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Analysis, Simulation and Feasibility Study of a Direct Compression Desalination System

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Abstract. *Natural resources are not being able to meet the rapidly growing demand for fresh water. A method of desalination by direct compression of water vapor is analyzed, simulated and studied for feasibility. The Specific Energy Consumption (SEC) of the desalinated water is found to decrease with increasing mass flow rate and temperature of inlet seawater, increasing area of the heat exchanger (HX) and decreasing number of tubes in the HX. For a system with 760 kg/s as inlet seawater mass flow rate, 22°C as inlet seawater temperature, 1000 m^2 of HX area, and 3000 tubes in HX the SEC is found to be as low as 5.1 kWh/ton. If the initial capital costs are also considered then this set of parameters may not be the most desirable. However, the system seems to be economically viable to supplement the potable water.*

Keywords: *Water desalination, Direct mechanical compression, Simulation, Condenser, Specific energy consumption*

1. Introduction

Desalination of seawater in coastal areas, apart from becoming increasingly feasible, is proving to be a cost-effective solution in some situations, to the problem of water scarcity in the domestic, industrial and agricultural sectors. An intensive effort is needed to make the seawater desalination technologies much more efficient and competitive. This paper deals with one such technology.

The existing desalination technologies in the world can be classified into two broad categories, membrane and thermal technologies. The reverse osmosis method falls under the former category. Membrane methods are very good for desalination of low concentration salt water as their efficiency is concentration dependent. On the other hand the performance of thermal methods is not affected much by the concentration. The product water is also very pure. Hence these are better at higher concentrations. The direct compression desalination system studied here falls under the thermal category.

Mechanical Vapor Compression (MVC) desalination system has been analyzed and modeled for parametric study (Aybar, 2002). The Specific Energy Consumption (SEC) reported by the author is about 11.5 kWh/ton. Design parameters and operational features have been summarized with a case study on a 500 m^3/day MVC plant with a SEC of 10.4 – 11.2 kWh/ton (Veza, 1995). A performance evaluation of a MVC system is presented for varying brine concentration, recirculation rate and compressor speed (Bahar *et al.*, 2004). The reported performance is poor at 258kWh/ton. The variation of the SEC reported by different people results from a number of factors including differences in the size and configuration of the units, technological advances, and the quality of the feed stream being treated. There are also variations in what is included in the energy calculation. In some cases, authors have declined to include thermal energy obtained from waste heat sources as part of the calculation, and instead only account for energy that is used in addition to this heat or that is diverted from the main process (usually electric power generation) as it is typically run (Miller, 2003). This paper presents an analysis in the lines of the work in (Aybar, 2002) and (Veza, 1995). A theoretical study of the variation of SEC with inlet mass flow rate of sea water, inlet temperature of sea water, area of the heat exchanger and number of tubes in the heat exchanger is done.

2. Working principle

Figure 1 shows the schematic of the direct compression desalination system considered for study. Inlet sea water at a flow rate of m_1 kg/s and temperature $T_1{}^0$C is sent through the tube side of the heat exchanger, where it absorbs the heat given out on the condensing side and warms up to $T_2{}^0$C. It then enters the flash tank at a flow rate of m_2 kg/s ($m_2 = m_1$), from where m_4 kg/s is flashed as vapor at a temperature $T_4{}^0$C due to the low pressure p_4 kPa created by the compressor. T_4 is the saturation temperature at p_4. The remaining m_3 kg/s is sent back to the sea at a temperature of $T_3{}^0$C ($T_3 = T_4$). The flashed vapor is compressed to a higher pressure p_5 kPa and temperature $T_5{}^0$C. It is then condensed at $T_6{}^0$C which is the saturation temperature at p_6 kPa ($p_5 = p_6$) on the shell side of the heat exchanger to come out as product fresh water at a rate of m_6 kg/s ($m_4 = m_6$).

