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Second-Law Analysis of a Double-Effect Evaporator with Thermal Vapor Compression

Ali Snoussi* and Maha BenHamad

Laboratory of Applied Thermodynamics, National Engineering School of Gabes, University of Gabes, Omar Ibn El Khattab St, 6029 Gabes, Tunisia

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ABSTRACT

In this paper, we present a steady-state analysis of a double-effect evaporator with thermal vapor compression (MED-TVC) installed in the Tunisian Chemical Group (GCT) factory. A thermodynamic model including mass and energy balances of the system is developed and integrated in a Matlab program. The model resolution yields to the determination of the operating parameters of the plant and the Gain Output Rate (GOR) was found to be roughly equal to 5. In a second step, the simulation results served to conduct a second law analysis of the unit. The performance criterion used in this analysis is the second law efficiency, i.e., the ratio of the least theoretical work of separation to the actual work input to the plant. The second law efficiency was found to be 2.4%. The distribution of the irreversibility between the different components of the plant was, in addition, assessed. As a conclusion, it was established that the most irreversibility occurs in the thermo-compressor which contributes with more than 50% to the global imperfection and which presents an exergy efficiency of less than 77%. The remaining irreversibility comes from the three exchangers (the two evaporators and the condenser) with an average contribution of 16%. As it is very difficult to introduce modifications into an existing unit, we assume that the importance of the results is not limited to the studied unit. They serve, rather, as an aid to the future design of a MED-TVC plant.

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*Corresponding Author
Email: ali.snoussi@enig.u-gabes.tn
Tel: (0216)20010075

1. Introduction

There are two main desalination technologies: thermal technology and membrane technology. In thermal technology, the principle is to evaporate seawater and then condense the produced vapor. The two principal thermal processes are multi-stage flash (MSF) and Multi-effect distillation with thermal vapor compression (MED-TVC). Membrane technology consists of passing forcibly seawater through the atomic-sized pores of a membrane surface, the typical membrane process is reverse osmosis (RO). Reverse osmosis holds more than 61% of the global market of desalination while the contribution of thermal processes is less than 34% with roughly 27% for MSF [1]. The most reason is that thermal processes require three times the energy required for RO desalination i.e. almost 17 kWh per m^3 of freshwater versus only 5 kWh/m^3. In addition, RO requires only electrical (pumping) energy while thermal plants require both thermal (heating) and electrical (pumping) energy and, consequently, special materials (pipes, vacuum system, etc.) and the associated problems (leakage, scale on tubes, etc.) that make its construction, land and maintenance costs considerably high. Nevertheless, thermal technology occupies, and can still occupy in the coming years, an important place in countries having a diversity of thermal energy sources. The disadvantage of high energy consumption may be overcome when steam is freely available as in co-generation plants or when sustainable solutions based on low-grade energies are well established. Besides, considering the quality of the produced water, thermal technology is more competent with a salinity of about 100 ppm compared to 400 ppm for RO. To reach the same quality, an RO process ought to be preceded by pre-treatment facilities and/or to be followed by a second set of membranes.

The energy merit of a thermal desalination plant is usually evaluated using the Gain Output Rate GOR, which is defined as the ratio of the energy required to evaporate the distillate and the heat input to the system. This parameter indicates how often do we take advantage of the energy effect but does not give any indication about the possibilities of improvement of the efficiency of the plant. A more rigorous approach in this direction is to use a second-law based criterion which [2-6] permits the evaluation of the exergy destruction within the plant. Physically, there is exergy destruction associated with any irreversible process (heat-transfer, mass-transfer, flow) and the value of the exergy destruction indicates the extent of this irreversibility. Besides, with a second-law based criterion, it would be possible to determine not only the global imperfection of the plant but also its distribution and to locate the weak points where enhancement is possible.

During the last decade, numerous papers have dealt with second law analysis of desalination plants [7-16]. Different second-law performance criteria, i.e. exergy efficiency, entropy generation number, second law efficiency, have been applied to different types of plants. The main conclusion was that the second law performances of desalination plants are small, always less than 10%. In addition, the effect of certain parameters (number of stages, season, etc.) on the performances has been assessed. Two particularly important works in this direction were the papers by Sharqawy *et al*. [10] and by Mistry *et al*. [11]. In these two complementary papers, the authors presented a detailed development of the exergy approach and conducted exergy analysis for various desalination technologies.

