WASTE HEAT RECOVERY IN MULTI EFFECT DESALINATION SYSTEM WITH THERMAL COMPRESSOR

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ABSTRACT

The current paper is concerned with the characteristic performance prediction of a system consisting of a Multi-Effect Distillation (MED), and a thermal compressor (steam ejector) connected to the last effect. On expansion of the high-pressure motive steam in the nozzle part of the ejector, it entrains a part of the low- pressure steam produced in the last effect of the MED unit. The pressure of both amounts of steam is then raised in the diffuser to the pressure needed for the heating steam in the first effect. Thus the steam leaving the ejector is used as heating steam in the first effect. Mass flow rates of brine, vapor, condensate and distillate in each component of the studied system have been calculated by aid of mass conservation and heat balance equations. Based on this analysis the performance of the studied system has been predicted for different arrangements and operating conditions. The results obtained show that, coupling a steam ejector with a MED unit always improve the performance of the desalination process. For the same motive steam pressure, the performance ratio of a MED unit declines as the number of effects increases. Therefore, it is economically preferable to choose low number of effects.

Keywords: *Waste heat, recovery, multi effect desalination, thermal compressor, steam ejector.*

الملخص

يهدف هذا البحث إلي استنباط أداء منظومة اعذاب المياه المالحة ، والتي تتكون من وحدة اعذاب متعددة مراحل التقطير ، وضاغط حراري (قاذف بخار) ، يوصل بالمرحلة الأخيرة لوحدة الاعذاب . فعند تمدد بخار الماء ذو الضغط العالي في فوهة القاذف يتم سحب جزء من بخار الماء ذو الضغط المنخفض ، والمتكون في المرحلة الأخيرة ، وتختلط كميتي البخار مع بعضهم البعض ، ليستخدمان كبخار تسخين في المرحلة الأولي ، وذلك بعد رفع ضغطهما في ناشر القاذف . توصل هذا البحث إلي استنباط معادلات حفظ الكتلة والاتزان الحراري لكل جزء من أجزاء المنظومة المعنية وقد تم حلهم لحساب معدلات تدفق البخار والماء المالح والمتكثف وناتج التقطير. ومن ثم تم استنباط أداء الأنظومة المعنية لتركيبات وظروف تشغيل مختلفة . أظهرت النتائج تحسن دائم في أداء وحدة الاعذاب متعددة المراحل وذلك عند استخدام قاذف بخار . ويستنتج من هذا البحث انه للحصول علي معامل أداء كبير ، يفضل استخدام وحدة اعذاب ذو عدد قليل من المراحل .

NOMENCLATURE

c_w	specific heat of water (kJ/kg K)
h	specific enthalpy (kJ/kg)
L	latent heat (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
N	number of effects
PR	performance ratio
T	vapor temperature (^oC)
t	brine temperature (^oC)
ε	temperature elevation of salt water boiling
η	efficiency

SUBSCIPTS

b	brine
c	condensate
comp	compression
cond	condenser
d	distillate
e	evaporation
entr	entrained
f	feed salt water
fl	flashing
h	heating vapor
ms	motive steam
noz	nozzle
t	total
1,2,…, n	effect number

1. INTRODUCTION

Many areas of the world are severely lack of water supply. Most of these areas are located close to salty water resources, which are unsuited for direct consumption and most other applications. The transport of water from regions with sufficient water is rather expensive. Desalination is a means by which salty water can be utilized for direct consumption. According to [Marinos and Assimacopoulos and Provatas, 1991] and [Morris and Hanbury, 1991], the installed capacity of desalinated water systems in 1990 reached 13 Million m^3/day, which is expected to further increase due to the continuous growth of population number, and the need for more fresh water. The dramatic increase in desalinated water supply will create a series of problems, the most significant of which are those related to energy consumption. It has been estimated that a production of 13 million m^3 of potable water per day requires 130 million tons of oil per year [Kalogirou, 1998]. Given the current understanding of the greenhouse effect and the importance of CO_2 levels, the use of oil in such quantity is debatable. Thus, pollution would be a major concern. Consequently, any improvement in the efficiency of the available methods of desalination would result in reducing both the quantity of energy needed and the inherent pollutants.

The two methods most commonly used to desalt water are Multi-Effect Desalination (MED) and Multi Stage Flash Desalination (MSFD). For small and medium applications, MED system is cheaper than MSFD, and also requires the simplest sea-water treatment [Marion, 1993]. Therefore MED is superior to MSFD for such applications. Recent researches [Takada and Drake, 1983],[Abdel-Aal, and Al Naafa, 1993],[Rifert,and Sadrak, and Trkoz and Podberensy, 1991],[Michaels, 1992],[Greco, and Murat, 1991]and [Zarza and Schierebeck, 1993] concerning MED have been made to enhance the heat transfer in the effects. These researches led to appreciable decrease in the cost of MED plants. Also, attention has been given to the vacuum system [Zarza and Schierebeck, 1993] and the recovery of waste heat [Zarza, and Gregorzewski, and Gatzka, 1992]and [Zarza and Ajona, and Leon, 1994]. A MED plant was installed at the Plataforma Solar, the Almeria (PSA) in Spain in 1990 [Zarza, 1991] as a test plant, where two possible developments for this plant were investigated [Zarza, 1994]. The first is the design and implementation of a steam ejector-based vacuum system instead of the hydro ejector-based vacuum system. This led to a reduction of the ejector electrical demand by 50% [Zarza , 1994]. The second development is to incorporate a double effect absorption heat pump to the MED plant of Almeria [Zarza , 1994] to recover the heat rejected in the condenser, at low temperature, by elevating its temperature. This task reduced the plant thermal demand by 45% [Zarza , 1994] and [Garcia-Rodriguez and Gómes-Camacho., 1999]. The use of heat pump to recover the waste heat is complicated and expensive.