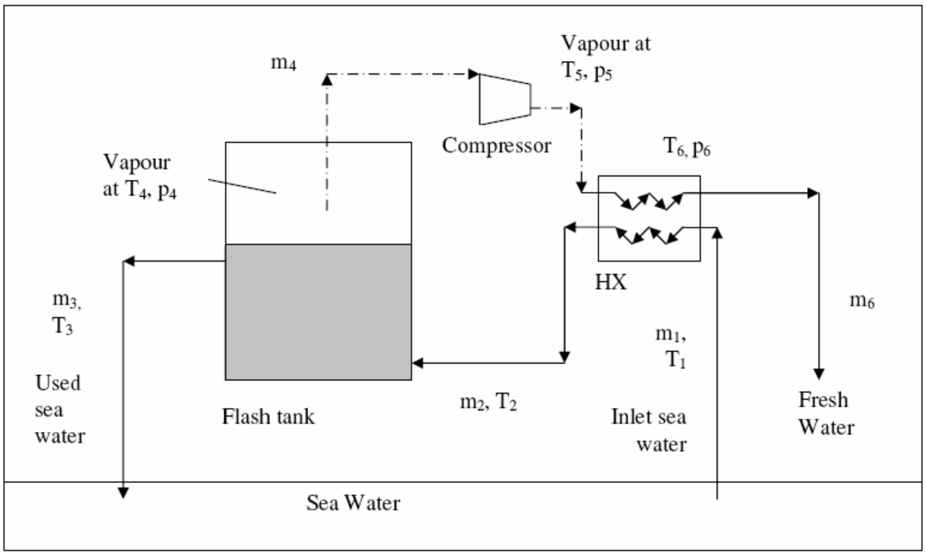

Figure 1. Schematic diagram of the direct compression desalination system

3. Thermodynamic Analysis

Thermodynamic analysis is carried out by varying the condensing temperature (T_6) and the flash tank evaporation temperature (T_4). The compressor is required to increase the saturation temperature from T_4 to T_5. The following equations are used.

Saturation pressure – temperature relation (AMS Glossary):

$$p = 0.6112\exp\left(\frac{17.67T}{T+243.5}\right) \qquad (-35°C < T < 35°C) \tag{1}$$

where, T is temperature in °C and p is vapour pressure in kPa.

Isentropic reaction equation:

$$\left(\frac{p_4}{p_5}\right)^{\frac{1-k}{k}} = \left(\frac{T_5+273.15}{T_4+273.15}\right) \tag{2}$$

where, p is in kPa and T is in ^{0}C and k is the heat capacity for water vapour (1.33).

Specific Energy Consumption (SEC): The *Specific Energy Consumption* is the energy consumed per ton of water desalinated (kWh/ton). Only the compressor work is considered

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$$SEC = R\times\frac{k}{k-1}\times(T_5-T_4)\times\frac{1000}{3600\times m_6} \tag{3}$$

where, R is the gas constant of water vapor (0.461 kJ/kg/K).

The independent variables are condensing temperature (T_6) and evaporating temperature (T_4). Their difference ($d = T_6 - T_4$) is also another useful parameter. The dependent parameters are SEC, p_4, p_5, p_6, pressure ratio (p_5/p_4), pressure difference ($p_5 - p_4$) and *tip speed (V)* which is calculated using the following equation (Stoecker and Jones, 1982), where, Δh is the change in enthalpy .

$$V = \sqrt{1000\Delta h} \tag{4}$$

4. Simulation Model

Modeling of the system is done using mass and enthalpy balance equations for flash tank, governing equation for compressor, and heat transfer equations for the heat exchanger.

<u>Flash tank</u>

Mass balance:

$$m_2 = m_3 + m_4 \tag{5}$$

Enthalpy balance:

$$m_2h_2 = m_3h_3 + m_4h_4 \tag{6}$$

where, $h_2 = C_{sea\ water} * T_2$, $h_3 = C_{sea\ water} * T_3$, $h_4 = 2501 + 1.84 * T_4$.
Saturation pressures are obtained from Eq. (1).

Compressor

The *compression* is assumed to be *isentropic*. Therefore Eq. (2) is used to relate the pressures and temperatures before and after the compressor.
The following *governing equation* is assumed for *compressor*.

$$m_4 = (0.8) + (0.2 * (2 - \frac{p_5}{p_4}))^{0.5} \tag{7}$$

The *Specific Energy Consumption (SEC)* is calculated as given in Eq. (3) since compressor is the major work input. Pumping losses are dealt with, separately, later.