In Tunisia, the problems of water supply are giving more and more concern to desalination solutions. During the last two decades, a great number of RO desalination stations have been installed and thermal projects are under study. In parallel to this development, there is a growing research effort in the field of desalination [17-20]. The present work constitutes a continuation of a previous paper [19] where we dealt with the economic assessment of MED-TVC plants using the data of an existing triple-effect unit. Here, we present a second-law analysis of an existing double effect desalination plant with thermal vapor compression installed in the Tunisian Chemical Group (GCT) factory. A thermodynamic model based on mass and enthalpy balances is developed and integrated into a matlab program in order to determine the operating parameters in the different points of the unit. The second-law efficiency of the plant is then determined and the assessment of the contribution of the different components is conducted. Although it is difficult to introduce the suggested improvements into an existing plant, we assume that the results given from the second-law analysis are more advantageous not only against simple first law analysis but to some extent against economic analysis as actualized economic data are always unavailable and uncertain.

2. Description of the MED-TVC Plant

The main features of the unit are given schematically in Figure **1**. The seawater enters in the tubes of the condenser where it is heated to the desired Top Brine Temperature using the heat of condensation of the vapor issued from the second effect. At the exit of the condenser, the cooling water is sent back to the sea while the feed flow rate is distributed equally between the two evaporation effects. The energy input to the plant occurs in the vapor thermo-compressor where the external motive steam serves to entrain and compress a certain amount of vapor issued from the second effect. The discharge steam enters the tubes of the first effect where it condenses causing the seawater feed to heat up and to be partially evaporated. The latter vapor condenses at its turn in the tubes of the second effect so that to heat up the feed of the second stage and to evaporate it partially. The non-evaporated fraction of the feed, constituting the brine solution of the first effect, is cascaded to the second effect. The pressure is reduced gradually from one effect to the next, a flash evaporation occurs when the brine solution enters the second effect. Flash evaporation takes place also when the condensed steam of the first effect is cascaded to the second effect. A part of the vapor produced in the second effect is entrained by the thermo-compressor while the remaining part is condensed on the tubes of the condenser. The final brine solution is that extracted from the second stage while the distillate is collected from the condensates of the steam of the first stage, the vapor produced in the first stage and the vapor condensed in the condenser.

