

## Performance Analysis of Thermal Vapour Compression Desalination System Coupled to Cogeneration Nuclear Power Plant

P. Asiedu-Boateng, S. Yamoah, F. Ameyaw, S. Dzide and K. Tuffour-Acheampong  
Nuclear Reactors Research Centre, Ghana Atomic Energy Commission, Box LG 80, Legon,  
Accra, Ghana

**Abstract:** Nuclear desalination of seawater has been identified as one of the affordable means of fresh water production. However this can only be achieved by the design of energy efficient desalination systems. This study focused on cycle analysis of the cogeneration nuclear power plant. A theoretical model of the Thermo Vapour Compression (TVC) desalination process was also developed and coupled to the cogeneration nuclear power plant. The modeled coupled system was developed on the computer code, NUCDES using FORTRAN language to investigate the effect of design and operating parameters on parameters controlling the cost of producing fresh water from TVC process. The results showed that the performance of the TVC desalination process and the efficiency of the cogeneration nuclear power improve with the motive steam pressure.

**Key words:** Cogeneration, compression, desalination, nuclear, thermal, vapour

### INTRODUCTION

The world is becoming increasingly aware of critical limitations in the availability of fresh water for agricultural, industrial and domestic uses. Desalination of seawater has been identified a viable approach to solving the problem of water shortages but energy unavailability presents another problem area. Desalination is the production of fresh water from saline water. Therefore, there is the need for research to be conducted into alternative energy sources aside the convectional fossil energy for producing fresh water and nuclear desalination is one such technology that has higher prospects (Kalogiru, 1997). The desalination of seawater using nuclear energy form cogeneration nuclear power plants is a feasible option to meeting the growing demand for potable water. This has been motivated by a wide variety of reasons, ranging from economic competitiveness of nuclear energy to energy supply diversification and to the spin-off effects of nuclear technology in industrial development (Misra and Kuptz, 2004)

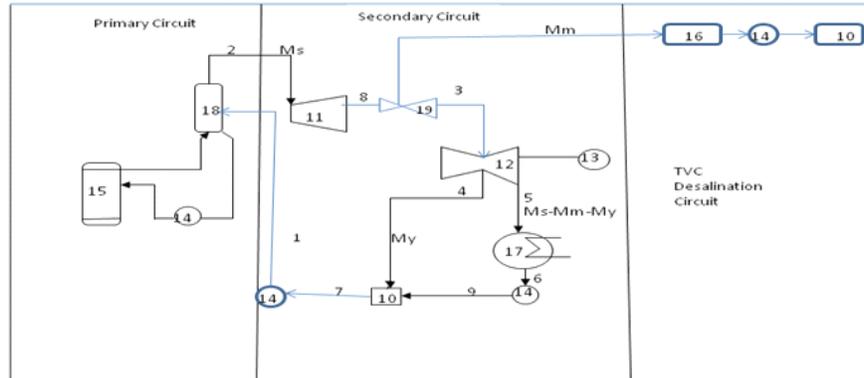
The high cost of energy usually results in high product cost of desalted water and desalination systems must be designed to ensure the most economic use of energy and other resources. Technological efforts must concentrate on reducing the capital, operation, maintenance, and above all, the energy costs. Technological improvements in plant design, heat-transfer technology, corrosion protection, and chemical selection will lead to extended life spans and reduced operation and maintenance costs of desalination units. Consequently, desalination plant must meet high standards of performance, including optimality, cost effectiveness,

reliability, and safety. These can be achieved by using modeling and modern computer-based simulation techniques. Therefore, the need to design and develop efficient and cost effective nuclear desalination technologies with small to medium scale capacities to meet the needs of hard pressed economies of the world that need desalination of saline water remain more paramount (Homig, 1978). This paper focuses on the investigation of the effect of design and operating parameters on parameters controlling the cost of producing fresh water from TVC process and the most efficient operating regimes.

