	121	113	101	83	69
Exergy destruction, MW	0.16	0.09	0.02	0.13	0.25
Exergetic efficiency	0.89	0.96	0.99	0.96	0.94
Second Stage Ejector					
Entrained ratio, Ra	0.92	0.85	0.83	0.91	0.98
Compression ratio, Cr*	5	4.5	4	3.5	3.2
Nozzle area, $A_{th,n}$ cm^2	24	34	44	52	60
$A_{e,n}/A_{th,n}$	8.7	7.7	7	6.4	6
$A_{th,d}/A_{th,n}$	19	19	19	18.2	17
Exergy destruction, MW	0.07	0.05	0.13	0.47	0.95
Exergetic efficiency	0.98	0.99	0.98	0.95	0.93

[*] Specified parameters



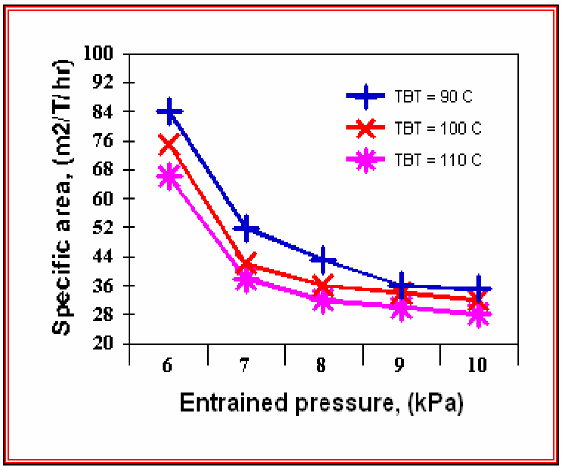


Fig. (2.a): Gain ratio of MSF-TVC variation vs. the entrained pressure

Fig. (2.b): Specific area variation of MSF-TVC vs. the entrained pressure

Figure (2.a) shows the variation of the gain ratio (GR) of the MSF-TVC against the pressure of the entrained vapor at different top brine temperature. The temperature of the brine which inlets to the last chamber is controlled by the suction pressure variation for feasibility of the system convergence.

The gain ratio (GR) decreases as the entrained pressure increases. This is because the mass of the flashed vapor decreases with the increase of entrained pressure so the motive steam increases. This figure shows that the gain ratio of the MSF-TVC has a little response to the top brine temperature (TBT). This is because the increase in the TBT has a direct effect on the heat transfer area not on the steam consumption.

Figure (2.b) shows the variation of the specific heat transfer area (unit area per unit distilled out put) against the entrained pressure. This figure shows that the total heat transfer area decreases with the increase of the entrained pressure. The heat transfer area of the brine heater increases as the entrained pressure increases as shown in table (4). This is because the cooling water enters the brine heater at a relatively higher thermal energy. Table (4) shows also that the heat transfer area of the vent chamber (rejection section) increases with the increase of the entrained pressure. This is because of the decrease of the logarithmic temperature difference *LMTD* of the condenser for the same thermal load. The heat transfer area of the recovery stages decreases according to the increase in the entrained pressure as shown in table (4). This is due to the increase in the *LMTD* of the heat recovery section. The inverse effect in the heat transfer area of brine heater, heat recovery chambers and vent chamber (rejection) showed that the total heat transfer area decreases as shown in table (4) and figure (2.b). Figure (2.b) shows also that the specific area inversely proportional to the TBT.