Another possibility to make use of the waste heat in the condenser of a MEDP is to couple a thermal compressor (steam ejector) to the last effect. On expansion of the high-pressure motive steam in the nozzle of the ejector, it entrains a part of the low-pressure steam produced in the last effect of the MED unit. The pressure of both amounts of vapor is then raised in the diffuser to the pressure of the heating steam in the first effect. Thus the vapor leaving the ejector is used with fresh vapor for heating the brine in the first effect evaporator. This idea was partially implemented in the test plant of Almeria. However, nothing has been reported about the effect of such coupling. Therefore, the current paper is concerned with the prediction of characteristic performance of a system consisting of a MED unit, and a steam ejector connected to the last effect. In the present paper the water vapor coming out of the ejector is taken to be the heating vapor in the first effect without adding any fresh vapor, on contrast to the trial made in Almeria [Zarza , 1994]and [Garcia-Rodriguez and Gómes-Camacho., 1999].

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2. DESCRIPTION OF THE STUDIED SYSTEM

The system studied in the present paper is shown schematically in Figure 1. It is composed of a MED unit and a thermal compressor (steam ejector) connected to the last effect. The MED unit has n Effects (Cells) numbered from 1 to n. Each effect (b) consists of an evaporator (c) and a pre-heater (d). The salt feed water is preheated from effect to effect in the pre-heaters (d) by condensing some of the vapor produced in each evaporator (c). The evaporators of the desalination plant use sprayed tube bundles for salt water evaporation. The n evaporators of the plant are operated at successively decreasing temperatures and pressures from effect (1). The evaporator tube bundle of the effect (1) is supplied by heating vapor rate $\dot{m}_{h,1}$. It condenses by giving heat to the sprayed salt feed water, where a part of the latest evaporates. The vapor thus produced goes on to effect (2), where it is also condensed in a tube bundle sprayed with feed water. The latent heat released by condensation of the vapor allows part of the feed water entering the effect (2) to evaporate at lower temperature and pressure. The same condensation/evaporation process is repeated in effects (3) to (n).

The thermal compressor (steam ejector) (a) is supplied with high-pressure saturated steam of rate $\dot{m}_{ms}$ as motive steam. On expansion of the motive steam in the nozzle of the ejector, it entrains a rate $\dot{m}_{entr}$ of the low-pressure vapor produced in the effect (n) of the MED unit. The pressure of both amounts of vapor; i.e. $\dot{m}_{ms} + \dot{m}_{entr}$ is then raised in the diffuser to the pressure needed for the heating vapor in the first effect. Thus, the vapor leaving the ejector is used as heating steam with rate $\dot{m}_{h,1}$ in the first effect. On contrast to the trial made in the test plant of Almeria [14,15], no fresh vapor is added to the vapor coming out of the ejector. This implies that: $\dot{m}_{h,1} = \dot{m}_{ms} + \dot{m}_{entr}$. The feed salt water with rate $\dot{m}_f$ as well as the outlet brine from the evaporator of the effect (n) pass through a heat exchanger (e). In this heat exchanger, the feed water temperature rises as a result of heat transfer from the outlet brine. $\dot{m}_f$ is then passed through the condenser (f), where the vapor exiting the pre-heater of the last effect is condensed and the feed water temperature is further raised.

3. ANALYSIS Of THE SYSTEM

3.1 Effects of the MED unit

For facilitating the analysis, the following assumptions are made:

1. The difference of vapor temperature between any two successive effects is equal. This implies that:

$$T_{h,1} - T_{v,1} = T_{v,1} - T_{v,2} = \dots = T_{v,n-1} - T_{v,n} = \Delta T \tag{1}$$

   This assumption has been based on the ideal design conditions, which result in uniform coefficient of heat transfer in all effects.

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2. The elevation ε of salt water boiling temperature is equal for all effects.

3. The increase in feed water temperature Δt in each pre-heater is equal to the decrease in vapor temperature ΔT from one effect to the successive one.

4. The specific heat of the salt water is constant and independent of the temperature and salt concentration.

The mass flow rates of vapor, brine, condensate and distillate in the effects (1), (2), and (n) are shown in Figure 2 (a, b, and c, respectively). Referring to fig. 2-a, water vapor with pressure $p_{h,1}$ and enthalpy $h_{h,1,i}$ is supplied with a rate, $\dot{m}_{h,1} = \dot{m}_{ms} + \dot{m}_{entr}$, to the evaporator of the effect (1) for heating the incoming feed water. $\dot{m}_{h,1}$ is condensed in the evaporator of the effect (1) and exits as saturated liquid (i.e. with enthalpy $h_{h,1,f}$). This causes an increase in the feed water temperature from t_1 to the boiling temperature T_1 of the feed water and produces an evaporation rate $\dot{m}_{e,1}$. Hence, a heat balance for the evaporator of the effect (1) is given by:

$$\dot{m}_{h,1} (h_{h,1,i} - h_{h,1,f}) = \dot{m}_f \, c_w (T_1 - t_1) + \dot{m}_{e,1} L_1 \tag{2}$$

From which it follows that:

$$\dot{m}_{e,1} = \frac{\dot{m}_{h,1} (h_{h,1,i} - h_{h,1,f}) - \dot{m}_f \, c_w (T_1 - t_1)}{L_1} \tag{3}$$