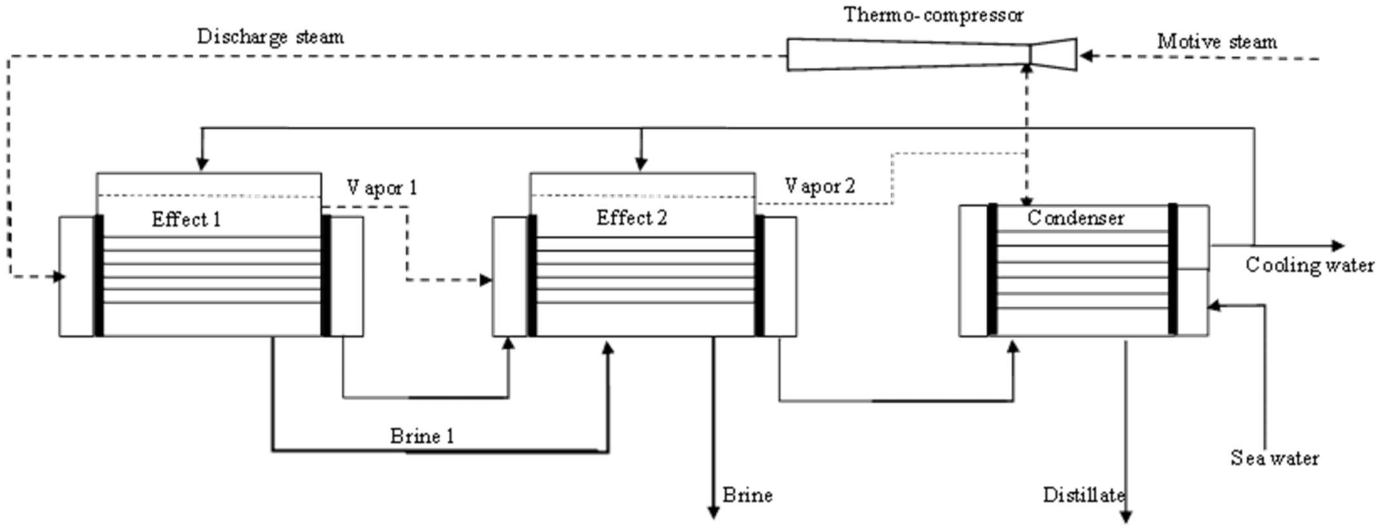

Figure 1: Schema of a double-effect desalination plant with thermal vapor compression.

3. Thermodynamic Modeling

The thermodynamic model we propose in this paper for the analysis of the second-effect desalination plant is a steady-state one that contains a number of assumptions:

- the two effects, the condenser and the thermo-compressor are assumed to be adiabatic,
- the distillate is supposed to be salt-free,
- the pressure drop is negligible anywhere in the plant,
- The effect of pressure on the properties of liquids is negligible.

The properties of the seawater (temperature T_{sw} and salinity x_{sw}) are always imposed. The same thing is to say for the properties (temperature T_m and pressure P_m) of the available motive steam. The top brine temperature T_f, which is the temperature of the feed seawater at the inlet of the effects, and the temperature of the discharge steam T_d are fixed so that to have as much as possible a low-temperature operating unit.

The temperatures of the two stages are then deduced:

$$T_d - T_1 = T_1 - T_2 = T_2 - T_f \tag{1}$$

The saturation temperature of pure water and the corresponding saturation pressure are calculated taking into account the boiling point elevation caused by the presence of salt

$$T_{vi} = T_i - BPE_i \quad P_i = P^0(T_{vi}) \tag{2-3}$$

3.1. Mass and Heat Balances

The seawater entering the plant is composed of the feed effectively used to produce distillate and cooling water:

$$W_{sw} = W_f + W_c \tag{4}$$

The feed flow rate has a fixed value while the cooling water flow rate changes (depending on the seawater temperature) so that to maintain a constant top brine temperature.

For the entire plant, the mass and energy balances are written as:

$$W_f = D + B \tag{5}$$

$$(W_f + W_c)h_{fsw} + W_m h_{gm} = W_c h_{ff} + (D + W_m)h_{fD} + Bh_{fB} \tag{6}$$

The feed flow rate to each stage is the same:

$$F = \frac{W_f}{2} \tag{7}$$

In the first effect, the mass and the enthalpy balances are as follows:

$$F = B_1 + D_1 \tag{8}$$

$$Fx_{sw} = B_1 x_1 \tag{9}$$

$$W_d \lambda_d = Fc_1(T_1 - T_f) + D_1 \lambda_1 \tag{10}$$

In the second effect, the mass and the enthalpy balances are as follows:

$$B_1 + F = B_2 + D_2 \tag{11}$$

$$Fx_{sw} + B_1 x_1 = B_2 x_2 \tag{12}$$

$$(D_1 \lambda_1 + D_1' \lambda_1') = Fc_2(T_2 - T_f) + B_1 c_1 (T_2 - T_1) + D_2 \lambda_2 \tag{13}$$

D_1' is the vapor flashed from the condensed steam of the first effect when it enters the second stage. This parameter is obtained from the following relations [15-16]:

$$D_1' \lambda_1 = W_d c(T_d - T_1') \quad T_1' = T_{v1} + NEA_1 \quad NEA_1 = 0.33 \frac{(T_d - T_{v1})}{T_{v1}} \tag{14-16}$$

In the condenser, the part of vapor that is not entrained by the thermo-compressor is condensed and sub-cooled. The condensed steam, accumulated from the two effects, is also sub-cooled in the condenser. The enthalpy balance of the condenser is then:

$$(W_f + W_c)c_{sw}(T_f - T_{sw}) = (D_2 - W_{ev})(\lambda_2 + c(T_{v2} - T_D)) + (D_1 + W_d)c(T_{v1} - T_D) \tag{17}$$

