ZIBELINE INTERNATIONAL PUBLISHING

ISSN: 2773-6202 (Online)
CODEN: JTIOCC

Journal of Technology & Innovation (JTIN)

DOI: http://doi.org/10.26480/jtin.01.2023.13.21

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REVIEW ARTICLE

# EFFICIENCY DETERIORATION OF CANDU NPPs DUE TO CHANGES IN CONDENSER COOLANT CHARACTERISTICS

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**ARTICLE DETAILS**

***Article History:***

Received 28 December 2022
Revised 05 January 2022
Accepted 01 February 2023
Available online 04 February 2023

**ABSTRACT**

This research investigates the harmful effects of changes in condenser seawater coolant temperature, fouling, and salinity, on the thermal performance of an existed CANDU nuclear power plant (NPP). A mathematical model is developed to relate seawater cooling temperature, fouling, and salinity to the output power and thermal efficiency of the plant. The model also elaborates the impact of the condenser performance on power and efficiency. The thermal efficiency of the considered CANDU NPP is reduced by 2.24%, for combined extreme increases in the condenser cooling seawater temperature, fouling factor of seawater and treated boiler feed water, and salinity of 10 ˚C, 0.0002, 0.00001 $m^2K/W$, and 100 g/kg, respectively. A rise in the condenser efficiency from 40 - 100 % increases the output power by 7.0136 % of the plant, and the thermal efficiency increases by about 2.34 %.

**KEYWORDS**

CANDU NPP, Condenser, Temperature, Fouling, Salinity, Efficiency

## 1. INTRODUCTION

The conclusive goal of designers of power stations, thermal or nuclear, is to achieve the best possible thermal efficiency. Operators afterwards should preserve and run the station close to the rated efficiency. The low temperature sink in a thermodynamic cycle is the heat rejected by the condenser, which is a crucial element of power plants (PPs). The proper operation in accordance with design data is of utmost importance for the station to operate efficiently. Therefore, it is desirable to have high temperature internally and low temperature in the external environment. This consideration encourages siting power plants alongside cold water. Most power plants have higher efficiencies in winter than in summer.

A power plant is designed according to calculated design conditions and data for optimum efficiency. However, in real life rated conditions cannot be preserved, since inlet conditions are not conforming with design data, hence the efficiency and output power decrease. A steam plant and the second cycle of a nuclear plant are composed of many components; each is designed for optimum operation in order to secure the overall plant efficiency. The condenser is not the largest equipment in the plant, but it is a base part in determining the plant efficiency.

Surface condensers are the types usually used in PPs. A large amount of heat is rejected to the environment through the condenser. The heat transfer area of a surface condenser in a PP is much larger than any heat exchanger in the plant. Attaining the lowest possible condensing temperature in the heat sink of a thermodynamic cycle is a fundamental goal in the design of surface condensers. Since saturation temperature and pressure of steam are proportionally related at low pressures, the goal of low condensing temperature necessitates a low condenser operating pressure. In general, as the condenser pressure increases, due to changes in its cooling water properties, both thermal efficiency and net output power decrease, and the steam consumption increases.

Condensers in PPs are normally cooled by water. Accordingly, factors affecting the condenser performance are related to the characteristics of the condenser cooling water. The present research concerns NPPs, which are mostly located near seas, and condensers are cooled by seawater. Three properties of the condenser cooling seawater are important: temperature, fouling, and salinity. In NPPs, actually all the waste heat has to be dumped into the condenser cooling water. Thermal PPs can run at higher temperatures than NPPs which must exclude damage of fuel rods for safety. So, the efficiency of modern thermal PPs is higher than that of nuclear ones. Thus, a lower efficiency plant will need to sink more heat than ones of higher efficiency, and this applied to NPPs. As the condenser is the sink, then it is important in NPPs to mitigate any unfavorable effects due to changes in condenser coolant characteristics, which could reduce the plant's efficiency. This is more pronounced in CANDU NPPs because of their lower efficiencies than other NPPs designs.

The water temperature differs significantly from one site to another. The design of the condenser in a power station depends on the inlet cooling water temperature. Therefore, the power station should be sited carefully according to the cooling water temperature of the source. Cooling water temperature increase means condenser pressure increase, and consequently plant efficiency decrease. For a pressurized water reactor nuclear power plant (PWR NPP), the output power and the thermal efficiency of the plant decrease by approximately 0.3929 and 0.16%, respectively, for 1 °C increase in the temperature of the condenser cooling seawater (Ibrahim et al., 2014). Increasing the inlet cooling seawater temperature by 15 $^0$C decreases the efficiency and the output power by 2 and 6%, respectively, of the 1450 MW power cycle of the APR 1400 PWR NPP in the United Arab Emirates located at the Arabian Gulf (Koo and Jeong, 2013). An increase of 1 $^0$C of the coolant water decreased the output power and thermal efficiency of a PWR NPP by about 0.45 and 0.12%, respectively (Duramaya and Sogut, 2006). A rise of 1 °C in the condenser cooling water temperature reduced the supply of a nuclear power by about 0.5% (Linnerud et al, 2011). For a 225 MW steam power plant, for every 1 $^0$C increase in the condenser cooling seawater temperature, the output power and efficiency decreased by about 0.171%,and 0.168%, respectively (Darmawan and Yuwoo, 2019). A 1 $^0$C increase in the inlet condenser cooling water temperature of a thermal PP decreased the output power by 33 MW (Pattanayak et al., 2019).

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| | **Website:**<br>www.jtin.com.my | **DOI:**<br>10.26480/jtin.01.2023.13.21 |

Cite The Article: Said M.A. Ibrahim, Ismail M.A. Aggour (2023). Efficiency Deterioration of Candu Npps Due to Changes in Condenser Coolant Characteristics. *Journal of Technology & Innovation, 3(1): 13-21.*

The fouling of heat exchangers is due to the accumulation of unwanted deposits on heat transfer surfaces. The fouled layer imposes an additional resistance to heat transfer, and the contraction of the flow area, due to fouling, results in an increased flow velocity for a given volumetric flow rate. There is an increased resistance to the flow of the fluid across the deposited surface. Also, the cooling water flow velocity is usually low in shell and tube steam condensers, and this gives rise to more fouling accumulation in the tubes. The practical consequences of fouling are reduction in the exchanger efficiency and excessive pressure drop across it. To counteract the expected fouling, the heat exchanger should have an excess area over that required to give the same heat rate in the clean condition. A study on a PWR NPP revealed that an increase in the cooling seawater fouling factor in the range of 0.00015–0.00035 $m^2$ K/W had led to a decrease in the plant output power and thermal efficiency of 1.36 and 0.448%, respectively (Ibrahim and Attia, 2015).

The effect of the thermal resistance of fouling for old and new condensers, after one year of operation, on the power output of a condensing turbine indicated that for old condensers the fouling resistance could reach 0.0007 $m^2$K/W, and this reduced the turbine power output by up to 4.1%, whereas for new condensers the power output drop did not exceed 1.5% (Alabrudzinski, et al., 2016). For evaporative coolers and condensers, it has been concluded that the maximum decrease in effectiveness due to fouling was 78% for condensers (Qureshi and Zubair, 2005). Steinhagen shows that undesirable design procedures and operation problems of heat exchangers typically oversize them by 70-80% of which 30-50% is attributed to fouling (Steinhagen, 1999). If the overall heat transfer coefficient for the fouled condition becomes half the coefficient for clean condition, then the heat transfer area is doubled (Gautam et al., 2017). The effect of fouling on the effectiveness and water outlet temperature of a cooling tower, demonstrated about 0.6% decrease in the effectiveness and about 1.2% increase in the water outlet temperature (Rehman et al., 2004).

