Water Technology

Alexandria 1996

تقنية المياة

الاسكندرية

Paper : "Thermal Analysis of Vapor Compression Desalination Plants",
Authors : *S.A. Abdel-Moneim, R.Y. Sakr, E.F.Atwan, M.A. El-Desouky and M.F. Abd Rabbo*

It gives me a great pleasure to inform you that your paper was accepted for presentation and for publication in the proceedings of the International Water Technology Conference.

Prof. Dr. M. Fayek Abd Rabbo

M. F. Abd Rabbo

Organizing Committee
Alexandria, Egypt, Feb. 26-29, 1996.

(IWTC 95)

Water Technology

Water Technology

Water Technology

INTERNATIONAL WATER TECHNOLOGY CONFERENCE

Under the auspices of H.E.

Dr.Abdul-Hady Rady

Minster of Water Resources and Public Works

Forum, Workshop, Exhibition

ALEXANDRIA EGYPT,

February 26-29, 1996

second announcement

THERMAL ANALYSIS OF VAPOR COMPRESSION DESALINATION PLANTS

S.A. Abdel-Moneim R.Y. Sakr, E.F.Atwan, M.A. El-Desouky and M.F. Abd Rabbo
Mech. Eng. Dept. Faculty of Eng. (Shoubra), Zagazig University,
108 Shoubra St., Cairo, Egypt

ABSTRACT

Sea and brackish water desalination can contribute to solve the problem of fresh water shortage in many arid regions in the world. Most of the installed desalination plants employ distillation processes, e.g. multiple effect distillation (MED), multistage flash (MSF), and vapor compression (VC). In this paper, the thermal operated (thermal compression) vapor compression unit (TVC) is analyzed using steam jet ejector for compression. The effect of the thermophysical properties of the sea water on the unit performance is studied. A modified model is presented to relate the properties of the saline water with water salinity and temperature. The salinity is varied from 1% to 12% (from 10000 to 120000 PPM) and the temperature is varied up to 200 °C.

In the present analysis the following assumptions are made:-

1- Heat exchanger and pipes are completely thermally insulated.

2- The analysis is carried out at steady state conditions.

3- The effect of scale formation, foaming, and crystallization of the brine within the investigated range of salinity are neglected.

Nomenclature

B	Brine flow rate
C	Specific heat
F	Feed flow rate
h	Enthalpy
M	Motive steam flow rate
m	Entrained vapor flow rate
P	Product flow rate
p	Pressure
T	Temperature

Greek Letters

ε	Boiling point elevation.
δ_1 , δ_2	Terminal temperature difference.
η	Efficiency

Subscripts

0,1,2,3,4,...	State points
a,b,c,d,...	State points
B	Brine
D	Diffuser
E	Entrainment
F	Feed
N	Nozzle
P	Product
w	Pure water

1-INTRODUCTION

Vapor compression desalination systems are widely used in arid areas, desert, and army camps. Generally in the vapor compression desalination unit, vapor is compressed to higher pressure and its temperature is increased due to the energy input. If the pressure and temperature rise is adequate, the recompressed vapor can be effectively utilized as a heat source for evaporating the same amount of saline water. When vapor is returned to the evaporator tube bundle, it is condensed after giving up its latent heat to evaporate the entrained vapor.

In fact, vapor produced by evaporation from the saline water is superheated because of the boiling elevation. Therefore, they will condense at a lower temperature than the boiling point of the saline water. Also, the latent heats of evaporating and condensing fluids are very nearly equal. Therefore, the energy required to keep the process in operation is only that needed to offset the point elevation on the evaporating side and to provide a small differential to ensure the desired flow of heat. Usual temperature difference of about 4 -5 ^{o}C, Aly [1], is maintained in order to minimize energy consumption and produce high performance ratio.

In the conventional (mechanical operated) VC unit, the latent heat from the evaporating side of the heat transfer surface is transferred to the condensing side of the same surface. The aim is to recover the latent heat from evaporating vapor rather than on rejecting this heat in a water cold condenser. The heat to be rejected from the system is in fact equal to compressor power input. This is far less than the latent heat involved in vapor production. Most of the energy required in this process is for driving the compressor, .