The entrainment ratio of the thermo-compressor is defined as:

$$Ra = \frac{W_m}{W_{ev}} \tag{18}$$

It may be calculated using the semi-empirical model proposed by El-Dessouky *et al*. [21]:

$$Ra \approx 0.235 \frac{(P_d)^{1.19}}{(P_{ev})^{1.04}} \left(\frac{P_m}{P_{ev}} \right)^{1.05} \tag{19}$$

The mass and energy balances of the thermo-compressor are then given by:

$$W_d = W_m + W_{ev} \tag{20}$$

$$W_d h_{gd} = W_m h_{gm} + W_{ev} h_{gev} \tag{21}$$

3.2. Second Law Relations

From a second law viewpoint, each stream is characterized by its exergy. Each process, of either heat or mass transfer, maybe also seen as a process of exergy transfer and is necessarily accompanied by exergy destruction. The object of the second law analysis is the assessment of global exergy destruction and its distribution among the different components of the plant. To conduct a second law analysis, we need first to decide regarding the exergy definition to use. Usually, authors limit the definition of exergy to its thermo-mechanical component, i.e. the maximum work obtained when the temperature and the pressure of the stream change to the "environment" temperature and pressure T_0 and P_0 with no change in the concentration. Indeed, when a stream is subject to composition variation, one must also take into account the chemical exergy, which is the maximum work obtained when the concentration of each substance in the stream changes to the concentration in the "environment" x_0 at the "environment" pressure and temperature (T_0, P_0). According to this definition, the specific exergy of a stream is [10]:

$$\begin{aligned} e = {} & (h - h(T_0, P_0, x)) - T_0(s - s(T_0, P_0, x)) + \\ & x[\mu_s(T_0, P_0, x) - \mu_s(T_0, P_0, x_0)] + (1 - x)[\mu_w(T_0, P_0, x) - \mu_w(T_0, P_0, x_0)] \end{aligned} \tag{22}$$

The conditions of the "environment" may be taken arbitrarily so that to simplify calculations. In this paper, the environmental conditions are those of pure water ($x_0 = 0$) at the temperature of the sea ($T_0 = T_{sw}$) and the atmospheric pressure ($P_0 = P_{sw}$).

3.3. Performance Criteria

The main performance criterion is the Gain Output Ratio GOR defined as the ratio of the energy required to evaporate the distillate and the heat input to the plant

$$GOR = \frac{D\lambda_{av}}{W_m \lambda_m} \tag{23}$$

As the latent heat of steam doesn't vary noticeably with temperature, the GOR is always simplified to the performance ratio Pr:

$$GOR = \frac{D\lambda_{av}}{W_m \lambda_m} \approx \frac{D}{W_m} = Pr \tag{24}$$

Based on a second-law analysis, many equivalent performance criteria may be used. We opt here to the second-law efficiency defined by Mistry et al. [11].

$$\eta_{II} = \frac{\text{minimum work of separation}}{\text{actual work of separation}} \tag{25}$$

The minimum work of separation is that necessary to separate the feed at T_{sw} into the desired distillate and the resulting brine also at T_{sw} [11]:

$$W_{sep}^{\min} = D g_D (T_{sw}) + B g_B (T_{sw}) - W_f g_{sw} (T_{sw}) \tag{26}$$

The actual work of separation used in the plant is the exergy content of the motive steam:

$$W_{sep} = W_m \left[\left(h_{gm} - h_{fm}(T_0, P_0) \right) - T_0 \left(s_{gm} - s_{fm}(T_0, P_0) \right) \right] \tag{27}$$

Besides, it is possible to define an exergy efficiency for each component of the plant:

$$\eta_{ex} = \frac{\text{outlet exergy}}{\text{inlet exergy}} = 1 - \frac{\text{destructed exergy}}{\text{inlet exergy}} \tag{28}$$