Salinity is a thermodynamic state variable that, along with temperature and pressure, controls the physical characteristics of saline water. Thus, changes in the salinity of the condenser cooling seawater could affect its performance. Increasing the condenser cooling seawater salinity in a PWR NPP, by 10, 50, and 100 g/kg resulted in losses in the thermal efficiency by 0.011, 0.06, and 0.14%, respectively, and the respective reductions in the output power for the same salinity values were 0.033, 0.039, and 0.044% (Ibrahim and Attia, 2015). For a PWR NPP, a loss of 0.2% in the plant efficiency was calculated for temperature and salinity values of 5 K, and 10000 ppm (Ibrahim and Badawy, 2014). The thermal performance of a seawater cooling tower was investigated to depict that the air effectiveness decreased with increasing the seawater salinity, with a maximum decrease of 15% for a salinity of 85 g/kg (Sharqawy et al., 2011).

We could not find research work, to our ability, concerning the combined effect of temperature, fouling, and salinity of the condenser cooling seawater on the thermal performance of NPPs except that by on PWR (Ibrahim and Attia, 2015). Some researchers exhibited a significant loss in the plant output power and thermal efficiency by up to 8.242 and 2.77%, respectively, for an increase in the condenser cooling seawater temperature from 15–30 °C, fouling factor from 0.00015–0.00035 $m^2$ K/W, and salinity from 0–100 g/kg (Ibrahim nd Attia, 2015).

Since the efficiency of NPPs are lower than that of thermal PPs, therefore, it is important to mitigate or avert if at all possible, all factors that reduce their efficiency. To achieve this, relations between such factors and the thermal efficiency and output power of NPPs should be studied and analyzed. The current research contributes to the negative impacts of variations in the condenser inlet cooling seawater temperature, fouling, and salinity on the thermal efficiency and power output of a particular CANDU NPP. CANDU reactors are pressurized heavy water reactors (PHWR), so the results concerned with PWRs are relevant. The paper covers the individual effects of these properties as well their combined impacts. We used a developed a model by based on thermodynamic and heat balance considerations, to calculate the required effects, and numerical solutions were performed by a computer program (Ibrahim and Attia, 2015). The work is useful since there is no research conducted on this type of plant relevant to the subject of the present study, as well as that CANDU NPPs have lower efficiency than PWR NPPs and Boiling Water Reactor (BWR) NPPs.

## 2. The Selected NPP

Point Lepreau nuclear power plant has been chosen as the present proposed CANDU NPP. The plant is located on the shores of the Bay of Fundy, approximately 40 km west of Saint John, New Brunswick, Canada. This is a 660 MW CANDU 6 plant; the first to generate electricity commercially. In CANDU NPPs, the channels, which contain the fuel bundles, are housed in a horizontal cylindrical tank (called calandria) that contains cool heavy water ($D_2O$) moderator at low pressure. Pressurized $D_2O$ coolant is circulated through the fuel channels (calandria tubes) and steam generators in a closed circuit. The coolant carries the fission heat to steam generators, where it is transferred to light water to produce steam to be delivered to the turbine. Components of the secondary cycle of the plant, which we are concerned with, are shown in Figure 1.

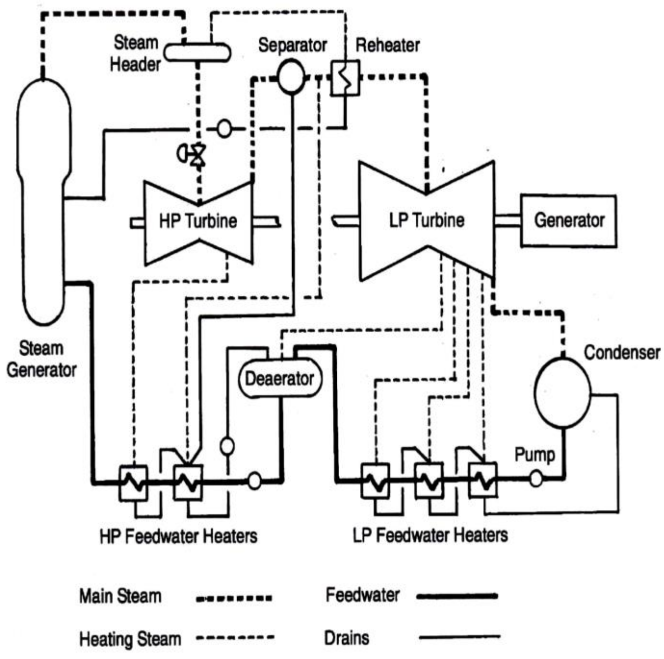



**Figure 1:** The secondary cycle of the proposed CANDU NPP (Garaland, 2014)

Technical design data of the secondary cycle of the Point Lepreau plant are given in Table 1

**Table 1:** Design data of the secondary cycle of the plant

| Item | Value |
|---|---|
| Vapor conditions:<br>Before the high-pressure turbine<br>Before the low pressure turbine<br>Inside the condenser | 4.55/258 MPa/°C<br>0.588/242 MPa/°C<br>4.0/30 kPa/°C |
| Number of preheating stages<br>Number of low-pressure preheating stages<br>Number of high pressure preheating stages | 6 stages<br>4 stages<br>2 stages |
| Feed water temperature | 187 °C |
| Generator power | 728 MW |
| Gross efficiency | 32.8% |
| Aggregate net power | 678 MW |
| Steam flow rate | 1047 kg/s |
| Condenser cooling water flow rate | 25.8 $m^3$/s |
| Condenser cooling water inlet temperature | 5 °C |
| Condenser cooling water temperature increase | 10 °C |

## 3. The Present Model

A mathematical model is used in this study to determine the impacts of changes in the condenser seawater cooling temperature, fouling, and salinity on the output power and thermal efficiency of a real CANDU NPP. The model equations were deduced according to the assumption in Table 2. The equations of the model were solved by a developed computer program as per the data of the plant given in Table 1 and the calculation values given in Table 3.

### 3.1 Model Assumptions

The assumptions made in the model are listed in Table 2

**Table 2:** Assumptions

| Item | Assumption |
|---|---|
| Steam exit conditions | Constant |
| Cooling water temperature difference | Constant |
| Condenser vacuum pressure at constant flow rate of coolant | Variable with coolant temperature |
| Mass flow rates of coolant and condensate | Constant |
| Total surface area of condenser tubes | Constant |
| Properties of condenser tubes material | Constant |
| Pressure drop across the condenser | None |
| Condenser heat transfer area | Constant |
| Condenser heat load | Constant |
| Potential and kinetic energies of the coolant flow | Neglected |
| Heat losses from equipment and pipes | Neglected |

## 3.2 Calculation Values

The model calculation values are given in Table 3.

**Table 3:** Model values

| Factor | Range of calculations |
|---|---|
| Inlet cooling water temperature | 5-15 °C |
| Fouling factor | 0.00015–0.00035 m²K/W |
| Salinity | 0 – 100 g/kg |
| Treated boiler feed water | 0.00005 - 0.00015 m²K/W |

## 3.3 Model Formulation

The following model equations were previously developed by (Ibrahim & Attia, 2015).

### 3.3.1 Thermodynamic Analysis

The energy balance equations for the various processes involving steady flow equipment such as the nuclear reactor, turbines, pumps, steam generators, and condensers are:

- Heat added to steam from reactor or steam generator, $Q_{add}$ is

$$Q_{add}=\dot{m}_{st}(h_{out}-h_{in})\ kW \tag{1}$$

Where: $\dot{m}_{st}$ is the mass flow rate of exit steam from reactor or steam generator, kg/s, $h_{in}$ is the enthalpy of inlet feed water to reactor or steam generator, kJ/kg, and $h_{out}$ is the enthalpy of outlet steam from reactor or steam generator, kJ/kg.