Abdel-Salam, et al, [2], carried out a thermo-economic analysis of a VC system. Three parameters, boiling temperature, temperature difference across the evaporator/condenser unit and the liquid-to-liquid heat exchanger effectiveness, were investigated Also, the system economics were performed through defining basic unit costs of the various system components. The results of this work show that, a saddle point in the global cost function which defines the optimum temperature difference across the evaporator/condenser unit for a given operating parameter. Also, the maximum recovery ratio is limited by the solubility limit of the sulfates scales for a given boiling temperature.

Steam jet compression system is useful when steam (motive steam) is available at temperature and pressure higher than that for the evaporated vapor. The steam jet reduces the pressure, does useful work in compressing the vapor and reduces the chance of scaling at high temperatures. Generally, a steam jet ejector operates at low efficiency of 25 - 30%, Gupta [3]. Also, if the conditions of vapor flow rate and terminal pressure differences are altered, this figure is further reduced. Steam jet ejectors are widely used because they have low initial cost, are easy and quick to install, are stable during operation and have low maintenance.

El-Dessouky, et al [4], performed an analysis to combine a TVC unit with a horizontal falling film (HFF) desalination process. The results show that the values of the thermal performance ratio is higher than the values obtained in the previous conventional combined TVC and HFF desalination process.

2- THERMAL ANALYSIS OF THE PROPOSED VC UNIT

The proposed system is shown in Fig. (1). The system is composed of the following equipment:

1- Evaporating-condensing unit
2- Steam jet ejector
3- Brine-product-feed heat exchanger
4- Motive steam boiler (not analyzed in the present paper)

In the present work the effect of salinity and the temperature level on the system overall performance are investigated. Data from Bromley [5], are utilized to correlate the specific heats for the feed and brine water within a range of salinity (% $1 \leq x \leq 12$%) and a range of temperature from 5 °C to 200 °C. By using the least square method, the following correlation is obtained for the specific heat for the saline water is:

$$C = 0.84 + 3.155\, x^{-0.0542}\, T^{0.014423} \quad \text{kJ/kg .°C} \qquad \text{.................... (1)}$$

2.1 Thermal Analysis of the Steam Jet Ejector

The present analysis is based on the model discussed by Gupta [3] and El-Dessouky [4]. Figure (2.a) shows the schematic diagram of the steam ejector and Fig. (2.b) shows the h-s diagram for the different processes take place through the ejector unit. The motive high pressure steam M flows to the nozzle and emerges at high velocity at the mixing chamber, which is at lower pressure than that of the motive steam, the high velocity steam entrains the vapor coming out from the inlet. The mixture is then compressed to the discharge pressure when flowing in the diffuser.

For M kg of motive steam and m kg of entrained vapor and taking into consideration the inefficiencies associated with the flow of steam and the mixture through the nozzle , diffuser and the mixing process, it follows that:

$$\eta_N = \frac{h_o - h_1}{h_o - h_{1s}} = \frac{0.5V_1^2}{h_o - h_{1s}} \qquad \text{.......................... (2)}$$

where

η_N is the isentropic efficiency of the nozzle and it is taken in the range from 0.7 to 0.9, Kern [6]. The efficiency of the diffuser expressed by η_D;

$$\eta_D = \frac{h_{3s} - h_{2'}}{h_3 - h_{2'}} \qquad \text{.........................} \qquad (3)$$

The diffuser efficiency η_D, is in the range from 0.85 to 1, Kern [6].

Assuming that the mixing process occurs at constant pressure, i.e. $p_2 = p_1 = p_{2'}$, a balance of the momentum in the mixing chamber for M kg of motive steam and m entrained vapor, the entrainment efficiency is given by:

$$\eta_E = \frac{\frac{1}{2}(M+m)V_{2'}^2}{(\frac{1}{2}MV_1^2 + \frac{1}{2}mV_2^2)}$$

Noting that $V_2 << V_1$, so the entrainment efficiency becomes:

$$\eta_E = \frac{\frac{1}{2}(M+m)V_{2'}^2}{\frac{1}{2}MV_1^2} = \frac{\frac{1}{\eta_D}(1+\frac{M}{m})(h_{3s} - h_{2'})}{\frac{M}{m}\eta_N(h_0 - h_{1s})} \qquad \text{....................} \qquad (4)$$

2.2 Thermal Analysis of the Evaporating-Condensing Unit

Mass and heat balance are carried out for the evaporating-condensing unit. Figure (3) shows a schematic diagram for the unit.