Heat exchanger (HX)

The *overall heat transfer coefficient* is calculated considering the heat transfer coefficient on the condensing side (h_{cond}) and water side (h_w) (Stoecker and Jones, 1982):

$$\frac{1}{U} = \frac{1}{h_{cond}} + R_m + \frac{OD}{ID}\frac{1}{h_{ff}} + \frac{OD}{ID}\frac{1}{h_w} \tag{8}$$

where, OD is outer diameter (0.018m) and ID is inner diameter (0.016m) of the tubes in the HX. R_m is the resistance of the metal (titanium) wall (3.5294e-005 m^2K/W) and the fouling factor (h_{ff}) is (2.8409e+003 W/m^2K). The condensing side heat transfer coefficient (h_{cond}) depends on the number of tubes in the HX, Δt (the temperature difference between the vapor and the wall of tube) and other properties of water. The heat transfer coefficient on the water side (h_w) depends on the velocity of water in the tubes and other properties of water.
Energy balance equation:

$$Q = m_6((2501 + 1.84T_5) - C_{sea\,water}T_6) = m_1 C_{sea\,water}(T_2 - T_1) \tag{9}$$

Heat transfer equation:

$$Q = U_o A(LMTD) \tag{10}$$

The *frictional loss* occurs mainly in the heat exchanger and is calculated by,

$$frictional\,loss = m_1 g H_f \;(in\,kW) \tag{11}$$

where,

$$H_f = \left(\frac{flv^2}{2gd}\right)(in\,meters) \tag{12}$$

f is Darcey friction factor (0.05), l is length of tube, v is velocity of sea water, g is gravitational constant = 9.81 m/s^2, d is inner diameter in m

5. Simulation Method

The independent parameters are inlet mass flow rate (m_2), inlet sea water temperature (T_4), the area of the heat exchanger (A) and the number of tubes in the heat exchanger (n). The dependent parameters are Specific Energy Consumption (SEC) and mass flow rate of desalinated water (m_6).

1. Specify the independent parameters m_1, T_1, A and n.
2. Assume T_4 and T_6
3. Calculate the saturation pressures p_4, p_6 ($p_6 = p_5$) at T_4 and T_6 respectively using Eq. (1).
4. Calculate m_4 using Eq. (7).
5. Calculate T_5 using Eq. (2).
6. Calculate T_2 using Eq. (9).
7. Calculate Q using Eq. (9).
8. Calculate LMTD

$$LMTD = \frac{T_2 - T_1}{\ln\left(\frac{T_6 - T_1}{T_6 - T_2}\right)} \tag{13}$$

9. Assume Δt (the difference in temperature between vapor and wall of the tubes).
10. Calculate overall heat transfer coefficient using Eq. (8), which depends on Δt.
11. Calculate Q using Eq. (10).
12. Estimate Δt from the calculated Q (in Step 11) using $Q = h_{cond}A\Delta t$ and compare with the assumed value. Iterate Steps 10 to 12 by changing Δt, for the required accuracy.
13. Compare Q (in Step 7) and Q (in Step 11). Iterate Steps 3 to 13 by changing T_6, for the required accuracy.
14. Estimate T_4 by Eq. (6). Iterate Steps 3 to 14 by changing T_4, for the required accuracy.
15. When the right temperatures are arrived at, calculate the SEC using Eq. (3).

6. Results and Discussion

Results obtained from the thermodynamic analysis and simulations are discussed in the following sections.

Thermodynamic Analysis

Figure 2 shows the variation of Specific Energy Consumption (SEC) with condensing temperature T_6. SEC decreases with increasing T_6, for a given d (d = T_6 - T_4). As T_6 increases, the pressure ratio decreases for the same d, which decreases ($T_5 - T_4$) as per Eq. (2), thereby, decreasing the SEC as per Eq. (3). Further, as expected the figure illustrates that for a given T_6, SEC increases with increasing d.