Table (4): Effect of entrained pressure variation on the MSF-TVC system

Entrained pressure, kPa*	6	7	8	9	10
Capacity, T/hr, *	**208**	**208**	**208**	**208**	**208**
$T_{seawater}$, °C , *	27	27	27	27	27
Rejected brine temp., °C , *	40	40	40	40	40
$T_{b,out,recovery}$, °C, *	45.5	51.5	56.5	60	63
TBT, °C, *	**110**	**110**	**110**	**110**	**110**
T_{steam} inlet the B/H , °C	111	113	113	113	111.5
Intake seawater, (T/h)*	1370	1370	1370	1371	1370
α_1, brine recycle ratio*	0.482	0.482	0.482	0.482	0.482
Make-up flow rate , (T/h)	660	660	660	660	660
feed salt concentration, g/l, *	45	45	45	45	45
X_r in circulation brine, g/l, *	63	63	63	63	63
Stages numbers, *	20 +1	20 +1	20 +1	20 +1	20 +1
B/H area, m^2	516	802	972	1140	1141
Heat recovery area, m^2	12195	5777	3991	3368	2971
Heat rejection area, m^2	1099	1462	1702	1773	1822
Total surface area, m^2	**13809**	**8040**	**6665**	**6281**	**5934**
Gain ratio	**14.5**	**9.8**	**7.7**	**6.5**	**5.6**
Exergy loss, MW	4	4	4.3	4.4	4.6
Exergy destruction, MW	4.2	5.9	7.3	8.6	9.8
Running cost, $/hr	310	412	502	585	662
Capital cost, $/hr	155	100	88	84	81
Chemical cost, $/hr	22	22	22	22	22

* Specified parameters

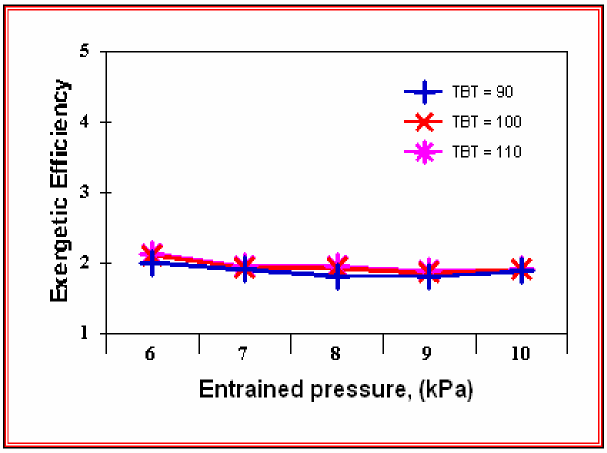


Fig. (2.c): Exergetic efficiency of MSF-TVC variation vs. the entrained pressure

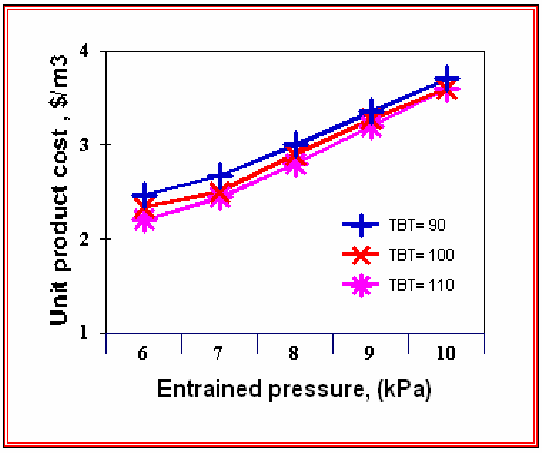


Fig. (2.d): Unit cost of MSF-TVC variation vs. the entrained pressure

Figure (2.c) shows the exergetic efficiency of the MSF-TVC system versus the entrained pressure variation. This figure shows that the exergetic efficiency of the system has a little decrease with the increase of the entrained pressure. Figure (2.c) shows also that the top brine temperature variation has a little effect on the exergetic efficiency. Table (4) shows that, the exergy losses associated with the rejected blow down increases with the increase in the entrained pressure; that is because of the increase of the blow down stream temperature. Table (4) shows also that the exergy destruction within the system increases as the entrained pressure increases that is owing to the irreversibility of heat transfer, and flashing process in the MSF stages. In addition the irreversibility due to the entrained vapor mixing and the presence of shock waves in the ejectors.

Figure (2.d) shows the unit product cost variation of the MSF-TVC system versus the entrained pressure. The thermoeconomic results show that the chemical cost of the treatment is constant while the entrained pressure is varying as shown in table (4). This is because the make up flow rate unchanged (660 ton/hr) with the entrained pressure variation. Table (4) shows that the running cost increases with increase of the suction pressure; that is due to the increase of the motive steam consumption and the increase of the pumping cost due to the increase of the brine circulation. Table (4) shows also that the capital cost however decreases with the increasing of the entrained suction pressure that is because of the decrease of the heating surface area. The inverse effect of the running and capital cost results in a minimum unit product cost at suction pressure of 6 kPa as shown in Figure (2.d).