Mass balance;

$$F = B + m \qquad \text{.........................} \qquad (5)$$

and

$$P = m \qquad \text{.........................} \qquad (6)$$

Salinity balance;

$$F x_F = B x_B \qquad \text{.........................} \qquad (7)$$

In the present analysis, the total dissolved solids (TDS) of the feed water is taken in the range from 35000 up to 45000 PPM. The maximum permissible (TDS) of the brine is taken such that:

$$x_B / x_F = 2 \qquad \text{.........................} \qquad (8)$$

or

$$B/F = P/F = 0.5 \qquad \text{.......................} \qquad (9)$$

Therefore, the heat balance of the evaporating-condensing unit is:

$$(1+\frac{M}{m})(h_3 - h_4) + \frac{F}{m}C_F T_b = \frac{B}{m}C_B T_c + h_2 \qquad \text{..................} \qquad (10)$$

where, T_c = The saturation temperature + ε

2.3 *Brine-Feed-Product Heat Exchanger Analysis*

The heat balance for the present heat exchanger is carried out and yields;

$$\frac{F}{m}C_F T_a + h_4 + \frac{B}{m}C_B T_c = C_w T_5 + \frac{B}{m}C_B T_d + \frac{F}{m}C_F T_b \qquad \text{.....................} \qquad (11)$$

where,

$$T_5 = T_d + \delta_1 = T_a + \delta_2 \qquad \text{........................} \qquad (12)$$

3- METHOD OF SOLUTION

The present system of non-linear equations, Eqs. (1) to (12) are solved numerically by direct iteration method. In order to carry out the computation the numerical values of h_g, h_f and T must be known in terms of pressure p. So steam table were used to get data for h_g, h_f and T in the range of temperature and pressure used. Curve fitting of the data yields the following:

$$T = (3.3594 + 1.4461 \ln P)^2 \qquad \text{..........................} \qquad (13)$$

$$h_g = (49.863 + 0.40745 \ln P)^2 \qquad \text{..........................} \qquad (14)$$

$$h_f = 4.2076\, T - 1.35 \qquad \text{..........................} \qquad (15)$$

The solution of the Eqs. (1) to (15) results in obtaining the mass flow rate distribution along the system passages and the performance ratio which is defined as the ratio between the mass of the product to the mass flow rate of the motive steam.

$$PR = P/M \qquad \text{..........................} \qquad (16)$$

4- RESULTS AND DISCUSSIONS

The saline water specific heat varied with the salinity and the temperature as shown in Fig. (4). The solid line shows the least square fit of the experimental data of Bromley [5]. Generally the specific heat decreases with the increase of the salinity and slightly affected by the temperature as shown in Fig. (4).

Figure (5) shows the performance of the steam jet ejector at different operating conditions. Also, these results are compared with the data of El-Dessouky [4] and the comparison shows good agreement of the present model.

5- CONCLUSIONS

From the present analysis, it may be concluded that:

1- The specific heat of the saline water is strongly affected by the salinity and slightly affected by the temperature level.

2- The performance ratio of the TVC unit increases with the increase of the motive steam pressure within the present investigated range from 5 up to 25 bar.

3- The performance ratio of the TVC unit decreases with the increase of the evaporator pressure within the present investigated range from 0.5 up to 1.0 bar.

4- The performance ratio of the TVC unit decreases with the increase of the condensing presser within the present investigated range from 1.0 up to 2.0 bar.

5- The performance ratio of the system also affected by the salinity of the feed water. A maximum reduction of in the actual system PR of 10 % is found for the higher range of the salinity (x>40000 PPM), compared with that for the case of pure water specific heat.