### THEOLOGICAL CONSIDERATIONS

**Coupling desalination plant to nuclear reactor:** Protective barriers must be included in all cogeneration modes to prevent potential carry-over of radioactivity to the product water (United States department of the interior, 2003) The selection criteria of coupling the nuclear power plant to the desalination system are based On reliability, safety multi-barrier isolation, inherent safety and efficiency (Wu and Zheng, 2002) were used.

**Cycle efficiency of cogeneration nuclear power plant:** As shown in Fig. 1, heat generated from the nuclear heating reactor core is conducted to the steam generator to produce steam. The steam generated flows to the high pressure turbine where the required amount of motive steam is bled to the steam jet ejector of the desalination system after expanding in a pressure reducing valve. The remaining steam is expanded isentropically to an intermediate pressure. Some steam is extracted at this



(1) Inlet of steam generator (2) Inlet of high pressure turbine (3) Inlet of low pressure turbine (4) Inlet of mixing chamber (5) Exit of low pressure turbine (6) Exit of condenser (7) Exit of mixing chamber (8) Exit of high pressure turbine (9) Exit of condenser (10) Mixing chamber (11) High Pressure turbine (12) Pass-out Turbine (13) Generator (14) Pump (15) Reactor core (16) Desalination plant (17) Condenser (18) Steam generator (19) Pressure reducing valve

Fig. 1: Parallel co-generation nuclear power plant coupled to desalination plant

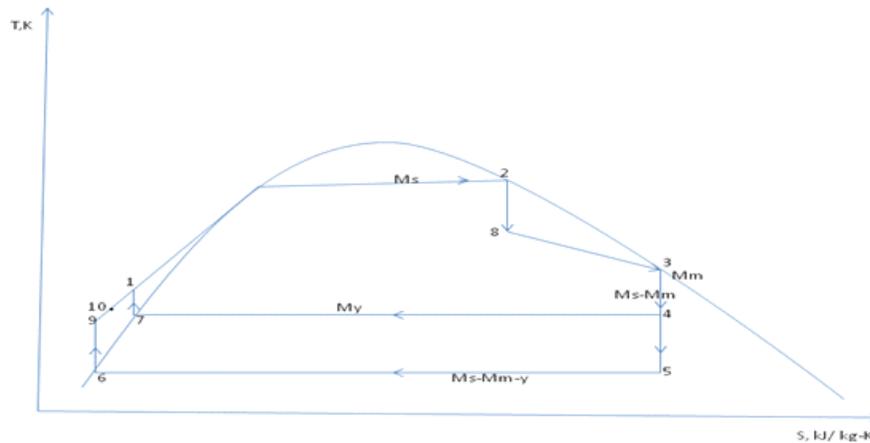


Fig. 2: Temperature-entropy diagram of the Fig. 1

state and routed to the open feed water heater, while the remaining steam continues to expand isentropically to the condenser pressure in the low pressure turbine to generate electricity. The steam leaves the condenser as a saturated liquid. The condensed water and the water from the desalination plant are combined with the water from the mixing chamber (open feed water heater) and returned to the steam generator via an isentropic pump. The fraction of steam extracted is such that the mixture leaves the heater as a saturated liquid at the heater pressure. Figure 2 is an illustration of the temperature -entropy diagram of the cycle of the cogeneration nuclear power plant. The pass-out turbine used enables variation in the power requirement and allows the steam to be expanded to the lowest possible temperature, the remainder being extracted at an intermediate temperature (Saad,1969; Rogers and Mayhew, 1996)

The cycle efficiency of the cogeneration nuclear power plant measures the fraction of the total thermal energy input that is converted into useful work. Neglecting pump work, the cycle efficiency is calculated with the expression given by (Cengel and Boles, 1989).

$$\eta = \text{total turbine work}(W_{TT})/\text{total heat input}(Q_{in}) \quad (1)$$