The temperature of the produced vapor in the effect (1) is $T_{v,1}$, which is related to the boiling temperature T_1 of the feed water by:

$$T_{v,1} = T_1 - \varepsilon \tag{4}$$

The vapor mass rate $\dot{m}_{e,1}$ passes through the pre-heater of the effect (1) where a rate $\dot{m}_{c,1}$ of it is condensed due to cooling by the feeding water whose temperature rises from t_2 to t_1. Consequently, the heat balance equation for the pre-heater of the effect (1) is given as:

$$\dot{m}_{c,1} L_1 = \dot{m}_f \, c_w \, \Delta T \tag{5}$$

From which $\dot{m}_{c,1}$ is obtained by:

$$\dot{m}_{c,1} = \frac{\dot{m}_f \, c_w \, \Delta T}{L_1} \tag{6}$$

The vapor mass flow rate $\dot{m}_{v,1}$ at the pre-heater exit of the effect (l) is then given by:

$$\dot{m}_{v,1} = \dot{m}_{e,1} - \dot{m}_{c,1} \tag{7}$$

The brine mass flow rate $\dot{m}_{b,1}$ leaving the effect (1) evaporator is calculated from:

$$\dot{m}_{b,1} = \dot{m}_f - \dot{m}_{e,1} \tag{8}$$

The heating vapor rate $\dot{m}_{h,1}$ comprises a portion $\dot{m}_{entr}$ coming up out of the evaporator of the last effect. Therefore the distillate rate of the effect (1) is given as:

$$\dot{m}_{d,1} = \dot{m}_{entr} \tag{9}$$

Considering the second effect (fig. 2-b), the brine rate $\dot{m}_{b,1}$ outlet from the effect (1) enters the evaporator of the effect (2) through a U tube, where its pressure is reduced from p_1 to p_2. This results in production of vapor with a rate $\dot{m}_{fl,2}$ by flashing. Accordingly, $\dot{m}_{fl,2}$ can be determined from:

$$\dot{m}_{fl,2}\ L_2 = \dot{m}_{b,1}\ c_w\ \Delta T \tag{10}$$

Rearranging Eq. (10), $\dot{m}_{fl,2}$ is calculated by:

$$\dot{m}_{fl,2} = \frac{\dot{m}_{b,1}\ c_w\ \Delta T}{L_2} \tag{11}$$

The heating vapor rate $\dot{m}_{h,2}$ entering the evaporator of the effect (2) is equal to $\dot{m}_{v,1}$ coming from the pre-heater of the effect (1). $\dot{m}_{h,2}$ condenses in the evaporator of the effect (2), and gives its latent heat $\dot{m}_{h,2}\ L_{h,2}$ to the brine where a vapor rate $\dot{m}_{e,2}$ is produced by evaporation. $\dot{m}_{e,2}$ is determined in an analogous way to $\dot{m}_{e,1}$, Eq. (2), from:

$$\dot{m}_{e,2} = \frac{\dot{m}_{h,2}\ L_1}{L_2} \tag{12}$$

The outlet distillate $\dot{m}_{d,,2}$ from the effect (2) evaporator is obtained by:

$$\dot{m}_{d,2} = \dot{m}_{e,1} \tag{13}$$

The produced vapor mass rates $\dot{m}_{e,2}$ and $\dot{m}_{fl,2}$ pass through the pre-heater of the effect (2), where a mass rate $\dot{m}_{c,2}$ from them condenses by transferring its latent heat $\dot{m}_{c,2}\ L_2$ to the feeding water. Consequently, the temperature of the feeding water rises from t_3 to t_2. $\dot{m}_{c,2}$ is calculated similarly to $\dot{m}_{c,1}$, Eq. (6), by:

$$\dot{m}_{c,2} = \frac{\dot{m}_f\ c_w\ \Delta T}{L_2} \tag{14}$$

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The outlet vapor rate $\dot{m}_{v,2}$ from the pre-heater of the effect (2) is obtained from:

$$\dot{m}_{v,2} = \dot{m}_{e,2} + \dot{m}_{fl,2} - \dot{m}_{c,2} \tag{15}$$

The brine mass flow rate $\dot{m}_{b,2}$ at the exit of the effect (2) evaporator is given then by:

$$\dot{m}_{b,2} = \dot{m}_{b,1} - \dot{m}_{e,2} - \dot{m}_{fl,2} \tag{16}$$

Determination of mass flow rates in an effect (i), lying between (3) and (n-1), is just as similar as for effect (2). In doing so, the subscripts (1) and (2) in Eqs. (10) to (16) are replaced by (i-1) and (i), respectively. Also, the same analysis of the effect (2) can be conducted for the effect (n). However, the ejector entrains a vapor mass rate $\dot{m}_{entr}$ out of the evaporator of this effect. Accordingly the vapor mass rate $\dot{m}_{v,n}$ exiting the pre-heater of this effect is calculated by:

$$\dot{m}_{v,n} = \dot{m}_{e,n} + \dot{m}_{fl,n} - \dot{m}_{entr} - \dot{m}_{c,n} \tag{17}$$

$\dot{m}_{v,n}$ along with $\dot{m}_{c,n}$ pass through the condenser, where the distillate rate exiting the condenser is given by:

$$\dot{m}_{d,cond} = \dot{m}_{e,n} + \dot{m}_{fl,n} - \dot{m}_{entr} \tag{18}$$

The total distillate rate $\dot{m}_{d,t}$ produced is then calculated as:

$$\dot{m}_{d,t} = \dot{m}_{d,1} + \dot{m}_{d,2} + \dot{m}_{d,3} + ... + \dot{m}_{d,n} + \dot{m}_{cond} \tag{19}$$