- Total turbine work, $W_T$ is

$$W_T=W_{HPT}+W_{LPT}=\dot{m}_{st}\,(h_{in}-h_{out})+\dot{m}_{st}\,(h_{in}-h_{out})\quad kW \tag{2}$$

Where: $W_{HPT}$ is the high pressure turbine work, kW, $W_{LPT}$ is the low pressure turbine work, kW, $\dot{m}_{st}$ is the mass flow rate of inlet steam to turbine, kg/s, $h_{in}$ is the enthalpy of inlet steam to turbine, kJ/kg, and $h_{out}$ is the enthalpy of outlet steam from turbine, kJ/kg.

- Pump work, $W_P$ is

$$W_p=W_{cp}+W_{fwp}=\dot{m}_{fw}\,(h_{in}-h_{out})+\dot{m}_{fw}\,(h_{in}-h_{out})\quad kW \tag{3}$$

Where: $W_{cp}$ is the condensate pump work, kW, $W_{fwp}$ is the feed water pump work, kW , $\dot{m}_{fw}$ is the mass flow rate of inlet feed water to pump, kg/s, $h_{in}$ is the enthalpy of inlet feed water to pump, kJ/kg, and $h_{out}$ is the enthalpy of outlet feed water from pump, kJ/kg.

- Heat rejected from condenser, $Q_{Rej}$ is

$$Q_{Rej}=(\dot{m}_{mix}\times h_{in}-\dot{m}_{fw}\times h_{out})\ kW \tag{4}$$

Where: $\dot{m}_{mix}$ is the mass flow rate of inlet mixture to condenser, kg/s, $h_{in}$ is the enthalpy of inlet mixture to condenser, kJ/kg, and $h_{out}$ is the enthalpy of outlet feed water from condenser, kJ/kg.

- Thus, the net work done, $W_{net}$ and thermal efficiency, $\eta_{th}$ can be calculated

### 3.3.2 Heat Balance Analysis

#### 3.3.2.1 Feed Water Heaters (FWHs)

Open and closed FWHs are commonly used in PPs where the job is accomplished in the former at constant pressure, and a heat exchanger is used in the latter to transfer heat between the hot bled steam and cold feed water without mixing, and the two streams can be maintained at their working pressures.

- Closed feed water heaters

Most feed water heaters are of shell and tube type. The condensed steam from each feed water heater drains successively to the next lower pressure heater and is returned to the feed water via heater drain pumps. The heat balance equation is

$$\dot{m}_{st}\times(h_1-h_2)=\dot{m}_{fw}\times(h_{out}-h_{in}) \tag{5}$$

Where: $\dot{m}_{st}$ is the steam mass flow rate extracted from turbine to feed water heater, kg/s, $\dot{m}_{fw}$ is the mass flow rate of inlet feed water to feed water heater, kg/s, $h_1$ is the enthalpy of inlet steam to feed water heater, kJ/kg, $h_2$ is the enthalpy of outlet steam from feed water heater, kJ/kg, $h_{in}$ is the enthalpy of inlet mixture to feed water heater, kJ/kg, and $h_{out}$ is the enthalpy of outlet feed water from feed water heater, kJ/kg.

- Deaerator

Deaerators in most PPs utilize low pressure steam bled from extraction locations in the steam turbine system. The heat balance is

$$(\dot{m}_{st}+\dot{m}_{fw})\times h_{out}=(\dot{m}_{st}\times h_1)+(\dot{m}_{fw}\times h_{in}) \tag{6}$$

Where: $\dot{m}_{st}$ is the mass flow rate of steam extracted from turbine to deaerator, kg/s $\dot{m}_{fw}$ is the mass flow rate of inlet feed water to deaerator, kg/s, $h_1$ is the enthalpy of inlet steam to deaerator, kJ/kg, $h_{in}$ is the enthalpy of inlet feed water to deaerator, kJ/kg, and $h_{out}$ is the enthalpy of outlet feed water from deaerator, kJ/kg.

#### 3.3.2.2 The Steam Generator

The main function of the steam generator is to supply high temperature and pressure steam to the turbine. The steam generator is the principal controlling link between the nuclear reactor and the turbine generator. The power in the primary coolant system and the secondary steam system must be balanced under all operating conditions. Any imbalance will cause an accumulation or depletion of the total heat content in the steam generator inventory, which in turn will result in a rise or fall of steam pressure. Steam pressure is therefore a key parameter in plant control. The steam generator separates the primary radioactive coolant circuit from the secondary steam circuit and hence prevents radioactivity from reaching the steam turbine house.

The primary coolant enters steam generators at nearly saturated conditions or saturated with some steam present. The heat balance equation is:

$$\dot{m}_{RCW}*C_{RCW}*(T_{HL}-T_{CL})=(\dot{m}_{st}*h_{out})-(\dot{m}_{fw}*h_{in}) \tag{7}$$

Where: $C_{RCW}$ is the specific heat of reactor coolant water of primary circuit, kJ/kg K, $T_{HL}$ is the temperature of reactor coolant water at hot leg, °C, $T_{CL}$ is the temperature of reactor coolant water at cold leg, °C, $\dot{m}_{RCW}$ is the mass flow rate of reactor coolant water of primary circuit, kg/s, $\dot{m}_{fw}$ is the mass flow rate of inlet feed water to steam generator, kg/s, $\dot{m}_{st}$ is the mass flow rate of outlet steam from steam generator, kg/s, $h_{in}$ is the enthalpy of inlet feed water to steam generator, kJ/kg, and

$h_{out}$ is the enthalpy of outlet steam from steam generator, kJ/kg.

### 3.3.3 Moisture Separator and Reheater

Reheating is preceded by the removal of extra moisture from the steam leaving the high-pressure turbine by separators. The heat balance of the separator is:

$$\dot{m}_r*(h_1-h_2)=(\dot{m}_s*h_s)+((\dot{m}_{st}-\dot{m}_s)*h_{out})-(\dot{m}_{st}*h_{in}) \tag{8}$$

Where: $\dot{m}_{st}$ is the tass flow rate of inlet steam to moisture separator and reheater, kg/s, $\dot{m}_s$ is the mass flow rate of drained water from moisture separator and reheater to deaerator, kg/s $\dot{m}_r$ is the mass flow rate of inlet reheating steam to moisture separator and reheater, kg/s $h_{in}$ is the enthalpy of inlet wet steam to moisture separator and reheater, kJ/kg, $h_{out}$ is the enthalpy of outlet superheated steam from moisture separator and reheater, kJ/kg, $h_s$ is the enthalpy of water drained from moisture separator and reheater to deaerator, kJ/kg, $h_1$ is the enthalpy of inlet reheating steam to moisture separator and reheater, kJ/kg, and $h_2$ is the enthalpy of outlet reheating steam from moisture separator and reheater to steam generator or feed water heater, kJ/kg.

### 3.3.4 The Condenser

The heat rejected in the condenser is:

$$Q_{Rej} = (\dot{m}_{mix} * h_{in}) - (\dot{m}_{fw} * h_{out}) = \dot{m}_{CW} * C * (T_{cwo} - T_{cwi}) \quad (9)$$

$$Q_{Rej} = U * A * \left(\frac{(Tcwo - Tcwi)}{ln\left(\frac{(Tc - Tcwi)}{(Tc - Tcwo)}\right)}\right) \quad (10)$$

Where: $\dot{m}_{CW}$ is the cooling water mass flow rate of condenser, kg/s, $\dot{m}_{fw}$ is the feed water mass flow rate outlet from condenser, kg/ s, $\dot{m}_{mix}$ is the mixture mass flow rate inlet to condenser, kg/s, $h_{in}$ is the enthalpy of inlet mixture to condenser, kJ/kg, $h_{out}$ is the enthalpy of outlet feed water from condenser, kJ/kg, $T_c$ is the condenser saturation temperature, °C, $T_{cwo}$ is the temperature of outlet cooling water from condenser, °C, $T_{cwi}$ is the temperature of inlet cooling water to condenser, °C, U is the overall heat transfer coefficient, W/m²K, C is the specific heat of water, kJ/kg K, and A is the eat transfer area, m².