5- REFERENCES

1- Aly, S.E., "Energy Savings in Distillation Plants Using Vapor Thermo-Compression", Desalination, vol. 49, 1984.

2- Abdel-Salam, M.S., El-Dib,A.F., Hanafi,A.S. and Mansour, B.M., "Thermo-Economic Analysis of The Vapor-Compression Desalination System", Journal of Eng. and Applied Science, Vol.40, No.4, pp.739-754, Aug.1993.

3- Gupta, C., "Momentum Transfer Operation", Tata McGraw Hill, 1979.

4- El-Dessouky, H.T. and Assassa, G.R., "Computer Simulation of The Horizontal Falling Film Desalination Plant", Desalination, 55, pp.119-138, 1985.

5- Bromley, L.A., "Further Studies of Properties of Sea Water and Its Concentrates to 200 °C", Univ. of Calif., Office of Saline Water, Report No. 522, Grant No. 14-01-0001-763, March 1970.

6- Kern, D.Q., "Process Heat Transfer", McGraw Hill, 1950.

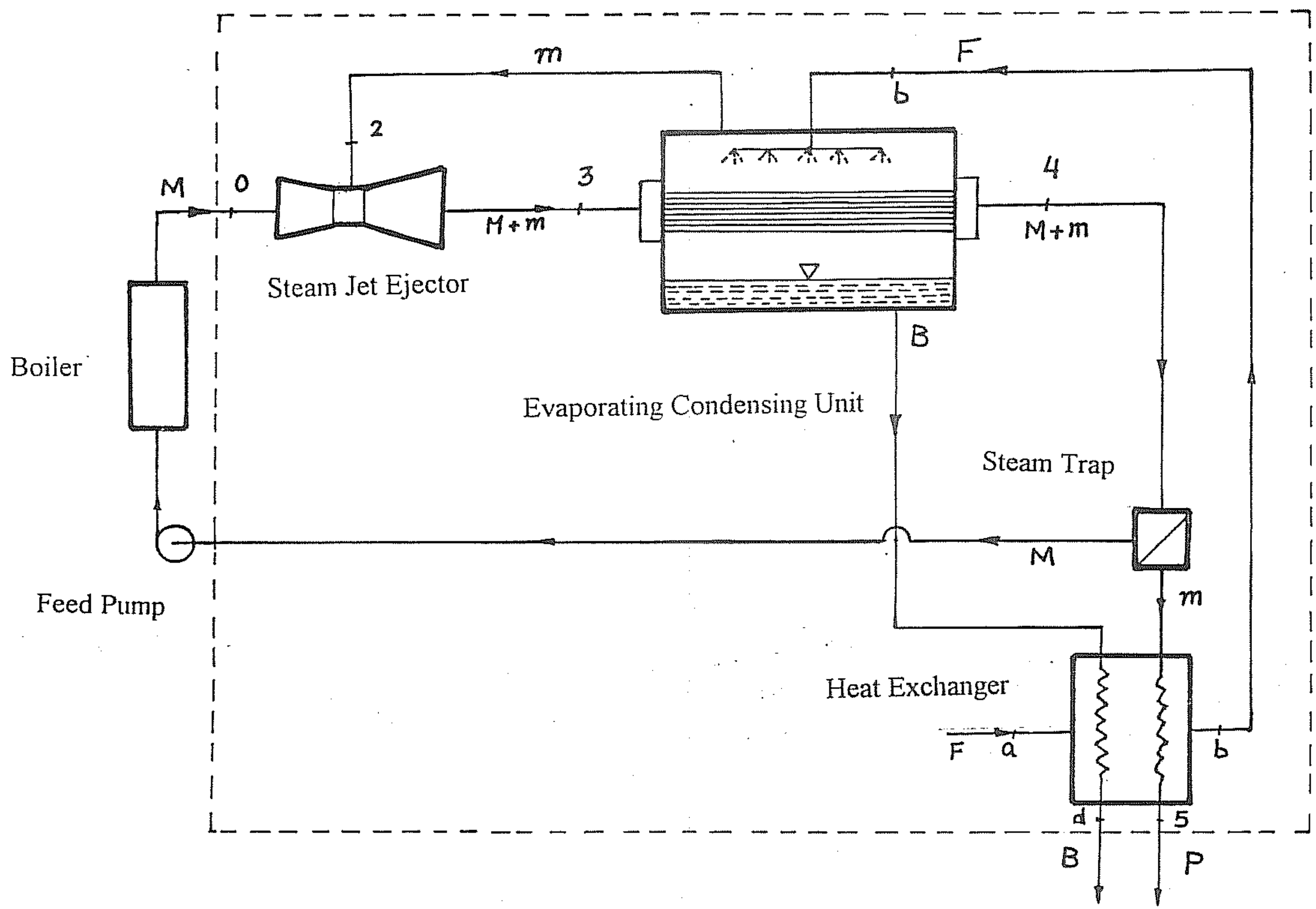

Fig.(1): The Thermal Vapor Compression (TVC) Desalination Unit.

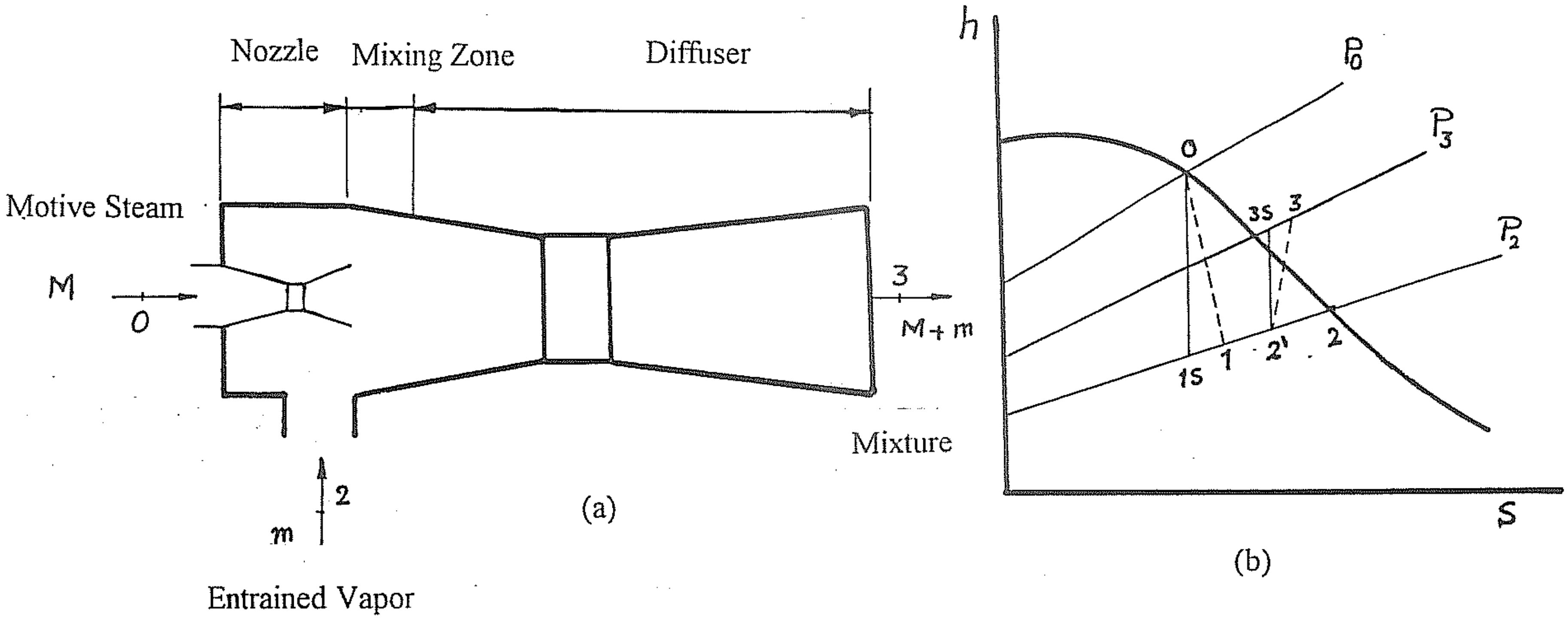

Fig.(2): Schematic and h-s Diagrams for The Steam Jet Ejector.