3.2 Steam Ejector

A section through a steam ejector is shown schematically in Figure 3. The pressure of steam and water vapor are shown in Figure 4 corresponding to the stations marked in fig. 3. Figure 5 represents schematically the T-s diagram for the compression process in the steam ejector. High pressure saturated steam (motive steam) at condition 1 expands in a convergent divergent nozzle and the exit condition is at state 2 where the pressure is equal to the saturated pressure of the water vapor in the last effect of the desalination unit and velocity is supersonic. It is to be noticed in this figure that both the shock wave compression and diffuser compression is assumed to be replaced by simple compression and the efficiency of compression is modified to take into account this assumption. Thus, it follows for nozzle efficiency, and compression efficiency that:

$$\eta_{noz} = \frac{h_1 - h_2}{h_1 - h_{2'}} \tag{20}$$

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$$\eta_{comp} = \frac{h_{6'} - h_4}{h_6 - h_4} \tag{21}$$

The steam coming out of the nozzle is at high velocity. The water vapor coming out of the evaporator of the last effect is at low velocity. Giving the required momentum to the last effect vapor is called entrainment of vapor. Thus the motive steam will lose some energy. This entrainment process is very inefficient. This is accounted for in introducing a term called entrainment efficiency, which may be given by:

$$\eta_{entr} = \frac{h_1 - h_{2''}}{h_1 - h_2} \tag{22}$$

where 2'' is the state of steam just before mixing with vapor of the last effect.

Applying the law of conservation of energy to the ejector, thus it results in:

$$\dot{m}_{ms} (h_1 - h_{2''}) = (\dot{m}_{ms} + \dot{m}_{entr}) (h_6 - h_4) \tag{23}$$

where $\dot{m}_{ms}$ and $\dot{m}_{entr}$ are mass flow rate of motive steam and flow rate of entrained vapor from the last effect of the desalination unit, respectively.

From Eqs. (20) through (23) the ratio of mass rates of motive steam to entrained vapor can be given by:

$$\frac{\dot{m}_{ms}}{\dot{m}_{entr}} = \frac{(h_{6'} - h_4)}{\eta_{comp}\, \eta_{entr}\, \eta_{nozz} (h_1 - h_{2'}) - (h_{6'} - h_4)} \tag{24}$$

For energy balance at entry to the mixing section, it follows that:

$$\dot{m}_{ms}\, h_{2''} + \dot{m}_{entr}\, h_3 = (\dot{m}_{ms} + \dot{m}_{entr})\, h_4 \tag{25}$$

4. SOLUTION PROCEDURE

To study the effect of connecting a thermal compressor (steam ejector) with a MED unit on the performance of desalination process, Eqs. (1-25) have to be solved to get the rates of vapor, condensate, distillate, and brine. For this purpose, it is necessary first to assign the design parameters of the system, and the inlet conditions of the feed salt water and the motive steam of the steam ejector. The solution is initiated by assuming a value for the ratio $\dot{m}_{ms}/\dot{m}_f$. Based on this value the ratio $\dot{m}_{ms}/\dot{m}_{entr}$ is determined by solving Eqs. (24) and (25), iteratively to get $\dot{m}_{ms}/\dot{m}_f$ and $\dot{m}_{entr}/\dot{m}_f$. It is to be noticed here that, unless the condition $h_1 - h_{2''} > h_6 - h_4$ is fulfilled, the ejector fails to entrain any vapor from the last effect for the given conditions of the motive steam.

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Having determined the ratio $\dot{m}_{ms}/\dot{m}_{entr}$, the ratio $\dot{m}_{h,1}/\dot{m}_f$, i.e. the rate of heating vapor in the first effect per unit mass rate of the feed salt water, can be determined as the sum $(\dot{m}_{ms}+\dot{m}_{entr})/\dot{m}_f$. With the calculated value of $\dot{m}_{h,1}/\dot{m}_f$ all mass flow rates per unit mass of the feed water can be determined by aid of Eqs. (1) through (19). The calculated ratio $\dot{m}_{d,t}/\dot{m}_f$ is then examined. Unless it is equal to a set value, $\dot{m}_{ms}/\dot{m}_f$ is modified and the whole calculation is performed again. This is continuously repeated till the value of $\dot{m}_{ms}/\dot{m}_f$ is found, which results in $\dot{m}_{d,t}/\dot{m}_f$ equal to the set value.

5. RESULTS AND DISCUSSION

The results obtained in the present paper are based on the design data listed in Table 1. They have been taken out of the available practical data for the existing MED plants and steam ejectors. The temperature $T_{v,n}$ of the vapor in the last effect is assumed to be amounted to 33^{o}C, which corresponds to a saturation pressure of 0.05 bar. The boiling temperature elevation ε of the salt water in all effects is 1.5^{o}C. The vapor temperature difference of each two successive effects is 3^{o}C. The temperature t_f of the feed salt water to the last effect, i.e. after passing through the heat exchanger and the condenser is less than the vapor temperature in the last effect by 2 ΔT; i.e. t_f=27^{o}C. The ratio $\dot{m}_{d,t}/\dot{m}_f$ of the total distillate rate and the feed salt water rate is 0.4. The efficiency of the nozzle, entrainment and compression is 0.85, 0.65, and 0.65, respectively. In addition to the data given in Table 1, it is necessary to know the pressure of the heating vapor in the first effect. This pressure is dependent on the number of effects N. It is determined as the saturation pressure of water vapor having a saturation temperature of $T_{v,n}$+(n+1) ΔT.