### 3.3.5 Main Factors Affected by Changes in Cooling Seawater Characteristics

- Inside overall heat transfer coefficient, $U_i$

$$U_i = \frac{1}{(A_i * (R_i + R_w + R_o + R_{f,i} + R_{f,o}))} \quad W/m^2K \quad (11)$$

Where: $A_i$ is the inside tube surface area, m² $R_i$ is the thermal resistance of inner seawater, K/W, $R_o$ is the thermal resistance of outer condensation film, K/W, and $R_w$ is the thermal resistance of tube wall, K/W. $R_{f,i}$ is the fouling factor thermal resistance inside condenser tubes, K/W, and $R_{f,o}$ is the fouling factor thermal resistance outside condenser tubes, K/W.

- Outside overall heat transfer coefficient, $U_o$

$$U_o = \frac{1}{(A_o * (R_i + R_w + R_o + R_{f,i} + R_{f,o}))} \quad W/m^2K \quad (12)$$

Where: $A_o$ is the outside tube surface area, m² $R_i$ is the thermal resistance of inner seawater, K/W, $R_o$ is the thermal resistance of outer condensation film, K/W, and $R_w$ is the thermal resistance of tube wall, K/W. $R_{f,i}$ is the fouling factor thermal resistance inside condenser tubes, K/W, and $R_{f,o}$ is the fouling factor thermal resistance outside condenser tubes, K/W.

- Thermal resistance of inside seawater, $R_i$

$$R_i = \frac{1}{A_i * h_i} \quad K/W \quad (13)$$

Where $A_i$ is the inside tube surface area, m², and $h_i$ is the eat transfer coefficient for flow inside circular tubes, W/m² K.

- Thermal resistance of outside seawater, $R_o$

$$R_o = \frac{1}{A_o * h_o} \quad K/W \quad (14)$$

Where: $A_o$ is the outside tube surface area, m², and $h_o$ is the film condensation heat transfer coefficient in bundles of horizontal tubes, W/m² K.

- Thermal resistance of tube wall, $R_w$

$$R_w = \frac{ln\left(\frac{r_o}{r_i}\right)}{2\, \pi\, L\, k} \quad K/W \quad (15)$$

Where: k is the thermal conductivity of tube, W/m K, $r_o$ is the outer radius, m, $r_i$ is the inner radius, m, and L is the tube length, m.

- Thermal resistance of seawater fouling factor, $R_f$

$$R_f = \frac{f}{A} K/W \quad (16)$$

Where: f is the fouling factor, m² K/W, and A is the tube surface area, m².

- Heat transfer coefficient of flow inside circular tubes, $h_{Ri}$

$$h_i = \frac{N_u * k_{sw}}{d} \quad W/m^2K \quad (17)$$

Where: $N_u = 0.023 * R_e^{0.8} * P_r^{0.4}$ , $k_{sw}$ is the seawater thermal conductivity, W/m K, and d is the tube diameter, m.

- Film condensation heat transfer coefficient in bundles of horizontal tubes, $h_o$ is (Holman, 2010)

$$h_o = 0.725 \left(\frac{g * \rho_l * (\rho_l - \rho_v) * h_{fg} * k^3}{\mu_l * (T_{st} - T_w) * N_h * d_o}\right)^{0.25} \quad W/m^2K \quad (18)$$

Where: $\rho_l$ is the liquid density, kg/m³, $\rho_V$ is the steam or vapor density, kg/m³, $\mu_l$ is the liquid dynamic viscosity, N/m².s, $T_{st}$ is the steam or vapor saturation temperature, °C, $N_h$ is the number of horizontal tubes, $h_{fg}$ is the latent heat for condensation, kJ/kg, k is the thermal conductivity of liquid, W/m K, g is the acceleration of gravity, m/s², and $T_w$ is the condenser tube surface wall temperature, °C.

- Seawater density, $\rho_{sw}$

A correlation to get the density of seawater, $\rho_{sw}$ is (Sharqawy et al, 2010)

$$\rho_{sw} = \begin{pmatrix} (a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4) + \\ (b_1S_p + b_2S_pT + b_3S_pT^2 + b_4S_pT^3 + b_5S_pT^4) \end{pmatrix} \quad kg/m^3 \quad (19)$$

Where: $a_1=9.999\times10^2$, $a_2=2.034\times10^{-2}$, $a_3=-6.162\times10^{-3}$, $a_4=2.261\times10^{-5}$, $a_5=-4.657\times10^{-8}$, and $b_1= 8.020\times10^2$, $b_2=-2.001$, $b_3=1.677\times10^{-2}$, $b_4=-3.060\times10^{-5}$, $b_5=-1.613\times10^{-5}$.

- Seawater specific heat, $C_{psw}$

The specific heat of seawater, $C_{psw}$ is a function of both temperature and salinity. A correlation for $C_{psw}$ is (Sharqawy et al, 2010)

$$C_{psw} = A + B\,T + C\,T^2 + D\,T^3 \; kJ/kg\,K \quad (20)$$

Where: $A = 5.328 - 9.76\times10^{-2} S_p + 4.04\times10^{-4}S_p^2$, $B = -6.913\times10^{-3} + 7.351\times10^{-4} S_p - 3.15\times10^{-6} S_p^2$, $C = 9.6\times10^{-6} - 1.927\times10^{-6} S_p + 8.23\times10^{-9} S_p^2$, and $D = 2.5\times10^{-9} + 1.666\times10^{-9} S_p - 7.125\times10^{-12} S_p^2$.

- Seawater thermal conductivity, $k_{sw}$

The thermal conductivity, $k_{sw}$, is an important property of seawater and most difficult to measure. Consequently, data on seawater thermal conductivity is rather limited. For aqueous solutions containing an electrolyte, such as seawater, $k_{sw}$ usually decreases with an increase in the concentration of dissolved salts. The $k_{sw}$ correlation is (Sharqawy et al., 2010)

$$log_{10}(k_{sw}) = log_{10}(240 + 0.0002\,S_p) + 0.434\left(2.3 - \frac{343.5 + 0.037\,S_p}{T + 273.15}\right)\left(1 - \frac{T + 273.15}{647 + 0.03\,S_p}\right)^{0.333} \quad W/m\,K \quad (21)$$

Where: the validity is for $0 < T < 180$ °C and $0 < S_p < 160$ g/kg, with accuracy of ± 3 %.

- Seawater dynamic viscosity, $\mu_{sw}$,

Dynamic viscosity of seawater, $\mu_{sw}$ changes as a function of both temperature and salinity. A correlation for it is (Sharqawy et al, 2010)

$$\mu_{sw} = \mu_w\,(1 + A\,S_p + B\,S_p^2)\; kg/m\,s \quad (22)$$

Where: $A = 1.541 + 1.998 \times 10^{-2}T - 9.52 \times 10^{-5}T^2$, $B = 7.974 - 7.561 \times 10^{-2}T + 4.724 \times 10^{-4}T^2$, and $\mu_w = 4.2844 \times 10^{-5} + (0.157\,(T + 64.993)^2 - 91.296)^{-1}$.