From Fig. 2 the cycle efficiency is calculated from the Eq. (2):

$$\eta = \frac{W_{TT}}{Q_{in}} = \frac{M_s(h_2 - h_8) + (M_y)(h_3 - h_4) + (M_s - M_y - M_m)(h_4 - h_5)}{M_s(h_2 - h_9)} \quad (2)$$

**TVC process description:** The single effect thermal vapour compression seawater desalination process in its simplest form, is illustrated schematically in Fig. 3. The main components of the unit are the evaporator, the steam

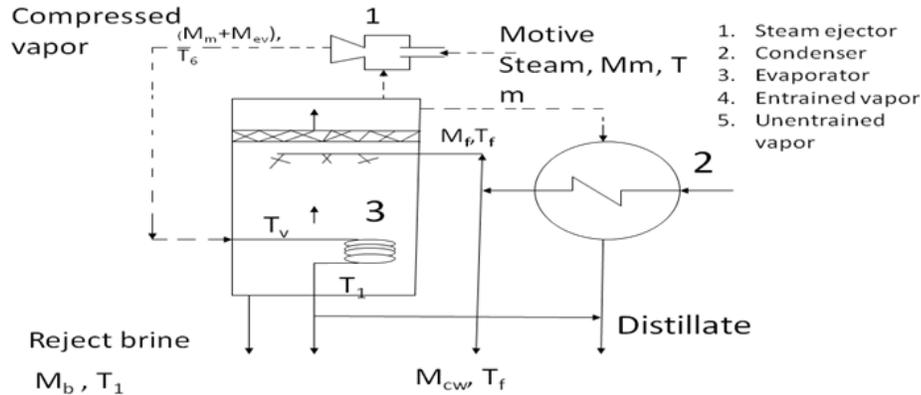


Fig. 3: Schematic of TVC desalination process.

ejector and the feed heater or the condenser. Auxiliary accessories such as the circulating pump, venting system and measuring and control instrumentations are not shown in the figure. A known mass of seawater ( $M_{cw} + M_f$ ) at temperature  $T_{cw}$ , and salt concentration  $X_f$ , is introduced into the tube side of the condenser where its temperature increases to  $T_f$ .

Heating of the feed seawater is essential as it enhances the thermal performance ratio of the process. After being heated in the condenser, the seawater stream leaving the condenser is divided into two parts. The first (larger) part, the cooling water, is rejected back to the sea and the second (smaller) part, the feed, is introduced into the evaporator. The function of the cooling water is the removal of the excess heat added to the system at a relatively high pressure and temperature. This heat is added to the system in the form of motive steam, which drives the steam ejector. The feed seawater is chemically treated and de-aerated to reduce foaming and control formation of scale resulting from the decomposition of the bicarbonate of seawater and the deposition of calcium sulphate due to its reversed solubility. De-aeration is done to reduce dissolved gases in seawater such as oxygen which causes corrosion, and scales and/ foaming which reduces heat transfer through affected surfaces and seriously pose operational difficulties and affect the performance of the evaporator (Ettouney *et al.*, 1999).

Within the evaporator, the feed water is sprayed at the top where it falls in the form of thin film down the succeeding rows of tubes arranged horizontally. The feed water temperature  $T_f$  is raised to the boiling temperature  $T_b$ . The boiling temperature is dictated by the nature of chemicals used to control the scale formation and the state of the heating steam. The vapour formed is free of salts. The boiling temperature,  $T_b$ , is higher than the temperature,  $T_v$ , of the generated vapour corresponding to the evaporation pressure by the Boiling Point Elevation (BPE) of the feed seawater. Boiling point elevation is the increase in boiling point due to dissolved salts at a given pressure. The vapour generated therein, flows through the

demister where entrained brine droplets are removed. The vapour should be completely free of these brine droplets to prevent contamination of both the product water and heat transfer surfaces on which it condenses. The presence of the brine droplets will erode the nozzles and diffusers in the steam jet ejector. The vapour flows from the demister to the condenser, where it splits into two portions: the first part condenses outside the tubes of the condenser, while the rest is entrained by the steam ejector, where it is compressed by motive steam to raise the pressure and temperature and then it is introduced into the evaporator shell side as the heat source (Ji *et al.*, 2007).