Results have been obtained for the studied system in the range of number of effects n=1-16, and pressure ratio of motive steam and condenser (i.e. last effect) p_{ms}/p_{cond} =1-1000. Samples of these results are presented in the following. The mass flow rates of vapor, brine, condensate, and distillate are listed in Tables 2, 3, and 4 for a 4 effects unit without thermal compressor, with thermal compressor (p_{ms}/p_{cond}=50) and (p_{ms}/p_{cond}=250), respectively. These Tables show clearly that the mass rate of heating vapor in the first effect differs in accordance with the motive steam pressure p_{ms}. However, the heat rate transferred to the brine in this effect is independent on p_{ms}. Therefore, all other mass rates are independent of p_{ms} except the distillate rate in the first effect as well as in the condenser.

It has been found for all cases studied in the present paper that the use of thermal compressor results in making the heating vapor in the first effect superheated. Figure 6 shows the degree of superheat ΔT_{sup} of the heating vapor in the first effect as a function of the pressure ratio p_{ms}/p_{cond} and for N=1-16. It is to be noticed that, each curve of this figure starts from a certain value of $p_{ms}/p_{cond,}$ depending on n. The start point of each curve represents the minimum pressure ratio p_{ms}/p_{cond} at which the ejector begins to entrain vapor out of the last effect.

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For stable heat transfer, it is preferable to have saturated water vapor as heating vapor. Therefore, water can be injected into the heating vapor of the first effect. In this case the mass rate of heating vapor in the first effect would be equal to that of the MED unit without thermal compressor.

In Figure 7 the mass rate ratio $\dot{m}_{entr}/\dot{m}_{ms}$ is plotted versus the pressure ratio p_{ms}/p_{cond} for n=1-16. It is clear from fig. 7 that the entrained vapor rate $\dot{m}_{entr}$ increases with growing motive steam pressure p_{ms}. The rate of increase is relatively great for n=1 and it declines as N is raised. This is attributed to the increase in heating vapor pressure in the first effect.

The mass ratio $\dot{m}_{entr}/\dot{m}_{e,n}$ is plotted in Figure 8 versus the pressure ratio p_{ms}/p_{cond}. It is evident from this figure that, for n=1, the entrained portion of the produced vapor rate in the last effect, increases very rapidly with rising ratio p_{ms}/p_{cond} till $p_{ms}/p_{cond} \approx 50$, then the increase becomes very slow. As n is raised the curve becomes flatter. This indicates that, for small n the effect of the motive steam pressure is pronounced in a small range in which tangible and rapid improvement in desalination process occurs as p_{ms} in increased. This effect is receded with increasing N.

For expressing the goodness of the studied system, the performance ratio PR is used. PR gives the number of kgs of distillate produced by 2300 kJ of heat input. Accordingly, PR is calculated by:

$$PR = \frac{2300\ \dot{m}_{d,t}}{\dot{m}_{ms}\ (h_1 - h_{h,1,f})} \tag{26}$$

The calculated performance ratio by aid of Eq. (26) for the studied system is plotted in Figure 9 versus the pressure ratio p_{ms}/p_{cond} as solid lines for n=1-16. For the purpose of comparison, PR is also drawn in fig. 9 as dashed lines for MED units without thermal compressors and for the same number of effects. This figure reveals clearly that PR is always higher in case with thermal compressor. The difference of PR between a MED unit with and without thermal compressor increases in all cases as p_{ms}/p_{cond} is raised . This difference is clearly very high at small N; i.e. at small n, the effect of using thermal compressor is very pronounced. In general, one can obviously see that at small number of effects, PR is higher than that at great n and the same p_{ms}/p_{cond}. Therefore, it is recommended to use the MED unit with small number of effects so that better PR is achieved and a lot of construction materials are saved. Meanwhile, the system becomes simpler in construction and operation.

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6. CONCLUSIONS

The results obtained in the present paper offer the following conclusions:

1. The capacity of the steam ejector to entrain water vapor from the last effect of a multi-effect desalination (MED) unit increases as the number of effects is reduced. This result is enhancing the effect of the thermal compressor on the performance of a MED unit at low number of effects.
2. The performance ratio of a MED unit to which a thermal compressor (steam ejector) is connected is always higher than that of the MED unit without steam ejector for the same number of effects.
3. On deciding to use a steam ejector with a MED unit, it is economically preferable to choose low number of effects to reach high value of performance ratio, and to have simpler system. Meanwhile, a great amount of construction materials can be saved and the cost of the desalinated water is reduced.

REFERENCES

1. Abdel-Aal, H.K. and Al Naafa, M.A., 1993, "Enhanced Evaporation of Saline Water in Multi-Purpose Solar Desalination Units", Desalination. 93.
2. Garcia-Rodriguez, L., and Gómes-Camacho, C., 1999, "Thermo-economic Analysis of a Solar MultiEffect Desallination Plant Installed at Plataforma Solar de Almeria (Spain)", Desalination 122.
3. Greco, N. and Murat, F., 1991, "Application of the ME Process at Low Temperature to a Large Sea Water Desalination Plant", IDA Conference, Washington D.C..
4. Kalogirou, S., 1998, "Use of Parabolic trough Solar Collectors for sea-water desalination", Applied Energy 60.
5. Marinos D., Assimacopoulos D., and Provatas F., 1991, "The experience of local authorities with water desalination in the Islands of Southern Europe", Proceedings of the New Technologies for the Use of Renewable Energy Sources in Water Desalination, Athens, Greece, Section V.
6. Marion, OJ., 1993, "Design and Operation Comparison of MSF and MED Systems", Desalination, 93.
7. Michaels, T.,1992,"Recent Achievement of low Temperature Multiple Effect Desalination in Western Areas of Abu-Dabi, UAE", Desalination 92 Proc., Arabian Gulf Regional Water Desalination Symposium, 15-17 Nov., Al Ain, UAE.
8. Morris R.M., and Hanbury W.T., 1991, "Renewable energy and desalination—a review", Proceedings of the New Technologies for the Use of Renewable Energy Sources in Water Desalination, Athens, Greece, Section 1.
9. Rifert, V.G.; Sadrak, A. I.; Trkoz, Ya.E. and Podberensy, V. A., 1991, "Heat Exchange Intensification at Condensation Inside Tubes of Film Type Evaporators of Desalination lnstallations", Desalination, 81.