Table (5) shows the effect of the heat recovery section stage's number on the design of the MSF-TVC system. This table shows that there is no visible solution for stages number less than 14. The maximum number of stages for the considered design conditions is limited by 21. This is because the maximum number of stages depends on the flashing range and the temperature drop due to flashing process inside chamber. The flashing range is fixed by 110 - 45.5 = 64.5. The temperature drop due to non equilibrium allowance (NEA), boiling point rise (BPR) and temperature loss due demister pressure drop. This value is calculated at brine salinity of 68 g/l by approximately of 2.5-3ºC. So the maximum number of stages is calculated by 64.5 /3 = 21.4. Table (5) shows also that the gain ratio (GR) unchanged with the stages number of the recovery section variation. This owing to constant value of consumed steam that heats the brine to the fixed TBT =110ºC. This is because of the nearly constant value of brine circulation ratio (BCR) and the LMTD across the brine heater. That in turn makes the brine heater area experiencing no change with increasing the stages number as shown in table (5). The area of the rejection section nearly experiences no change. This is due to constant LMTD and the thermal load. The area of the recovery section decreases with increasing the stages number that is owing to the constant thermal load and LMTD of the heat recovery section while the stages number increasing. The outcome results of table (5) show that the total surface area decreased dramatically with the increase of the stages number of the recovery section. Consequently, the unit product cost is reduced due to the increase of the stages number as shown in table (5). This table shows that the low unit product cost is obtained at 20 stages of heat recovery section. It should be noted there is no convergence beyond 20 stages as mentioned above.

Table (5): Effect of varying the number of stages of recovery section

Stage No.	**12**	**14**	**16**	**18**	**20**
Entrained pressure*, kPa	No visible solution	6	6	6	6
Distilled out put*, m^3/h		208	208	208	208
TBT, ºC		110	110	110	110
$T_{b,out,recovery}$ *, ºC		45.5	45.5	45.5	45.5
B/H area, m^2		516	516	516	516
LMTD of heat recovery, ^0C		5	5	5	5
BCR		8.8	8.8	8.8	8.8
First steam ejector, Ra		0.56	0.56	0.56	0.56
Second steam ejector, Ra		0.92	0.92	0.92	0.92
Heat recovery area, m^2		19470	15079	13246	12194
Heat rejection area, m^2		1099	1099	1099	1099
Total surface area, m^2		21,782	16,694	14,861	13,809
Gain ratio, GR		14.2	14.2	14.2	14.2
Unit product cost, $\$/m^3$		2.58	2.35	2.26	2.21

Table (6) summarizes the thermoeconomic variables which are calculated for each unit of the MSF-TVC system. These variables include the exergetic efficiency, rate of exergy destruction, average costs per unit fuel exergy c_F, and product exergy c_p, cost rate of exergy destruction $\dot{C}_D$, investment and O & M cost rate $\dot{Z}^{CI+OM}$, relative cost difference *r*, and exergoeconomic factor *f*. Table (6) shows that 20 stages of the heat recovery section and the vent chamber have the highest values of the sum $\dot{Z}+\dot{C}_D$ in the MSF-TVC process. Therefore, these are the most important units to be considered for improvement from the thermoeconomics point of view. The zero value of the exergoeconomic factor (*f*) for the desuperheater shows that the costs associated with the desuperheater are almost exclusively due to exergy destruction. Table (6) also shows that, the relatively high value of (*f*) and relative cost difference (r) of the first stage steam ejector among all units suggests a reduction in the capital investment costs of these components. The exergy destruction in the vent chamber represents 12% of the total exergy destruction in the whole system.