**Modeling of the tvc desalination process:** The main data usually required in the modelling of a thermal vapour compression desalination system are the amount of motive steam required for each unit mass of product distillate water (thermal performance ratio), the specific heat transfer area of heat exchangers and the amount of cooling water flowing into the condenser. A complete mathematical description of the thermal vapour compression desalination system requires equations for mass, energy, salt balances as well as heat transfer rate equations.

$$\text{Overall mass balance is given by } M_f = M_d + M_b \quad (3)$$

$$\text{Overall salt balance is given by } M_f X_f = M_b X_b \quad (4)$$

$$M_b = M_d X_f / (x_b - x_f) \quad (5)$$

**Evaporator:** The prediction of the performance of an evaporator involves relating quantities such as the total heat transfer rate, the inlet and outlet temperatures, the overall heat transfer coefficient, and the total surface area for heat transfer (Incopera and DeWitt, 1981).

The latent heat of the condensing steam in the shell side of the evaporator is given by:

$$Q_e = Q_s = (M_m + M_{ev})(h_s^v - h_s^l) = (M_m + M_{ev}) \dot{q}_s = M_s \dot{q}_s \quad (6)$$

where  $\lambda$  is latent heat of vapourization. The subscript  $s, e$  and  $ev$  denote the compressed or mixed steam (condensation side of the evaporator), evaporator and entrained vapour respectively and the superscripts  $v$  and  $l$  denote the vapour and liquid state respectively

$$Q_w = M_f \int_{T_f}^{T_b} C_p dT = M_f C_p (T_b - T_f) \quad (7)$$

$$C_p = (A + BT + CT^2 + DT^3) \times 10^{-2} \quad (8)$$

$T$  is the saturation temperature in °C. The variables A, B, C, and D are evaluated as functions of the seawater salinity, X in g/Kg with (Homig, 1978). The latent heat required to evaporate a certain quantity  $M_d$  of the seawater is also given by (El-Dessouky *et al.*, 1995):

$$Q_v = M_d \lambda_v$$

$$\lambda_v = 2589.583 + 0.9156T - 4.8343 \times 10^{-2}T^2 \quad (9)$$

$$Q_e = M_f C_p (T_b - T_f) + M_d \lambda_v \quad (10)$$

The saturation temperature of the vapour formed is given by:

$$T_v = T_b - \text{BPE} \quad (11)$$

the boiling point elevation is evaluated as a function of the seawater salinity and temperature from the empirical correlation given below by (El-Dessouky *et al.*, 1998):

$$\text{BPE} = X(F + GX) \times 10^{-3} \quad (12)$$

with F and G given as function of temperature, T:

$$F = (6.71 + 6.34 \times 10^{-2}T + 9.74 \times 10^{-5}T^2) \times 10^{-3} \quad (13)$$

$$G = (22.238 + 9.59 \times 10^{-3}T + 9.42 \times 10^{-5}T^2) \times 10^{-8} \quad (14)$$

The saturation pressure of vapour formed is given in terms of the saturation temperature in the form (Al-Najem *et al.*, 1997):

$$P_v = 10.1724607 - 0.6167302T_v - 1.832849 \times 10^{-2}T_v^2 - 1.77376 \times 10^{-4}T_v^3 + 1.47068 \times 10^{-6}T_v^4 \quad (15)$$