### 3.3.6 Condenser Efficiency and Loss Factor

The present model studies the effect of the condenser efficiency, $\eta_c$ on the exhaust steam temperature and pressure, condenser loss factor (LF), and output power and thermal efficiency of the plant. The LF is defined as the ratio of the heat released by the steam entering the condenser to the heat gained by the cooling water. $\eta_c$ and LF are calculated from (Dutta et al., 2011).

$$\eta_c = \left(\frac{(T_{cwo} - T_{cwi})}{(T_c - T_{cwi})}\right) \% \quad (23)$$

$$LF = \left(\frac{(\dot{m}_{mix} * h_{in}) - (\dot{m}_{fw} * h_{out})}{\dot{m}_{CW} * C * (T_{cwo} - T_{cwi})}\right) \quad (24)$$

Where: $\dot{m}_{mix}$ is the mass flow rate of inlet mixture to condenser, kg/s, $\dot{m}_{fw}$ is the mass flow rate of outlet feed water from condenser, kg/s, $\dot{m}_{CW}$ is the mass flow rate of inlet cooling water to condenser, kg/s, $h_{in}$ is the enthalpy of inlet mixture to condenser, kJ/kg, and $h_{out}$ is the enthalpy of outlet feed water from condenser, kJ/kg.

## 4. Results and Discussion

### 4.1 Thermodynamic Data and State Diagrams of Plant Components

Thermodynamic heat balance calculations are performed by employing the computer software engineering equation solver (EES) to determine the thermodynamics properties at inlet and exit of each component in the steam cycle of the proposed NPP. Thus, key parameters could be obtained to get the needed state of the plant such as the heat added to steam, heat rejection, turbine output power, and overall thermal efficiency. Figure 2 illustrates the EES model equivalent of the thermodynamic and heat balance analyses for the secondary cycle of the CANDU NPP.

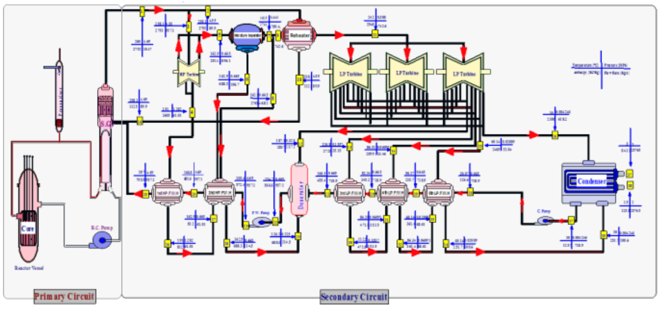


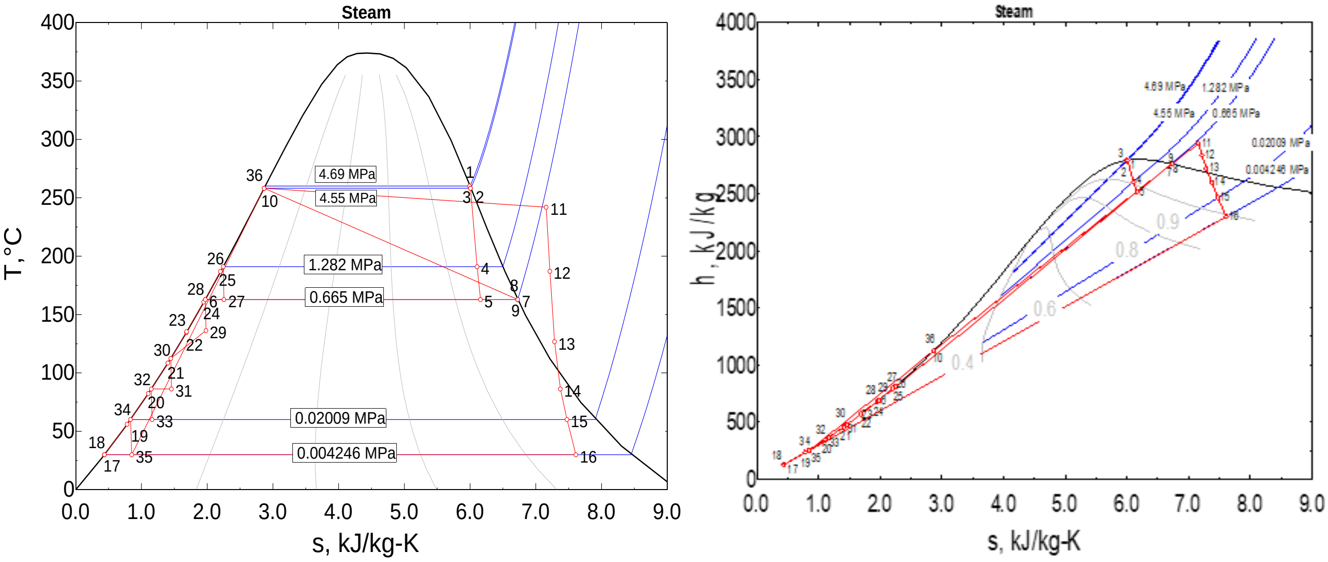

**Figure 2:** Model equivalent of thermodynamic and heat balance for NPP

The calculated inlet and exit thermodynamic properties of each component in the secondary cycle of the plant at design conditions are presented in Table 4.

**Table 4:** Thermodynamic Data for the Proposed NPP

| Point No. | Temperature, T (⁰C) | Pressure, p (MPa) | Enthalpy, h (kg/kJ) | Entropy, s (kJ/kg K) | Quality, X | Mass flow rate,(kg/s) |
|---|---|---|---|---|---|---|
| 1 | 260 | 4.69 | 2792 | 5.993 | 0.9975 | 1047 |
| 2 | 258.1 | 4.55 | 2793 | 6.006 | 0.9975 | 89.9 |
| 3 | 258.1 | 4.55 | 2793 | 6.006 | 0.9975 | 957.1 |
| 4 | 191 | 1.282 | 2605 | 6.108 | 0.9079 | 61.01 |
| 5 | 162.9 | 0.665 | 2514 | 6.159 | 0.8809 | 896.1 |
| 6 | 162.9 | 0.665 | 688.3 | 1.972 | 0 | 106.7 |
| 7 | 162.9 | 0.665 | 2761 | 6.725 | 1 | 789.4 |
| 8 | 162.9 | 0.665 | 2761 | 6.725 | 1 | 46.82 |
| 9 | 162.9 | 0.665 | 2761 | 6.725 | 1 | 742.6 |
| 10 | 258.1 | 4.55 | 1125 | 2.867 | 0 | 89.9 |
| 11 | 242 | 0.588 | 2940 | 7.158 | 100 | 742.6 |
| 12 | 187.1 | 0.325 | 2837 | 7.215 | 100 | 23.72 |
| 13 | 126.8 | 0.1557 | 2725 | 7.286 | 100 | 35.15 |
| 14 | 86.31 | 0.06091 | 2599 | 7.374 | 0.9761 | 33.46 |
| 15 | 60.14 | 0.02009 | 2465 | 7.474 | 0.939 | 32.04 |
| 16 | 30 | 0.004246 | 2300 | 7.61 | 0.895 | 618.2 |
| 17 | 30 | 0.004246 | 125.7 | 0.4366 | 0 | 718.9 |
| 18 | 30.07 | 0.665 | 126.6 | 0.4373 | -100 | 718.9 |
| 19 | 56.17 | 0.665 | 235.7 | 0.7825 | -100 | 718.9 |
| 20 | 82.34 | 0.665 | 345.2 | 1.103 | -100 | 718.9 |
| 21 | 108.5 | 0.665 | 455.4 | 1.402 | -100 | 718.9 |
| 22 | 134.7 | 0.665 | 566.6 | 1.683 | -100 | 957.1 |
| 23 | 135.4 | 4.69 | 572.4 | 1.687 | -100 | 957.1 |
| 24 | 160.8 | 4.69 | 681.6 | 1.946 | -100 | 957.1 |
| 25 | 187 | 4.69 | 795.9 | 2.202 | -100 | 957.1 |
| 26 | 191 | 1.282 | 812 | 2.245 | 0 | 61.01 |
| 27 | 162.9 | 0.665 | 812 | 2.256 | 0.05967 | 61.01 |
| 28 | 162.9 | 0.665 | 688.3 | 1.972 | 0 | 214.5 |
| 29 | 136.3 | 0.325 | 688.3 | 1.982 | 0.05335 | 214.5 |
| 30 | 112.5 | 0.1557 | 471.8 | 1.446 | 0 | 35.15 |
| 31 | 86.31 | 0.06091 | 471.8 | 1.457 | 0.04817 | 35.15 |
| 32 | 86.34 | 0.06091 | 361.4 | 1.15 | 0 | 68.61 |
| 33 | 60.14 | 0.02009 | 361.4 | 1.162 | 0.04652 | 68.61 |
| 34 | 60.17 | 0.02009 | 251.7 | 0.833 | 0 | 100.6 |
| 35 | 30 | 0.004246 | 251.7 | 0.8524 | 0.05188 | 100.6 |
| 36 | 258.1 | 4.69 | 1125 | 2.867 | -100 | 89.9 |