Table (6): Thermoeconomic analysis of the MSF-TVC at α_1 = 0.482 & $T_{b,out,recovery}$ =45.5

Unit	Exergy Efficiency	Exergy destruction	cf	cp	CD	Z	CD +Z	r	f
		MW	$/GJ	$/GJ	$/h	$/h	$/h		
Desuperheater 0	0.93	0.30	15.01	16.16	16.24	0.00	16.24	0.07	0.00
Ejector1	0.89	0.16	12.43	14.23	7.16	1.20	8.36	0.13	0.14
Ejector2	0.98	0.07	11.15	11.39	2.81	0.56	3.37	0.02	0.17
brine heater0	0.73	0.81	21.09	29.22	61.27	5.00	66.27	0.39	0.08
stage:1	0.84	0.27	60.18	72.41	59.49	5.91	65.40	0.17	0.09
stage:2	0.89	0.18	66.77	76.32	43.01	5.91	48.93	0.13	0.12
stage:3	0.89	0.16	70.39	79.94	40.84	5.91	46.75	0.12	0.13
stage:4	0.90	0.14	74.02	83.57	38.63	5.91	44.54	0.11	0.13
stage:5	0.90	0.13	77.68	87.26	36.45	5.91	42.37	0.11	0.14
stage:6	0.91	0.12	81.43	91.07	34.40	5.91	40.31	0.11	0.15
stage:7	0.91	0.11	85.29	95.06	32.53	5.91	38.45	0.10	0.15
stage:8	0.91	0.10	89.33	99.31	30.92	5.91	36.83	0.10	0.16
stage:9	0.92	0.09	93.60	103.92	29.61	5.91	35.52	0.10	0.17
stage:10	0.92	0.08	98.18	108.99	28.66	5.91	34.57	0.10	0.17
stage:11	0.92	0.08	103.15	114.68	28.13	5.91	34.04	0.10	0.17
stage:12	0.91	0.07	108.63	121.18	28.07	5.91	33.99	0.10	0.17
stage:13	0.91	0.07	114.75	128.72	28.55	5.91	34.47	0.11	0.17
stage:14	0.90	0.07	121.70	137.65	29.64	5.91	35.56	0.12	0.17
stage:15	0.89	0.07	129.71	148.44	31.44	5.91	37.36	0.13	0.16
stage:16	0.88	0.07	139.09	161.76	34.08	5.91	39.99	0.14	0.15
stage:17	0.86	0.07	150.27	178.62	37.73	5.91	43.64	0.16	0.14
stage:18	0.84	0.07	163.85	200.63	42.64	5.91	48.55	0.18	0.12
stage:19	0.80	0.08	180.70	230.49	49.18	5.91	55.09	0.22	0.11
stage:20	0.76	0.08	202.14	273.12	57.92	5.91	63.83	0.26	0.09
Brine mixer1	1.00	0.02	33.33	33.43	2.41	0.00	2.41	0.00	0.00
Circu. pump	0.82	0.08	30.15	37.72	8.31	3.58	11.89	0.20	0.30
Vent MSF1	0.30	0.46	119.68	276.88	198.78	10.55	209.33	0.57	0.05
B/D Pump	0.67	0.01	27.78	45.31	1.47	0.32	1.79	0.39	0.18
FEED pump	0.78	0.03	25.10	40.11	2.50	1.19	3.69	0.37	0.32

Table (7): Thermoeconomic analysis of the MSF-TVC at α_1 = 0.482 & $T_{b,out,recovery}$ =44.5