The condensation temperature of the steam on heat side of the evaporator is calculated in terms of the saturation pressure as (Al-Juwayhel *et al.*, 1997):

$$T_s = 42.6776 - \left[ \frac{3892.71}{\left( \frac{\ln P_v}{1000} \right) - 9.48654} \right] - 273.15 \quad (16)$$

$$Q_e = M_f CP(T_b - T_f) + M_d \lambda_v = U_e A_e (T'_s - T_b) \quad (17)$$

$$A_e = M_f CP(T_b - T_f) + M_d \lambda_v / U_e (T'_s - T_b) \quad (18)$$

$$U_e = 1969.5 + 12.057T_b - 8.5989 \times 10^{-2}T_b^2 + 2.565$$

**The condenser:** The condenser has the following three functions: removal of the excess heat from the desalination system, improving of the process thermal efficiency; and adjusting the boiling temperature inside the evaporator. The condensing steam is used to raise the temperature of the seawater from the intake temperature;  $T_{cw}$  to the evaporator feed temperature,  $T_f$ :

$$Q_c M_c (h_v - h_l) = M_c \lambda_v \quad (19)$$

$$h_v = 2501.689845 + 1.80691615T_v + 5.087717 \times 10^{-4}T_v^2 - 1.1221 \times 10^{-5}T_v^3 \quad (3.37) \quad (\text{El-Dessouky } et al., 1995).$$

$$h_l = -0.033635409 + 4.2075511T_v - 6.200339 \times 10^{-4}T_v^2 + 4.459374 \times 10^{-3}T_v^6 \quad (3.38) \quad (\text{El-Dessouky } et al., 1995).$$

The quantity of heat needed to warm the intake seawater from  $T_{cw}$  to  $T_f$  is given by:

$$Q_{M_f+M_{cw}} = (M_f + M_{cw}) \int_{T_{cw}}^{T_f} C_p dT \quad (20)$$

$$Q_c = M_c (h_g - h_f) = M_c \lambda_v = Q_{M_f+M_{cw}}$$

$$= (M_f + M_{cw}) CP (T_f - T_{cw}) \quad (21)$$

$$M_{cw} = [Q_c / C_p (T_f - T_{cw})] - M_f \quad (22)$$

The feed seawater temperature is given by (Ji *et al.*, 2007):

$$T_f = T_b - 5 \quad (23)$$

The specific cooling water flow rate is given by:

$$\text{SMc} = M_{cw} / M_d \quad (24)$$

The heat transfer between the condensing vapour and the intake seawater can be written in terms of an overall heat transfer coefficient  $U_c$ , the condenser heat transfer area  $A_c$  and the logarithmic mean temperature difference as follows:

$$M_c \lambda_v = M_{cw} C_p (T_f - T_{cw}) = U_c A_c (\text{LMTD})_c \quad (25)$$

$$A_c = [(M_f + M_{cw}) C_p (T_f - T_{cw})] / U_c (\text{LMTD})_c \quad (26)$$

$$\text{LMTD} = \frac{T_f - T_{cw}}{\ln \left[ \frac{T_c - T_{cw}}{T_c - T_f} \right]} \quad (27)$$

$$U_c = 1719.4 + 3.2T_v + 1.6 \times 10^{-2}T_v^2 - 2 \times 10^{-4}T_v^3 \quad (28)$$

(El-Dessouky *et al.*, 1995; Ettouney *et al.*, 1999). The overall heat transfer area is given by:

$$A_s = \frac{A_e + A_c}{M_d}$$

**The steam jet ejector:** The ejector which is shown in Fig. 4 is a pumping device which uses jet action of a high pressure and temperature primary motive fluid to entrain and accelerate a slower secondary fluid (load). The resulting kinetic energy of the mixture is subsequently used for self-compression to a higher pressure thus fulfils the function of a compressor. The steam ejector is used to compress the vapour from pressure  $P_{ev}$  (which is the vapour pressure leaving the evaporator) to an exit pressure (which is the vapour pressure entering the first effect) by using an external source of steam at a pressure  $P_m$  greater than the vapour pressure