Figure 3 shows the calculated thermodynamic and heat balance analyses of the steam Rankine cycle of the NPP, on T-s and h-s diagrams as obtained from the heat balance of the plant. The state conditions for points 1 to 36 are as given in Table 4.




(a) (b)

**Figure 3:** (a) T- s diagram and (b) h-s diagram for the selected CANDU NPP

### 4.2 The Effect of Condenser Coolant Temperature on $\dot{W}_{net}$ and $\eta_{th}$

The heat balance model calculates the saturation temperature $T_C$ and corresponding condensate pressure $P_C$ and $T_{cwe}$, for $T_{cwi}$ between 5 to 15 °C and condenser terminal temperature difference ($TTD_C=T_C - T_{cwe}$) from 15 to 17 °C. The relation between $T_{cwe}$ and $T_{cwi}$ is linear, since ($T_{cwe}-T_{cwi}$) is constant with no effect of $TTD_C$. $T_C$ and $T_{cwi}$ are linearly related, with approximately 1°C difference in $T_C$ for subsequent values of $TTD_C$ for constant value of $T_{cwi}$.

The results show that the condensate saturation pressure Pc increases by 0.000249, 0.000512, 0.001381, and 0.003135 MPa, for increases in $T_{cwi}$ of 1, 2, 5, and 10°C, respectively. As $P_C$ increases, the enthalpy of the extracted steam from low pressure turbines increases and consequently $\dot{W}_{net}$ and $\eta_{th}$ decrease. Figure 4 demonstrates variations of $\dot{W}_{net}$ and $\eta_{th}$ with $T_{cwi}$. Decreases in $\dot{W}_{net}$ by 0.465, 0.9312, 2.329, and 4.664%, are due to increases in $T_{cwi}$ of 1, 2, 5, and 10°C, respectively. The respective reductions in $\eta_{th}$ are 0.16, 0.31, 0.78, and 1.56%.

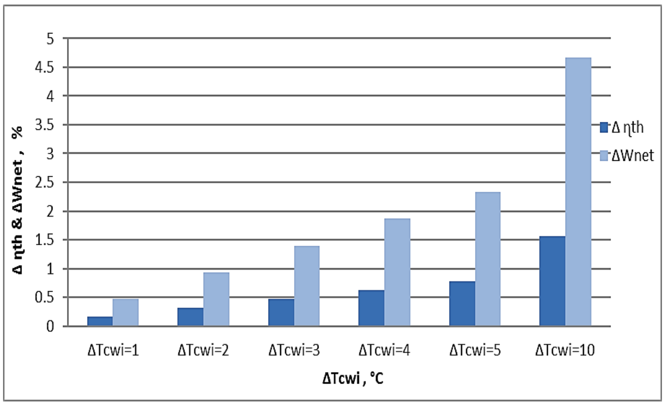



**Figure 4:** The impact of $T_{cwi}$ on $\dot{W}_{net}$ and $\eta_{th}$

### 4.3 The Effect of Condenser Coolant Fouling on $\dot{W}_{net}$ and $\eta_{th}$

The model calculated decreases in the inside overall heat transfer coefficient, $U_i$ of 27, and 220.8 W/m²K for increases in $F_i$ of 0.00002, and 0.0002 m²K/W, respectively for an assigned $T_{cwi}$. The outside overall heat transfer coefficient, $U_o$ decreases by are 24, and 200.3 W/m²K for $F_i$ increases of 0.00002, and 0.0002 m²K/W, respectively for constant $T_{cwi}$.

The condenser temperature, $T_C$ increases by 0.3 and 3 °C for $F_i$ increases by 0.00002, and 0.0002 m²K/W, respectively for a given $T_{cwi}$.

The condenser vacuum pressure $P_C$ increases by 0.000073 and 0.000787 MPa as $F_i$ increases by 0.00002, and 0.0002 m²K/W, respectively for a fixed $T_{cwi}$.

The negative impact of $F_i$, on $\dot{W}_{net}$ and $\eta_{th}$ is illustrated in Figure 5. It is seen that $\dot{W}_{net}$ decreases by 0.139 and 1.397% due to increases in $F_i$ by 0.00002, and 0.0002 m²K/W, respectively for constant $T_{cwi}$., whereas $\eta_{th}$ decreases by 0.05 and 0.47% for increases in $F_i$ of 0.00002, and 0.0002 m²K/W, respectively for constant $T_{cwi}$.

Any increase in the fouling of condenser cooling seawater causes an increase in the fouling thermal resistance of the condenser and this leads to reductions in $U_i$ and $U_o$. This deterioration in condenser overall heat transfer coefficients decreases the heat rejection from steam, thus increasing the exhaust temperature and pressure, leading to a reduction in the plant $\dot{W}_{net}$ and $\eta_{th}$.

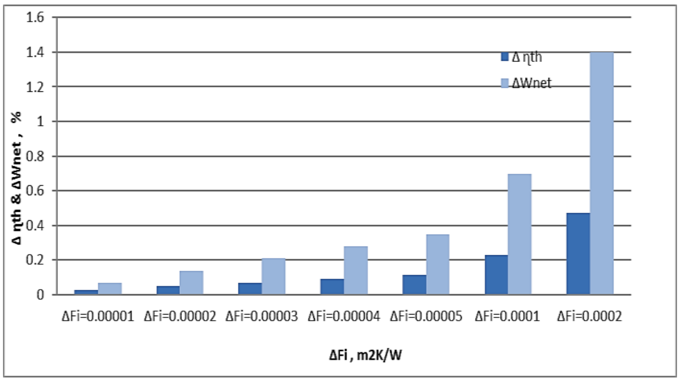



**Figure 5:** The effect of $F_i$ on $\dot{W}_{net}$ and $\eta_{th}$

### 4.4 The Effect of Condenser Coolant Salinity on $\dot{W}_{net}$ and $\eta_{th}$

Changes in the salinity affect many properties of saline water. The model results indicate that $\rho_{sw}$ increases by 8 and 79 kg/m³ as $S_p$ increases by 10, and 100 g/kg, respectively. $C_{P,sw}$, decreases with $S_P$ increase, and decreases as $T_{cwi}$ increases until the salinity reaches 20%, afterwards $C_{P,sw}$ increases with $T_{cwi}$ increase . $C_{P,sw}$ decreases by 0.063 and 0.535 kJ/kg.K when $S_p$ increases by 10, and 100 g/kg, respectively for constant $T_{cwi}$.