10. Takada, M. and Drake, J.C., 1983, " Application of Improved High-Performance Evaporators", Desalination, 45.
11. Zarza, E.; Ajona, J.I. and Leon, J.,1994, "Improvement of a Solar Multi-Effect Desalination System with a Double Effect Absorption Heat Pump", Proceedings of the ASME/JSME/JSES International Solar Energy Conference, San Francisco, CA, USA.
12. Zarza, E. and Schierebeck, A., July 1993, "Acceptance tests and Evaluation of New Vacuum System", Internal Report: R-21/93 EZ-ASch, Plataforma Solar deAlmeria.
13. Zarza, E., August, 1994"Solar Thermal Desalination Project. Phase II Results & Final Project Report", Secretaria Tecnica del CIEMAT.
14. Zarza, E.,1991"Solar Thermal Desalination Project. First Phase Results & Second Phase Description", Secretaria Tecnica del CIEMAT, Madrid.
15. Zarza, E.; Gregorzewski, A.; and Gatzka, M., October 1992"Starting-up and Acceptance Tests of the PACSOL Lithium/Bromide Heat Pump", Internal Report R-29/92 EZ/AG/MG. Plataforma Solar de Almeria.

Table 1: Design parameters for the studied system

MED unit :	
Vapor temperature in the last effect $T_{v,n}$	33°C
Elevation in salt water boiling temperature ε	1.5°C
Vapor temperature difference between two successive effects ΔT	3°C
Temperature of feed salt water at entrance of the pre-heater of the effect (n), $t_f=T_{v,n}-2\,\Delta T$	27°C
Ratio of total distillate rate to feed salt water rate $\dot{m}_{d,t}/\dot{m}_f$	0.4
Steam ejector :	
Nozzle efficiency η_n	0.85
Entrainment efficiency η_{entr}	0.65
Compression efficiency η_c	0.65

Table 2. Mass flow rates for a MED unit without thermal Compressor (n=4)

No. of effect	$\dot{m}_h/\dot{m}_f$	$\dot{m}_{fl}/\dot{m}_f$	$\dot{m}_e/\dot{m}_f$	$\dot{m}_c/\dot{m}_f$	$\dot{m}_d/\dot{m}_f$	$\dot{m}_b/\dot{m}_f$
1	.1099	.0	.1018	.005233	.0	.8982
2	.09653	.004687	.09625	.005218	.1018	.7973
3	.09572	.004148	.09543	.005202	.1009	.6977
4	.09438	.003619	.09410	.005187	.09958	.6000
condenser					.09772	.4000

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Table 3. Mass flow rates for a MED unit with thermal compressor (n=4, p_{ms}/p_{cond}=50)

No. of effect	$\dot{m}_h/\dot{m}_f$	$\dot{m}_{fl}/\dot{m}_f$	$\dot{m}_e/\dot{m}_f$	$\dot{m}_c/\dot{m}_f$	$\dot{m}_d/\dot{m}_f$	$\dot{m}_b/\dot{m}_f$
1	.1078	.0	.1018	.005233	.05943	.8982
2	.09653	.004687	.09625	.005218	.1018	.7973
3	.09572	.004148	.09543	.005202	.1009	.6977
4	.09438	.003619	.09410	.005187	.09958	.6000
condenser					.038287	.4000

Table 4. Mass flow rates for a MED unit with thermal compressor (n=4, p_{ms}/p_{cond}=250)

No. of effect	$\dot{m}_h/\dot{m}_f$	$\dot{m}_{fl}/\dot{m}_f$	$\dot{m}_e/\dot{m}_f$	$\dot{m}_c/\dot{m}_f$	$\dot{m}_d/\dot{m}_f$	$\dot{m}_b/\dot{m}_f$
1	.1077	.0	.1018	.005233	.0757	.8982
2	.09653	.004687	.09625	.005218	.1018	.7973
3	.09572	.004148	.09543	.005202	.1009	.6977
4	.09438	.003619	.09410	.005187	.09958	.6000
condenser					.024147	.4000