The main methods available in the literature to determine the performance of a steam ejector include chart by (Power, 1994), the gas dynamic method by (Solokov and Zinger, 1977), and CFD by (Park *et al.*, 2005). The method used in this work is correlation developed by (El-Dessouky, 1997) which is based on the chart developed by (Power, 1994). The main data required from analyzing a steam jet ejector is the determination of the mass of motive steam per mass of entrained vapour (entrainment ratio),  $R_a$ , given the pressure of motive steam ( $P_m$ ), entrained vapour ( $P_{ev}$ ), and discharge pressure ( $P_s$ ) (du Plessis *et al.*, 2006). The ejector entrainment ratio is evaluated using following correlation:

$$R_a = \frac{M_m}{M_{ev}} = 0.296 \frac{(P_s)^{1.19}}{(P_{ev})^{1.04}} \left( \frac{P_m}{P_{ev}} \right)^{0.015} \frac{(PCF)}{(TCF)} \quad (29)$$

The mass flowrate of the motive steam is given by:

$$M_m = R_a \times M_{ev} \quad (30)$$

$$PCF = 3 \times 10^{-7}(P_m)^2 - 0.0009(P_m) + 1.6101 \quad (31)$$

$$TCF = 2 \times 10^{-8}(T_v)^2 - 0.0006(T_v) + 1.0047 \quad (32)$$

The ratio of the pressure of compressed discharge mixture to the pressure of the entrained vapour and is given by;

$$C_r = \frac{P_s}{P_{ev}} \quad (33)$$

**Wire mesh mist eliminator:** The pressure loss which occurs while the vapour generated from evaporator is flowing through the demister leads to a drop in temperature. The pressure drop and the subsequent temperature drop can be estimated by the following correlation developed by (El-Dessouky *et al.*, 2000) as:

$$\Delta p_{de} = 9.583 \times 10^{-2} \rho_{de}^{1.597} V_{de}^{0.7197} L_{de}^{1.388} \quad (34)$$

where  $\rho_{de}$ ,  $V_{de}$  and  $L_{de}$  are the demister pad density, vapour velocity in the demister and thickness of the demister pad respectively. Therefore, the temperature loss with vapour flow  $\Delta T_{de}$  can be obtained according to the corresponding saturated pressure difference:

$$P_c = P_v - \Delta P_{de} \quad (35)$$

$$T_c = T_b - BPE - \Delta T_{de} \quad (36)$$

A computer code, NUCDES developed by (Asiedu-Boateng *et al.*, 2011) in FORTRAN language was used to conduct the analysis. The effect of design and operating parameters on parameters controlling the cost of producing fresh water from TVC process the coupled the cogeneration nuclear power plant was studied and the result presented graphically.

## RESULTS AND DISCUSSION

**Cogeneration nuclear power plant cycle efficiency:** The dependence of the cycle efficiency of the cogeneration nuclear plant on the motive steam pressure is shown in Fig. 5.

The cycle efficiency of the cogeneration nuclear power plant is found to increase with the increase in pressure of the motive steam. This is because motive steam with high pressure possesses higher energy and therefore can transfer correspondingly higher amount of heat energy to the desalination plant and into useful work. Large amount of heat transferred to the desalination plant will lead to correspondingly large amount of product water production. This implies that it will be more profitable to operate at high motive pressure.

As is shown in Fig. 6, the system thermal performance (steam economy) decreases with increase in brine boiling temperature but improves with increase in motive steam pressure. The decrease in performance at high boiling temperature is caused by the increased motive steam consumption, which results from the increase in vapour pressure at higher temperatures. Therefore larger amounts of motive steam are needed to compress the vapour to higher pressures. High pressure motive steam has large amount of energy to compress large amounts of entrained vapour and hence improves the system performance. This means operating the desalination plant at high brine boiling temperature will draw more steam from the boiler of the nuclear plant and increase the cost of fresh water production but operating at high steam pressures will lead high amount of desalted water from the desalination unit.