$\mu_{sw}$ goes up to 0.000026 and 0.000364 kg/m.s for $S_p$ increases by 10, and 100 g/kg, respectively for a given $T_{cwi}$.

$k_{sw}$ decreases by 0.0006 and 0.0058 W/m.K, as $S_p$ increases by 10, and 100 g/kg, respectively for fixed $T_{cwi}$.

$h_i$ decreases by 42 and 484 W/m²K, for increases in $S_p$ by 10 and 100 g/kg, respectively for unchanged value of $T_{cwi}$. Changes in $h_i$ vary overall heat transfer coefficients of the heat exchanger. The present model shows that both $U_i$ and $U_o$ decrease as $S_P$ increases, and increase as $T_{cwi}$ increases. $U_i$ and $U_o$ decrease by 2 and 22 W/m²K, and by 2 and 20 W/m²K, respectively for respective $S_p$ increases by 10 and 100 g/kg for constant $T_{cwi}$. The decrease in overall heat transfer coefficients of the condenser is mainly responsible for lowering the output power and thermal efficiency of the plant.

Figure 6 reveals that $\dot{W}_{net}$ decreases with $S_P$ and $T_{cwi}$ increase. $\dot{W}_{net}$ decreases by about 0.0465 and 0.465% of nominal power for $S_p$ increases by 10 and 100 g/kg, respectively at constant $T_{cwi}$. Figure 7 displays that $\eta_{th}$ decreases as $S_P$ and $T_{cwi}$ increase. $\eta_{th}$ decreases approximately by 0.02 and 0.16% for $S_p$ increases of 10, and 100 g/kg, respectively for one value of $T_{cwi}$.

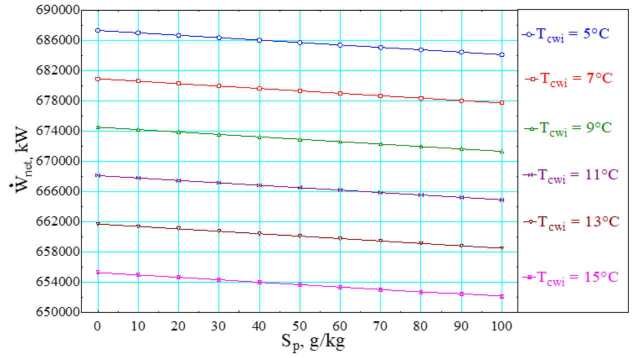



**Figure 6:** Variations of $\dot{W}_{net}$ with $S_p$ for different $T_{cwi}$

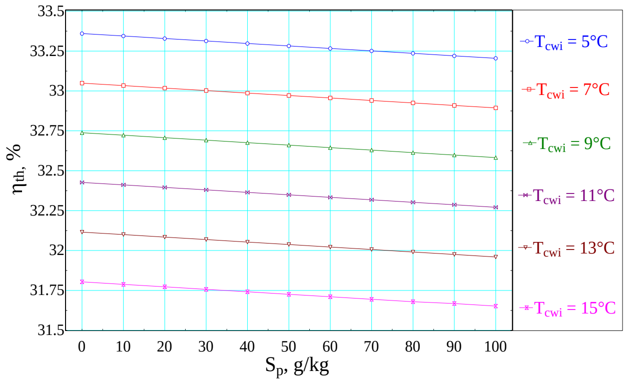



**Figure 7:** Variations of $\eta_{th}$ with $S_p$ for different $T_{cwi}$

The decrease in $h_i$ leads to reductions in $U_i$ and $U_o$, hence increases in the temperature and pressure of the turbine exhaust. Accordingly, reductions in the output power and the thermal efficiency of the plant take place. The results show that the increase in $\mu_{sw}$, due to salinity increase has the worst adverse effect on $\dot{W}_{net}$ and $\eta_{th}$ of the plant, while the increased $\rho_{sw}$ accounts for the least effect. $\mu_{sw}$ increase from 0.001306 - 0.001627 kg/m.s, decreases $\dot{W}_{net}$ by about 1122 kW, and $\eta_{th}$ by about 0.05 %, while an increase in $\rho_{sw}$ from 999.5-1078 kg/m$^3$, decreases $\dot{W}_{net}$ by about 257 kW, and $\eta_{th}$ by about 0.012 %.

### 4.5 The Combined Effect of Condenser Coolant Temperature, Fouling, and Salinity on $\dot{W}_{net}$ and $\eta_{th}$

All characteristics of cooling seawater can change all together and not separately. Therefore, it is important to find out the combined effect of changes in temperature, fouling, and salinity of seawater on the thermal performance of the plant. The following results are for the model data values given previously in Table 3. Figure 8 represents that $\dot{W}_{net}$ decreases with increasing $T_{cwi}$, F, and Sp. $\dot{W}_{net}$ goes down by 0.6704 % of nominal power for an increase in $T_{cwi}$ by 1 °C, fouling factor of seawater, $F_i$ and treated boiler feed water, $F_o$ by 0.00002 and 0.00001 m$^2$K/W, respectively, and $S_p$ by 10 g/kg. For high increases of $T_{cwi}$ b y 10 °C, $F_i$ and $F_o$ by 0.0002 and 0.0001 m$^2$K/W, respectively, and $S_p$ by 100 g/kg, $\dot{W}_{net}$ decreases by as high as 6.7095 % of nominal power. As shown, $\eta_{th}$ decreases by 0.22 % for increases in $T_{cwi}$ by 1 °C, $F_i$ and $F_o$ by 0.00002 and 0.00001 m$^2$K/W, respectively, and $S_p$ by 10 g/kg. For top increases of $T_{cwi}$ by 10 °C, $F_i$ and $F_o$ by 0.0002 and 0.0001 m$^2$K/W, respectively, and $S_p$ by 100 g/kg, $\eta_{th}$ decreases by a high value of 2.24 %.

Quite large losses in both of $\dot{W}_{net}$ and $\eta_{th}$ can result from combined variations in the condenser cooling sea water temperature, fouling and salinity. Large deteriorations can be achieved for extreme increases in coolant conditions, with reductions in $\dot{W}_{net}$ and $\eta_{th}$ of 6.7095 % and 2.24 %, respectively. Such values can be realized with regard of the adverse global environmental changes. On top of that CANDU plants have lower efficiency than BWR and PWR NPPs. Any decrease in $\eta_{th}$ is accountable.

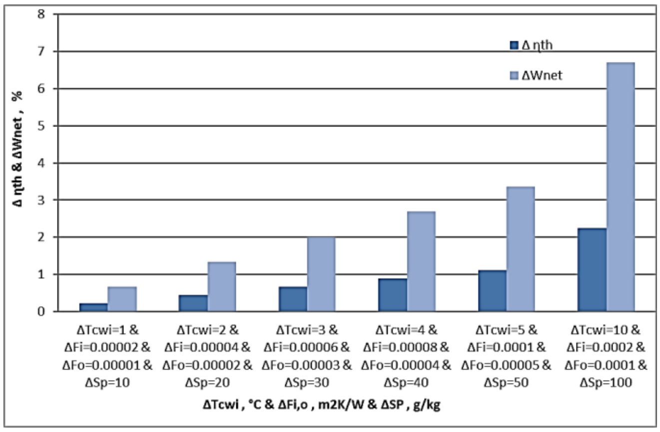



**Figure 8:** The combined effect of $T_{cwi}$, $F_i$, $F_o$ and $S_p$ on $\dot{W}_{net}$ and $\eta_{th}$

### 4.6 The Effect of Condenser Performance on $\dot{W}_{net}$ and $\eta_{th}$

Calculations are conducted to determine the effects of changing the condenser efficiency, $\eta_C$ from 40 – 100 %, on the condenser loss factor, LF, exhaust steam pressure $P_C$ and temperature $T_C$, on $\dot{W}_{net}$ and $\eta_{th}$ of the plant. Figure 9 depicts that $T_C$ and $P_C$ decrease with increasing $\eta_C$. For an increase in $\eta_C$ from 40 – 100 %, $T_C$ decreases by about 15 $^0$C and $P_C$ decreases approximately by 0.00254 MPa. This reflects positively on $\dot{W}_{net}$ and $\eta_{th}$.