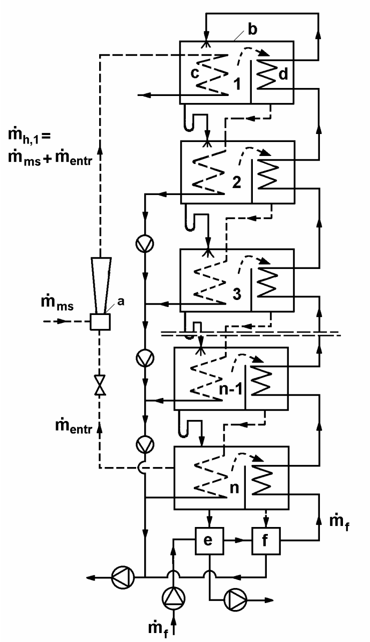


Figure 1. A schematic of the studied system of MED unit with thermal compressor

a thermal compressor (steam ejector) b effect

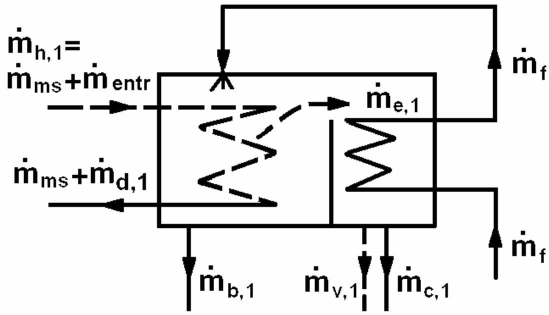
c evaporator d pre-heater

e heat exchanger f condenser

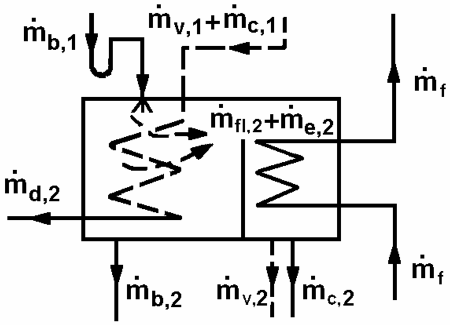
1, 2, …, n effect number

______ brine, distillate, condensate ----- steam, water vapor

a. **the first effect**



b. **the second effect**



c. **the last effect**

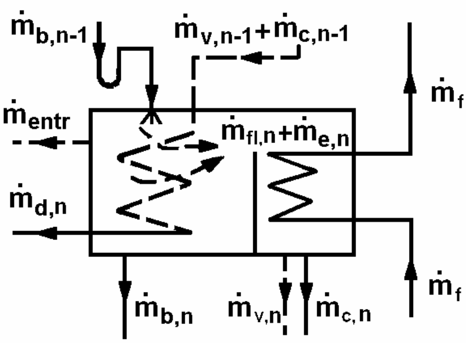


Figure 2. Mass flow rates of brine, steam, vapor, condensate and distillate in the effects of the studied system

____ brine, distillate ---- steam, water vapor, water vapor + condensate

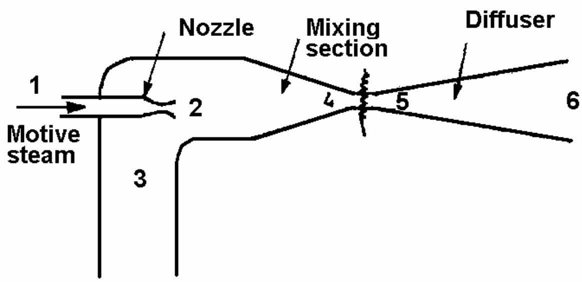


Figure 3. A Section through a steam ejector

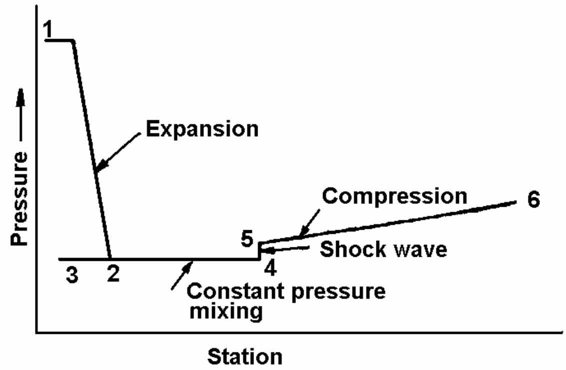


Figure 4. Pressure at the different stations of a steam ejector

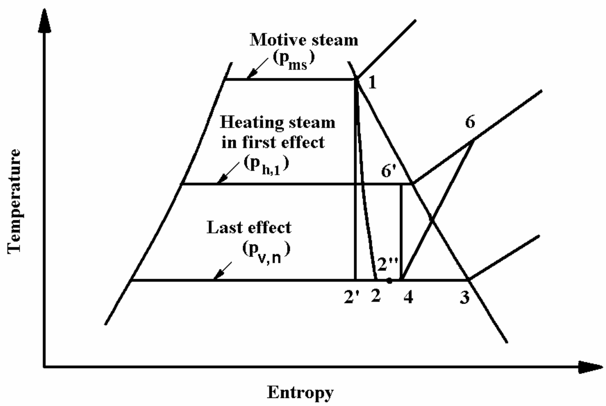


Figure 5. Representation of the steam and water vapor at the different stations of the steam ejector on T-s diagram

Top

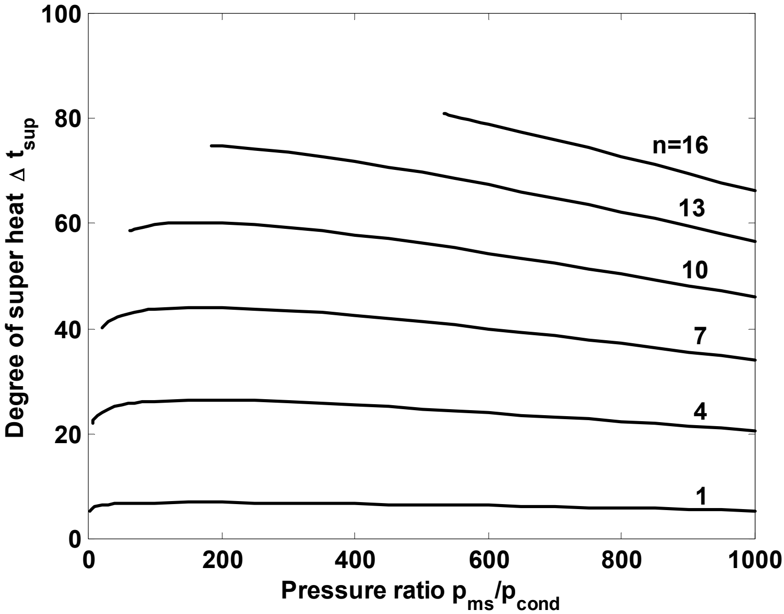


Figure 6. Dependence of superheat degree of the heating vapor in the first effect on the motive steam pressure