**Specific cooling water flowrate:** The variation in cooling water flow rate with compression ratio is shown in Fig. 7. Cooling water flow rate reduces as the compression ratio increases. As the compression ratio increases, the pressure

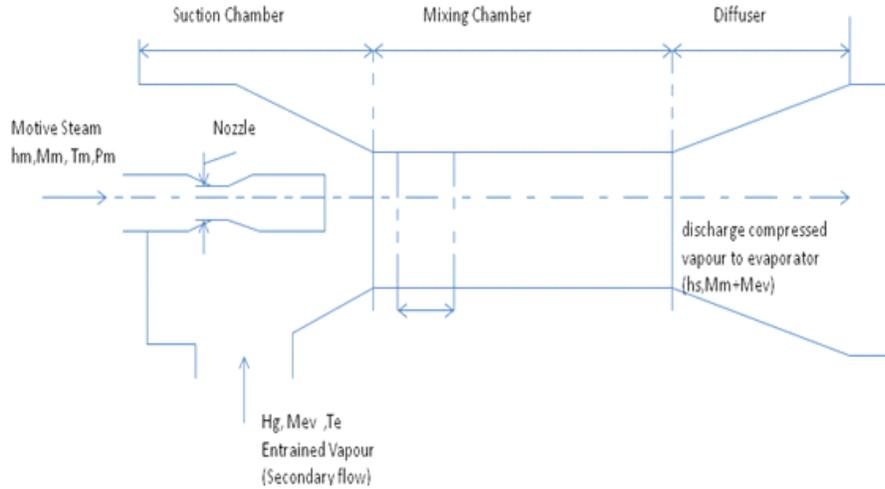


Fig. 4: Schematic of steam jet ejector

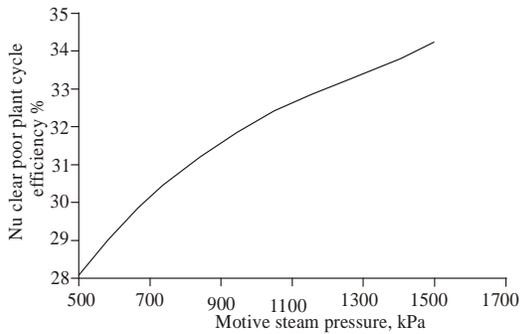


Fig. 5: Effect of motive steam pressure on cycle efficiency of cogeneration nuclear power plant

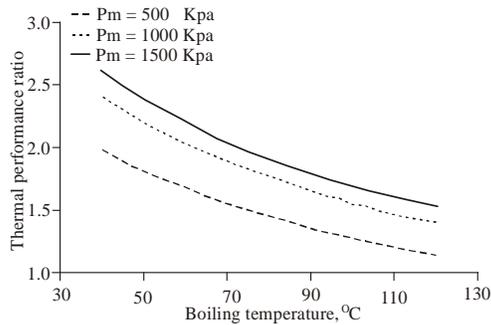


Fig. 6: Effect of brine boiling temperature on thermal performance ratio

of the compressed steam increases which subsequently increases the boiling temperature and saturation temperature thereby increasing the heat transfer coefficient. This condition improves heat transfer across the condenser tubes thus reducing the heat transfer area

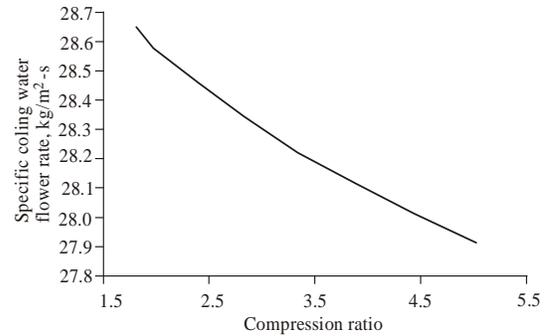


Fig. 7: Effect of compression ratio on specific cooling water flow rate specific heat transfer area

and consequently reducing the specific cooling water flow rate. Therefore operating at higher compression ratio will also ensure a more economical desalination since a smaller heat exchanger (ie condenser) will be required.