When the exhaust steam temperature goes down, the amount of heat transferred to cooling seawater increases and this leads to a reduction in the loss factor. As presented in Figure 10, an increase in $\eta_C$ from 40 – 100 % results in reducing LF by about 0.046, which is in favor of the plant's performance.

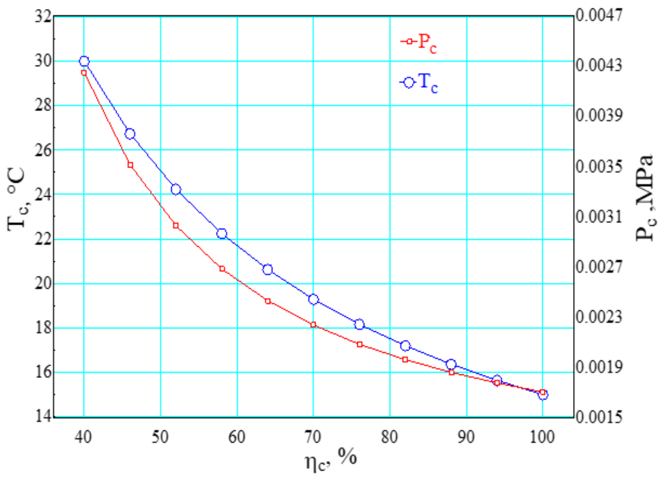



**Figure 9:** Variations of Tc and Pc with $\eta_C$

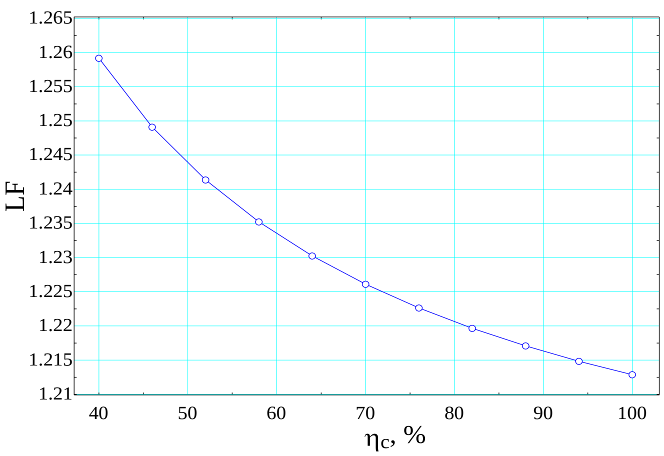



**Figure 10:** Variations of LF with $\eta_C$

Figure 11 gives the effect of the condenser efficiency $\eta_C$ on $\dot{W}_{net}$ and $\eta_{th}$ of the plant. It is clearly seen that increasing $\eta_C$ increases both $\dot{W}_{net}$ and $\eta_{th}$. For an increase in $\eta_C$ from 40 – 100 %, $\dot{W}_{net}$ increases by 48207 kW, i.e. by 7.0136 %, and $\eta_{th}$ increases by 2.34 %.

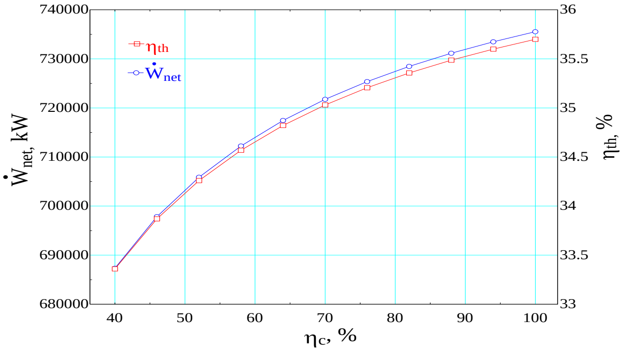



**Figure 11:** Variations of $\dot{W}_{net}$ and $\eta_{th}$ with $\eta_C$

## 5. CONCLUSIOS

The main findings of this research are:

1- An increase of 1 $^0$C in the temperature of the coolant seawater can result in decreases of 0.465 and 0.16 % in the power output and the thermal efficiency, respectively.

2- The output power and the thermal efficiency are reduced by 1.397 and 0.47 %, respectively for an increase in the fouling factor of the condenser cooling seawater of 0.0002 m$^2$K/W.

3- Many thermo physical properties of seawater are affected by changes in its salinity. The end effect is on the condenser overall heat transfer coefficients. An increase in the condenser cooling seawater salinity of 100 g/kg, decreases the power output and overall thermal efficiency by approximately 0.465 and 0.16%, respectively.

4- The output power and overall thermal efficiency decrease by 6.7095 and 2.24 %, respectively for combined extreme increases in the condenser cooling seawater temperature, fouling factor of seawater, fouling factor of treated boiler feed water, and salinity of 10 °C, 0.0002 m$^2$K/W, 0.0001 m$^2$K/W, and 100 g/kg, respectively.

5- A rise in the condenser efficiency from 40 - 100 % increases the output power and thermal efficiency by 7.0136 and 2.34 %, respectively. Thus, environmental factors should be used in the design of condensers to compensate for the expected decrease in performance under actual operating conditions for the life time of the NPP. Condenser design and proper operation are rather important.

The highest adverse impact on the efficiency is due to increases in the condenser seawater cooling temperature, followed by fouling then salinity. Decreases in the thermal efficiency resulting from power losses are significant, and cannot be tolerated, especially for CANDU NPPs, since their efficiencies are lower than BWR and PWR NPPs. Engineers and scientists are working tirelessly and large funds are being spent in order to increase the efficiency by one or less percent. The combined deteriorating impacts of all considered factors together are likely to take place rather than their individual effects. Moreover, the extreme values could be achieved in the light, for instance of increased global climate impacts, increasing maritime trade, human activities, and increased water desalination.

The site of NPPs built on seas should be carefully selected, for minimum changes in the characteristics of seawater. However, undesired impacts on the thermal performance of the plant are unavoidable along the operation time, and these should be minimized or halted if possible. It is recommended to launch an additional devise in new NPPs to keep the seawater condenser coolant temperature, fouling, and salinity within acceptable design values. The temperature could be controlled by a heat exchanger working in conjunction with a refrigeration unit, or via blowing down hot water and adding fresh cold water. Fouling could be controlled chemically, mechanically. Salinity may be brought down by adding soft water or/and chemicals. All can be contained in one automatically controlled unit with specific proper controlling sensors for each property. The cost of such additional component would be more than compensated by lowering the high maintenance costs in NPPs, fuel cost, waste disposal, and keeping the plant performing within its rated efficiency.

## DECLARATION OF CONFLICT OF INTEREST

The authors declare that they do not have any actual or potential conflict of interest including any financial, personal or other relationships with other people or organizations regarding publishing this submitted work.

## STATEMENT OF FINANCIAL POSITION

This study received no particular financing from governmental, private, or non-profit sponsors.

## AUTHORS CONTRIBUTION STATEMENT

S. Ibrahim: Conceptualization, Methodology, Writing- Original draft preparation, Visualization, Investigation, Supervision, Validation, Writing-Reviewing and Editing. I. Aggour: Data curation, Software, Plotting Figures. The authors reviewed the paper contents and approved the final version of the manuscript.

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