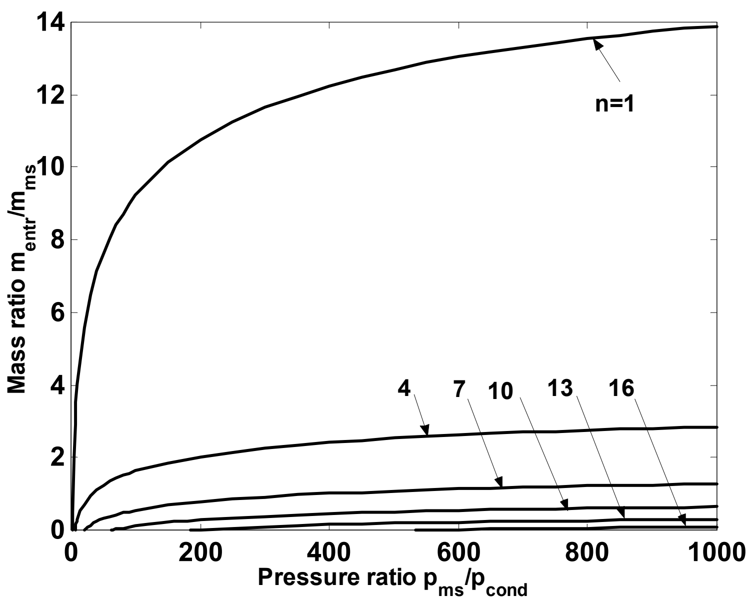


Figure 7. Effect of motive steam pressure on the mass flow ratio of entrained vapor and motive steam

Top

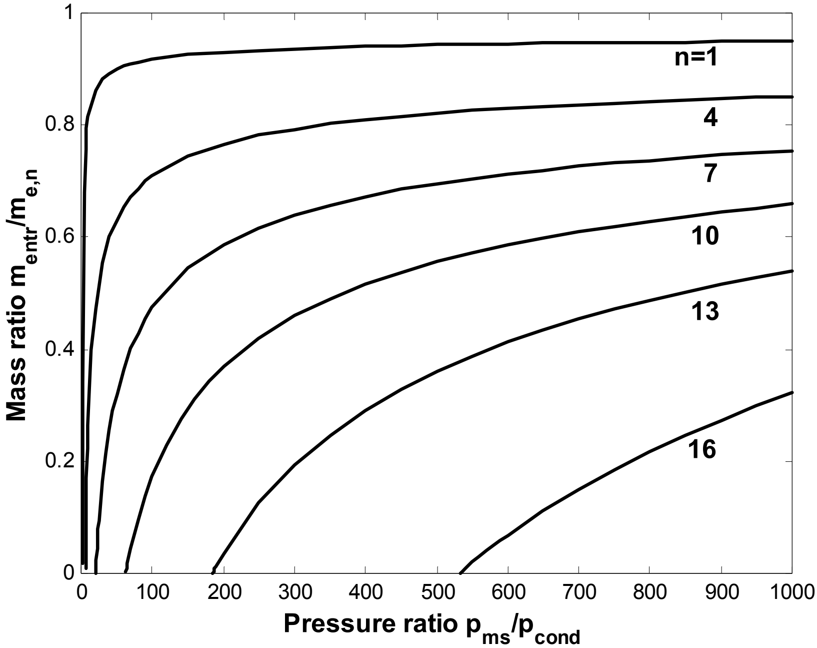


Figure 8. Dependence of the mass flow ratio of entrained vapor and released vapor in the last effect on motive steam pressure

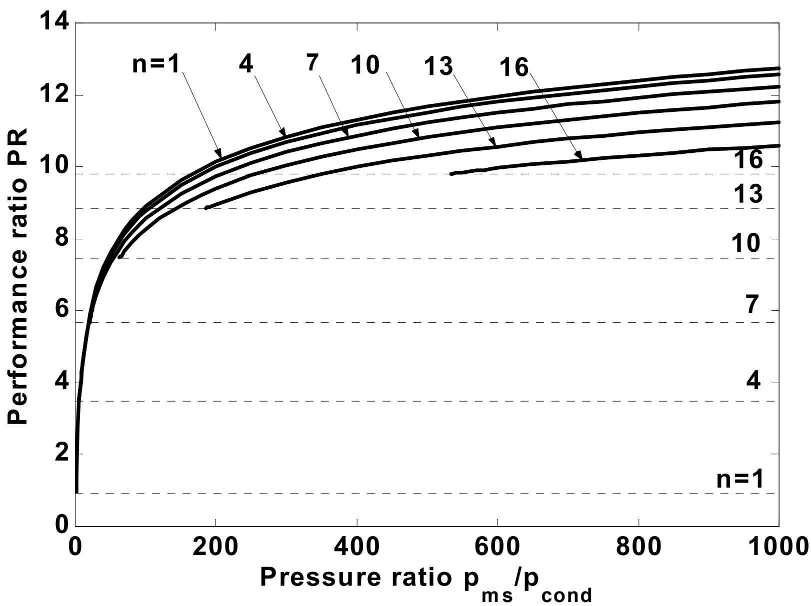


Figure 9. Comparison of performance ratio of a MED unit with and without thermal compressor