The effect of boiling temperature on the overall specific heat transfer area at selected compression ratios is shown in Fig. 8. The overall specific heat transfer area decreases with increasing brine boiling temperature. This is attributed to the fact that the heat transfer coefficient increases at higher boiling temperatures. The heat transfer rate is augmented by the increase in heat transfer coefficient which in turn reduces the heat transfer area. The overall heat transfer area also decreases with increasing compression ratio which produces high pressure compressed steam. As is shown in Fig. 9 the heat transfer area decrease with increase in the compression ratio.

This is due to the fact that high compression ratio produces high pressure compressed steam condensing with high condensing temperature. The latter results in

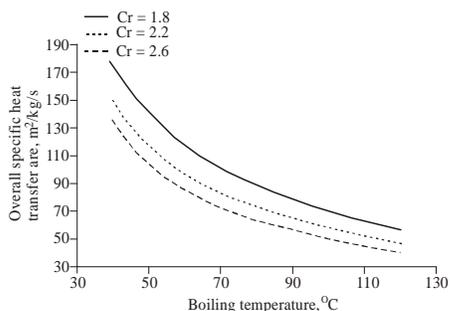


Fig. 8: Effect of boiling temperature on overall specific heat transfer area

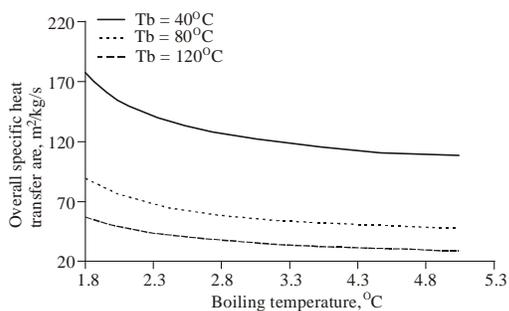


Fig. 9: Effect of compression ratio on overall specific heat transfer area

high driving force for heat transfer with a corresponding reduced heat transfer area. Hence it will be more economical to operate both the evaporator and condenser at moderately low brine boiling temperatures and high compression ratios since such desalination units will lead the use of small sized and more economic heat exchangers (evaporator and condenser).

### CONCLUSION

- The efficiency of cogeneration nuclear power plant increased from 28 to 35 as a function of the motive steam pressure between 500kPa and 1500kPA
- The overall specific heat transfer area is decreases at higher boiling temperatures, compression ratio and motive steam pressure. Hence operating the heat exchangers (evaporator and condenser) at lower operating conditions favours the desalination process from economic point of view.
- The specific cooling water flow rate decreases as brine boiling temperatures and motive steam pressures increases.

### LIST OF SYMBOLS

A (m <sup>2</sup> /kg-s)	Specific heat transfer area
Cp (kJ/kg-°C)	Specific heat capacity at constant pressure
Cr (dimensionless)	Compression ratio

g (m/s <sup>2</sup> )	acceleration due to gravity
hg (kJ/kg)	Specific enthalpy water of vapour
h <sub>l</sub> (kJ/kg)	Specific enthalpy liquid of water
M (kg/s)	Mass flow rate
P (kPa)	Pressure
Q (kJ/s)	Heat transfer rate
Ra (dimensionless)	Entrainment ratio
SMc (kg/m <sup>2</sup> -s)	Specific cooling water flow rate
SSc (kg/kWh)	Speificic steam consumption
T (°C)	Temperature
ΔT (°C)	Temperature drop
U (Kw/m <sup>2</sup> -°C)	Heat transfer coefficient
V, v, (m/s)	Velocity
W (kJ)	Work done
X (g/kg)	Salinity of seawater
Cw	cooling water
<b>Greek Letters</b>	
ρ (kg/m <sup>3</sup> )	Density
η (dimensionless)	Cycle efficiency
λ(kJ/kg)	Latent heat

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