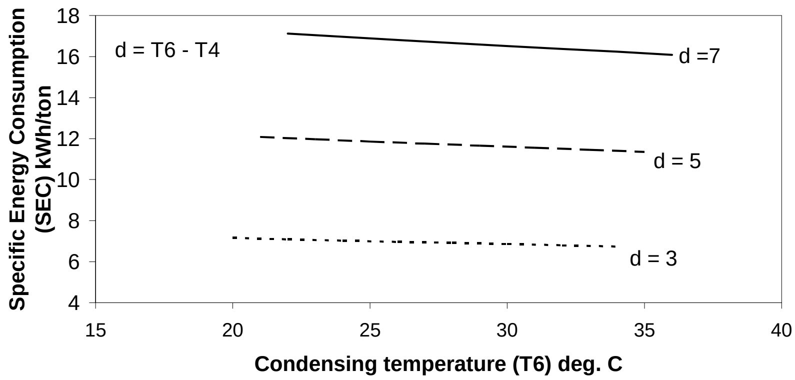

Figure 2. Variation of SEC with T_6 for various d

Figure 3 shows that the tip speed decreases with increasing condensing temperature. It is expected, since tip speed is also proportional to ($T_5 - T_4$) just like in the case of SEC explained above. For d = 3, 5 and 7 the tip speeds are in the range of 150, 200 and 240 m/s respectively. Assuming a rotational speed of 60 rps the corresponding diameters are 0.8, 1 and 1.27 m. The diameters required seem to be practically possible, and the tip speeds are within the usual limitation of 300 m/s [11]. Figure 4 shows the variation of difference in pressures at the condensing and evaporating temperatures with increasing condensing temperature. The difference in pressures at the condensing and evaporating temperatures increases with increasing T_6 for the fixed d.

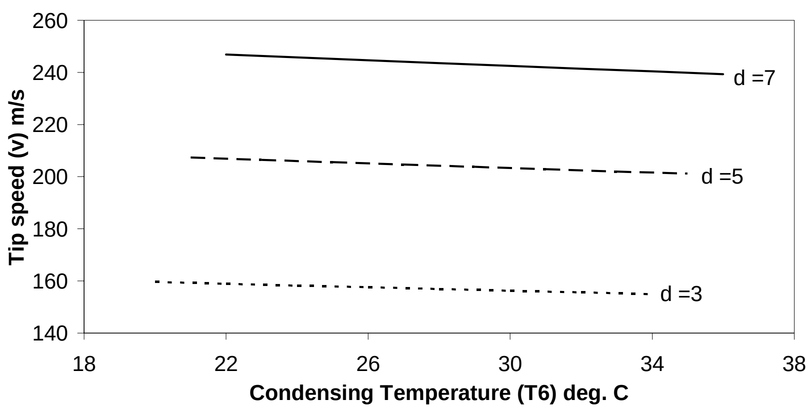

Figure 3. Variation of tip speed with T_6 for various d

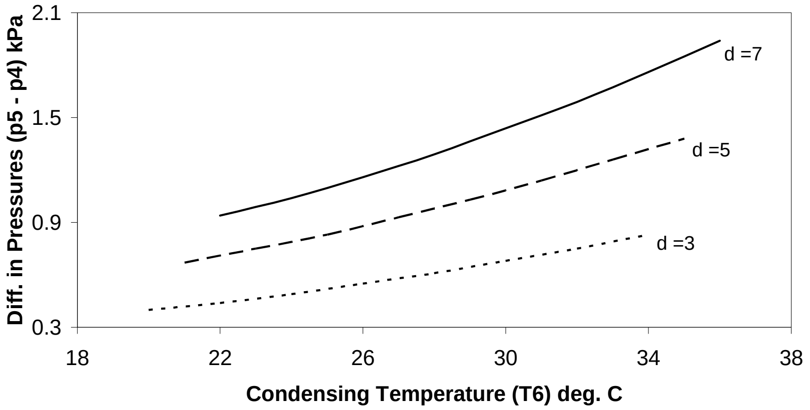

Figure 4. Variation of difference in pressure with T_6 for various d

Simulation

In Fig. 5 all independent parameters are scaled down to vary between 1 and 5. Inlet mass flow rate varies from 220 to 380 kg/s. Inlet temperature varies from 20 to 24^{0}C. Area of heat exchanger varies from 200 to 400 m^2. Number of tubes varies from 300 to 700. When a parameter is changed the other parameters are fixed at typical values which are taken as 220 kg/s, 22^{0}C, 300 m^2 and 700. The figure shows that SEC decreases with increasing inlet sea water mass flow rate. With higher flow rate, the overall heat transfer coefficient U of the heat exchanger is higher; therefore, the LMTD decreases for the same heat exchanged or for the same yield. Further, the HX exit water temperature (T_2) decreases due to increased flow. These two together contribute to lowering the condensing temperature (T_6) which in turn results in a lower T_5. SEC decreases since it depends on ($T_5 – T_4$).

Figure 5 also shows that SEC decreases with increasing inlet water temperature. An increase in the water inlet temperature basically increases T_4. SEC decreases as it depends on (T_5 – T_4). Further, the figure shows that SEC decreases with increasing area of the heat exchanger (HX). An increase in the area, results in an increased UA of the HX, therefore LMTD decreases for the same Q, which leads to lower T_5, decreasing the SEC. SEC increases with increasing number of tubes in the HX. This happens because, with higher number of tubes, the velocity of sea water in the tubes decrease, which in turn reduces U and increases LMTD, resulting in a higher T_5 for the same Q thereby increasing the SEC.