(a) Schematic diagram (b) h-s diagram

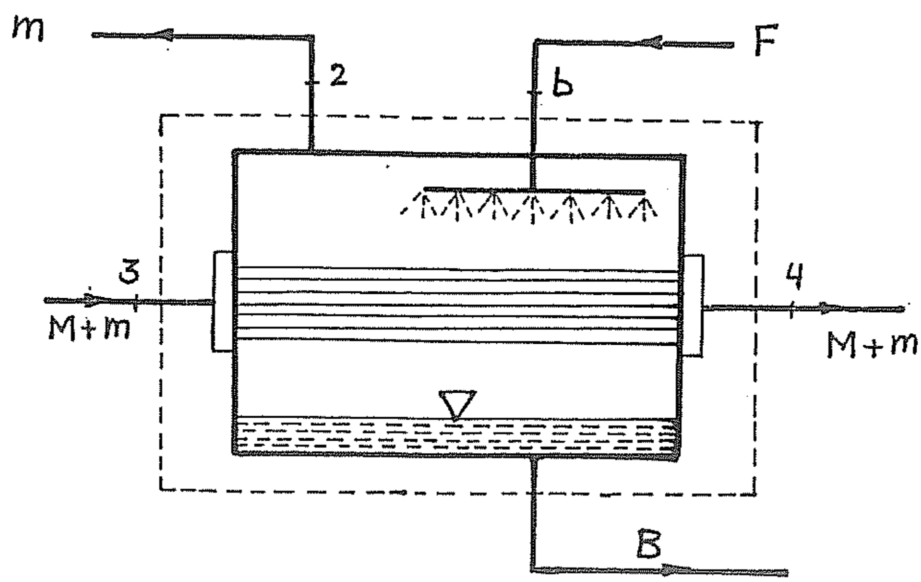

Fig.(3): Schematic Diagram for The Evaporating Condensing Unit.

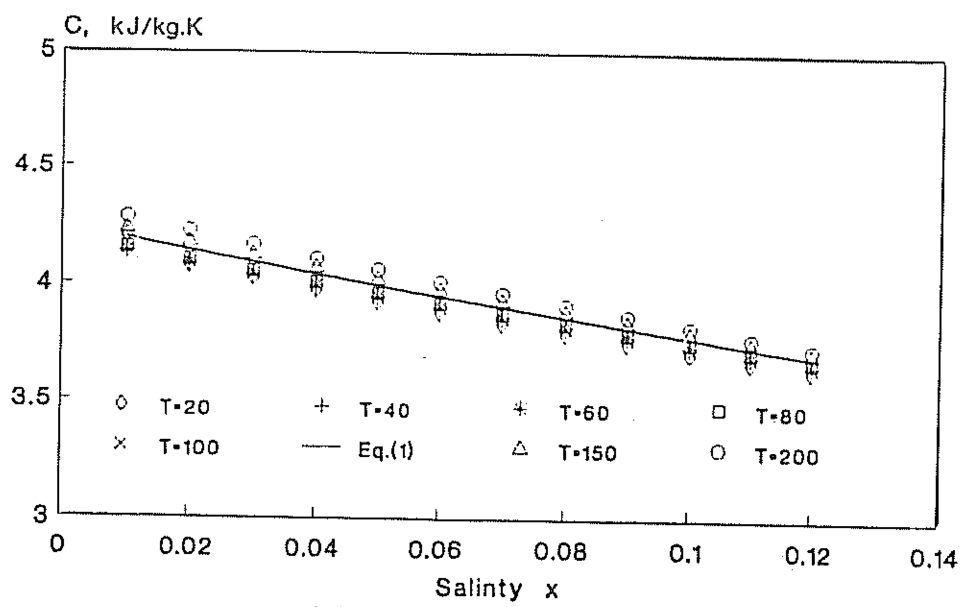

Fig.(4): Saline water specific heat variation with the salinty at different temperatures.