Unit	Exergy Efficiency	Exergy destruction	cf	cp	CD	Z	CD +Z	r	f
		MW	$/GJ	$/GJ	$/h	$/h	$/h		
Desuperheater (	0.93	0.30	15.01	16.16	16.41	0.00	16.41	0.07	0.00
Ejector1	0.72	0.36	13.74	19.21	17.81	0.71	18.52	0.28	0.04
Ejector2	0.92	0.29	12.15	13.20	12.68	0.42	13.10	0.08	0.03
brine heater0	0.74	0.69	22.20	30.21	55.45	0.46	55.91	0.36	0.01
stage:1	0.85	0.27	66.49	79.77	64.78	8.92	73.70	0.17	0.12
stage:2	0.90	0.17	73.69	83.89	45.27	8.92	54.19	0.12	0.16
stage:3	0.90	0.15	77.69	87.81	42.33	8.92	51.25	0.12	0.17
stage:4	0.91	0.13	81.70	91.73	39.37	8.92	48.29	0.11	0.18
stage:5	0.91	0.12	85.76	95.72	36.48	8.92	45.39	0.10	0.20
stage:6	0.92	0.10	89.92	99.84	33.74	8.92	42.65	0.10	0.21
stage:7	0.92	0.09	94.22	104.17	31.23	8.92	40.15	0.10	0.22
stage:8	0.93	0.08	98.74	108.79	29.02	8.92	37.94	0.09	0.24
stage:9	0.93	0.07	103.53	113.81	27.17	8.92	36.09	0.09	0.25
stage:10	0.93	0.07	108.70	119.35	25.74	8.92	34.66	0.09	0.26
stage:11	0.93	0.06	114.33	125.58	24.78	8.92	33.70	0.09	0.26
stage:12	0.93	0.06	120.58	132.72	24.34	8.92	33.26	0.09	0.27
stage:13	0.93	0.05	127.61	141.03	24.49	8.92	33.41	0.10	0.27
stage:14	0.92	0.05	135.64	150.88	25.30	8.92	34.21	0.10	0.26
stage:15	0.92	0.05	144.95	162.81	26.86	8.92	35.78	0.11	0.25
stage:16	0.90	0.05	155.93	177.55	29.31	8.92	38.23	0.12	0.23
stage:17	0.89	0.05	169.11	196.24	32.82	8.92	41.74	0.14	0.21
stage:18	0.87	0.06	185.25	220.64	37.66	8.92	46.58	0.16	0.19
stage:19	0.84	0.06	205.46	253.73	44.22	8.92	53.14	0.19	0.17
stage:20	0.80	0.06	231.46	300.90	53.12	8.92	62.03	0.23	0.14
Brine mixer1	1.00	0.02	37.12	37.22	2.68	0.00	2.68	0.00	0.00
Circu. pump	0.82	0.08	30.15	37.72	8.31	3.58	11.89	0.20	0.30
Vent MSF1	0.37	0.34	149.78	326.56	185.69	8.87	194.57	0.54	0.05
B/D Pump	0.67	0.01	27.78	45.31	1.47	0.32	1.79	0.39	0.18
FEED pump	0.78	0.03	25.10	40.10	2.50	1.19	3.69	0.37	0.32

By reducing the brine temperature before introducing to the pool of the vent chamber from 45.5 to its lower limit of 44.5ºC; the sum $\dot{Z}+\dot{C}_D$ of the vent chamber is reduced by 48.5 % as shown in table (7). The capital investment (Z) of the steam ejectors is sequentially reduced by 6 %. However the capital investment of the heat recovery section (Z) increases due to the increase in the heat transfer area as shown in table (7). As the investment of the 20 stages is dominated, the unit product cost is increased by 12%. So it is not economical to reduce the temperature of the brine before introducing it to the pool of the vent chamber.

The splitter ratio of the make up flow rate (α_1) is considered as a key design variable. The results of varying the splitter ratio (α_1) from 0.482 to 0.28 and fixing the entrained pressure of 6 kPa are shown in table (8). Figure (3) shows the interface of the MSF-TVC system with the design results. Table (8) shows that by reducing α_1 from 0.482 to 0.28, the gain ratio (GR) increases by 11 %. However, the total heat transfer is increased by 14 %. As a result the unit product cost is reduced only by 3 %. The numerical results show that there is no feasible system solution at further reduction of the splitter ratio (α_1).

Table (8): Analysis of the MSF-TVC system at different value of α_1

α_1, brine recycle ratio*	0.482	0.28
Entrained pressure, kPa*	6	6
Capacity, T/hr, *	**208**	**208**
$T_{b,out,recovery}$, ºC, *	45.5	45.5
TBT, ºC, *	**110**	**110**
Intake seawater, (T/h)*	1370	1370
Make-up flow rate , (T/h)	660	384
feed salt concentration, g/l, *	45	45
X_r in circulation brine, g/l, *	63	63
X_{reject}, in reject brine, g/l	70	70
Stages numbers, *	20+1	20 +1
B/H area, m^2	516	304
Heat recovery area, m^2	12,195	14282
Heat rejection area, m^2	1098	1098
Total surface area, m^2	**13,809**	**15685**
Gain ratio, GR	**14.5**	**15.7**
Unit product cost, $/$m^3$	**2.23**	**2.15**