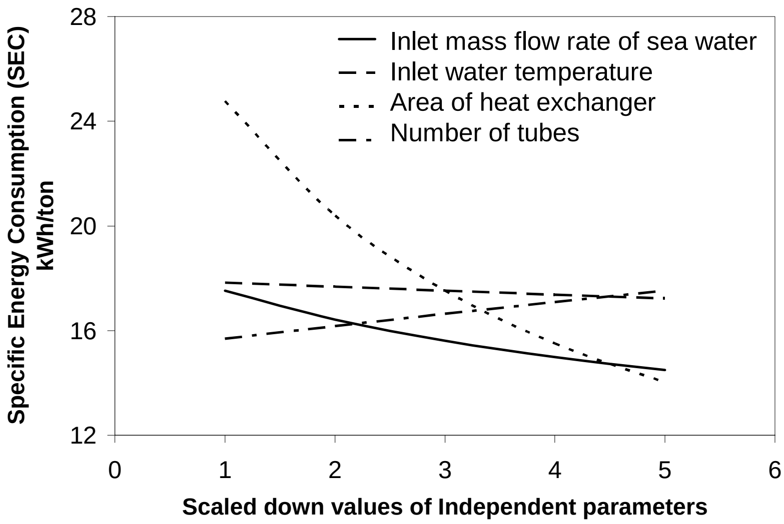

Figure 5. Variation of SEC with system parameters

Thus it is shown that a decrease in SEC can be achieved by either an increase in inlet mass flow rate of water, or increase in the inlet temperature, or increase in the area or a decrease in the number of tubes in the HX. Increasing the flow rate results in an increase in velocity if the number of tubes is fixed. Therefore, the number of tubes is fixed at a very large value of 3000 restricting the velocity of water in the tubes to 1.7 m/s. The area is put as 1000 m^2 and inlet temperature was put at 22^{0}C. The simulation was now run for a range of flow rate values from 220 to 960 kg/s. Figure 6 shows that at lower flow rates, the benefit of increasing the flow rate is higher than that at higher flow rates. From the graph, we see that at around 760 kg/s the graph tends to flatten out because the inlet temperature is fixed, and LMTD cannot reduce indefinitely and at the same rate as in the beginning. Therefore, the ideal flow rate was fixed at 760 kg/s for further simulations.

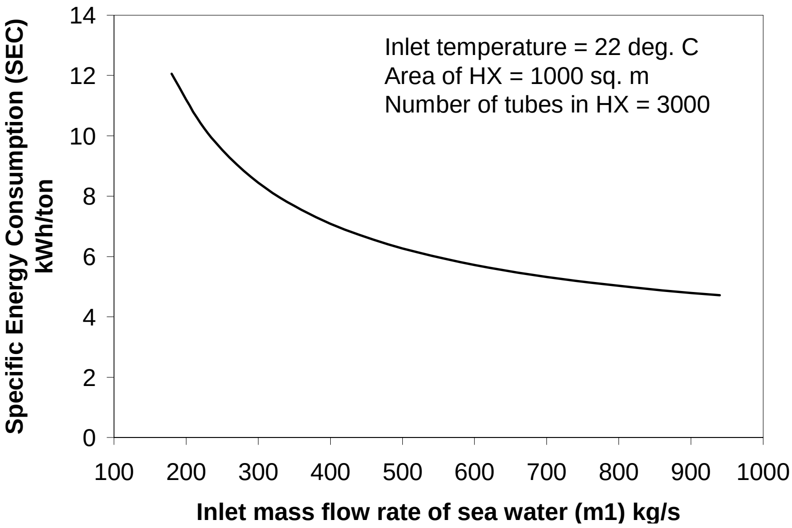

Figure 6. Influence of inlet mass flow rate (m1) on SEC

A simulation is performed for a range of inlet temperatures from 18 to 34^{0}C. The SEC showed a continuous decrease, though the decrease is not much. Further, it would not make economic sense to heat up the inlet water to reduce the SEC; therefore, inlet temperature is fixed at the normal temperature of 22^{0}C. Figure 7 shows the result of the simulation where the area was varied, with inlet flow rate at 760 kg/s and inlet temperature at 22^{0}C. The graph shows a great reduction of SEC for initial increments in area, but flattens out after around 1000 m^2. Again, as stated above the reason is that the LMTD cannot reduce indefinitely at the same rate as in the beginning. Thus this graph helps in fixing HX area at 1000 m^2.