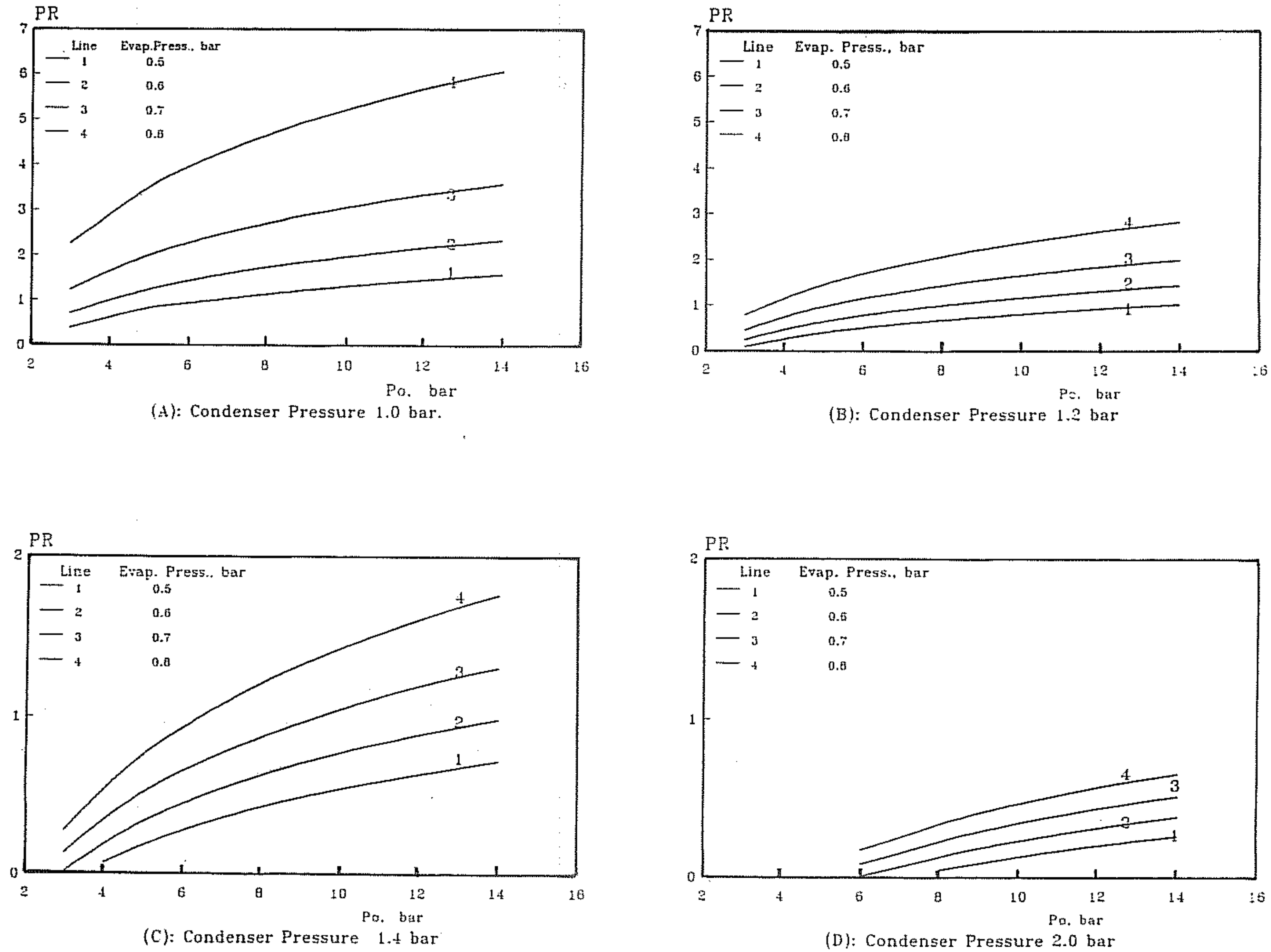

Fig.(5): The Performance Ratio of The TVC Unit
at Different Operating Conditions