* Specified parameters

The MSF-TVC system at its economical design conditions (entrained pressure = 6 kPa and TBT =110ºC, $T_{b,out,recovery}$ = 45.5ºC) is compared with the conventional brine circulation MSF system. Table (9) shows that, under the same operating and design conditions, the gain ratio of the MSF-TVC system is 96% higher than that of the conventional MSF-BR system. The heat transfer area of the MSF-TVC is 52 % higher than the conventional MSF. The exergetic efficiency of the MSF-TVC system is 46 % higher than that of the MSF system. Consequently, the unit product cost of the MSF-TVC is 19 % less than the MSF configuration.

Table (9): Comparison between MSF-TVC and MSF configurations

		MSF	MSF-TVC
Entrained pressure, kPa*		-	6
Capacity, T/hr, *		**208**	**208**
$T_{seawater}$, ºC , *		27	27
$T_{recycle}$, ºC		39.85	38
$T_{b,out,recovery}$, ºC, *		48.8	44.5
TBT, ºC, *		**110**	**110**
T_{steam}, ºC, *		113	111.5
Intake seawater, (T/h)*		1371	1370
Brine recirculation, (T/h)		1856	1836
α_1, brine recycle ratio*		0.482	0.28
Make-up flow rate , (T/h)		660	384
feed salt concentration, g/l, *		45	45
X_r in circulation brine, g/l, *		63	63
X_{reject}, in reject brine, g/l		70	70
Stages numbers, *		17+3	20 +1
B/H area, m^2		529.6	293
Heat recovery area, m^2		8954	14282
Heat rejection area, m^2		839	1098
Total surface area, m^2		**10,323**	**15674**
Gain ratio		**8**	**15.7**
Exergy input, MW		10.6	8
Exergy out, MW		0.16	0.18
Exergy loss, MW		4	3.7
Exergy destruction, MW		6.4	4.2
Exergy Efficiency		1.51	2.2
Total input cost	Chemical cost, $/hr	22	12
	Running cost, $/hr	433	274
	Capital cost, $/hr	121	172
Unit product cost, $/m^3$		**2.66**	**2.15**

* Specified parameters

CONCLUSION

The proposed MSF-TVC system is analyzed and investigated under different operating conditions by using the thermoeconomic methodology. The optimum operating conditions of the proposed system is achieved at 6 kPa entrained pressure, top brine temperature110ºC, splitter ratio equal to 0.28, and $T_{b,out,recovery}$ = 45.5ºC, 20 stages of heat recovery section and one stage of heat rejection section. Under these design conditions the unit product cost is calculated by 2.15 $/m^3. The comparison between the MSF-TVC system and the conventional MSF system showed that the gain ratio of the MSF-TVC system is 96% higher than that of the conventional MSF brine circulation plant. The heat transfer area of the MSF-TVC is 52 % higher than the conventional MSF. The exergetic efficiency of the MSF-TVC system is 46 % higher than that of the MSF system. The unit product cost of the MSF-TVC system is 19 % lower than that of the conventional brine circulation multi stage flash (MSF-BR) system.

NOMENCLATURE

A = cross sectional area, m^2
$\dot{C}$ = cost flow rate, $/hr
c = cost per unit exergy, $/GJ
C_r = Compression ratio of the ejector
$\dot{E}$ = Exergy flow rate, MW
f = Exergoeconomic factor
h = Specific enthalpy, kJ/kg
P = Pressure, kPa
$\dot{m}$ = Mass flow rate, kg/s
N = amortization year
s = Specific entropy , kJ/kg
T = Temperature, K
r = Relative cost difference
v = specific volume, m^3/kg
U = overall heat transfer coefficient , W/m^2.k
$\dot{W}$ = Power, MW
$\dot{Z}$ = Rate of the capital cost
λ = Latent heat
$LMTD$ =Logarithmic meat temperature difference, k
η_{II} = Exergetic efficiency (second law efficacy)

Superscripts:
CI = Capital investment

$_{OM}$ = Operating & Maintenance

Subscripts:
0 = Dead state
D = Destruction
d =Distillate
F = Fuel
L = Loss
p = Product
i = inlet
o = outlet
cw = cooling water

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