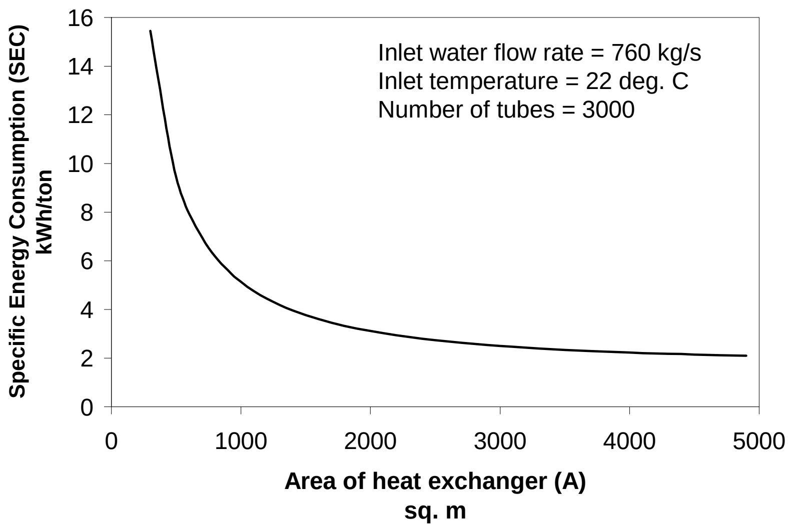

Figure 7. Influence of area of HX (A) on SEC

A simulation where only number of tubes is varied was also carried out, for the above fixed values of the other independent parameters. It showed that SEC increases with increase in number of tubes. But, it is desirable to increase the number of tubes so as to reduce the velocity and the frictional losses and the length of the HX. Therefore, a graph with a combined plot of frictional losses, compressor work and the total energy was plotted with varying number of tubes (Fig. 8). It is seen that an optimum number of tubes in HX exists which is about 4750 for the selected operating parameters in the present study.

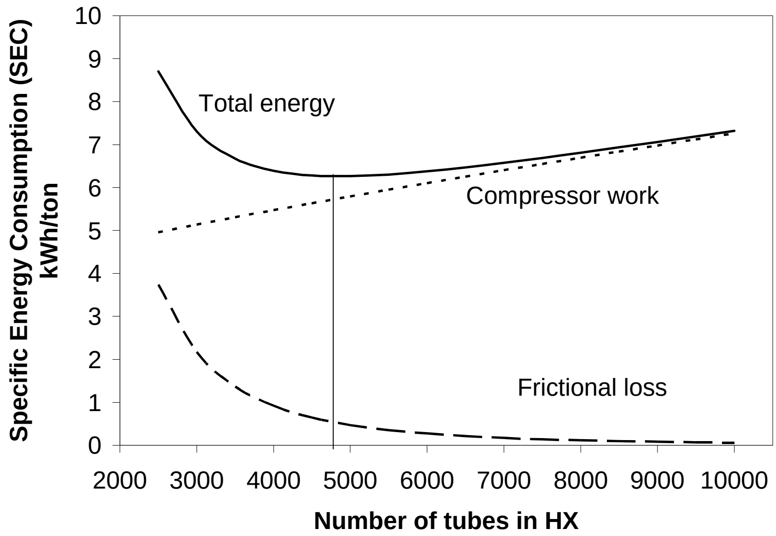

Figure 8. Optimal number of tubes in HX

7. Conclusions

In this paper, a direct compression desalination system was analyzed, simulated and studied for feasibility. It was found that Specific Energy Consumption (SEC) decreases with increasing mass flow rate of inlet sea water, increasing inlet water temperature, increasing area of the heat exchanger (HX) and decreasing number of tubes in the HX. For a system with 760 kg/s as inlet flow rate, 22^0C as inlet water temperature, 1000 m^2 of HX area, and 3000 tubes in HX the SEC is found to be as low as 5.1 kWh/ton. If the initial capital costs are also considered then this set of parameters may not be the most desirable. The SEC increases to 17.5 kWh/ton when the inlet flow rate is 220 kg/s as flow rate, inlet temperature is 22^0C, HX area is 300 m^2, and the number of tubes in the HX is 700. This value of SEC is found to be slightly on the higher side when compared with those reported for the existing methods.

8. References

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