4. Results and Discussion

4.1. Main Results

The developed model is resolved using a MATLAB program. The thermodynamic properties of the different streams are taken from [21-22] and detailed in the appendix. The operating parameters of the plant are first determined and compared with the actual parameters. The energy and exergy analysis is then conducted to determine the performances of the plant and to interpret the distribution of the thermodynamic imperfection. Table **1** shows the main data of the plant while Table **2** shows the detailed results at all the points of the plant. Comparing the calculated results to the actual data, we noted that the values are sensibly the same with the exception of the distillate flow rate. In that, the actual distillate flow rate is 25 t/h while the calculated flow rate is 27 t/h which corresponds to a difference of less than 8%. Regarding the simplifying assumptions made in the model (adiabatic components, negligible pressure drop..etc.), we assume that this difference is reasonable.

Table 1: Main data.

Sea water temperature (°C)	15	Motive steam temperature (°C)	170
Sea water flow rate (t/h)	100	Motive steam flow rate (t/h)	5
Sea water salinity (kg/kg)	0.039	Discharge steam pressure (bar)	0.25
Cooling water flow rate (t/h)	0	First effect pressure (bar)	0.19
Top brine temperature (°C)	50	Second effect pressure (bar)	0.15
Motive steam pressure (bar)	5.8	Distillate flow rate (t/h)	25

Table 2: Properties and operating parameters at different points.

	W(t/h)	*T*(°C)	*P*(bar)	*x*(kg/kg)	*h*(kJ/kg)	*s*(kJ/kgK)	*e*(kJ/Kg)
Motive steam	5	170	5	0	2755,1	6,7533	810,68
Entrained vapor	9,648	54,24	0,15	0	2599,5	8,0046	294,58
Discharge steam	14,648	65	0,25	0	2618,3	7,8319	363,16
Condensate of discharge steam	14,648	65	0,25	0	272,11	0,8935	16,2488
Vapor from first stage	13,7	59,22	0,19	0	2608,3	7,9228	326,91
Condensate from second stage	28,348	59,22	0,19	0	247,96	0,8214	12,862
Vapor from second stage	13,62	54,24	0,15	0	2599,5	8,0046	294,58
Vapor to condenser	3,972	54,24	0,15	0	2599,5	8,0046	294,58
Distillate	27,33	24	2	0	100,73	0,3532	0,5737
Seawater	100	15	3	0,039	57,73	0,2112	4,3623
Feed to effects	100	50	3	0,039	199	0,6677	14,0975
Brine from first effect	36,3	60	0,19	0,0537	236,64	0,7696	24,598
Brine from second effect	72,66	55	1	0,05367	215,82	0,7101	20,0958

The Gain output rate is of the order of 5. This is principally due to the thermo-compressor. In that, in MED plants without thermo-compressor, the *GOR* is sensibly equal to the number of effects. The role played by the thermo-compressor is very important not only in raising the performance of the unit but also in making it possible to operate at low temperatures.

From a second-law viewpoint, we note that the efficiency is very low, approximately equal to 2.4. This order of magnitude is expected and is always found by different authors [7-16] depending on the type and the size of the unit. The explanation resides in the combination of a number of irreversible processes in each component of the plant i.e. heat transfer coupled to separation or mixture and to expansions and compressions.

4.2. Second-Law Analysis

The very low value of the second law efficiency shows *a priori* that it would be possible to introduce important improvements on the plant. The first step towards this eventual improvement is to assess the distribution of the irreversibility among the components and processes of the plant.

From Table **3** and Figure **2**, it is clear that the exergy destruction is equally distributed between the thermo-compressor and the heat exchangers. In that, the sum of the exergy destructed in the exchangers is of the order of 48% and that of the thermo-compressor is almost 52%.

Table 3: Exergy balances of different components.

	Ex_{in} (kW)	Ex_{out} (kW)	Ex_{deg} (kW)	η_{ex}
Entire plant	4489,63	1478,709	3010,921	32.93%
Thermo-compressor	6895,51	5319,568	1575,94	77%
Effect 1	6024,44	5609,587	414,8559	93%
Effect 2	6314,46	5836,952	477,5094	92%
Condenser	1970,91	1425,429	545,4845	72%

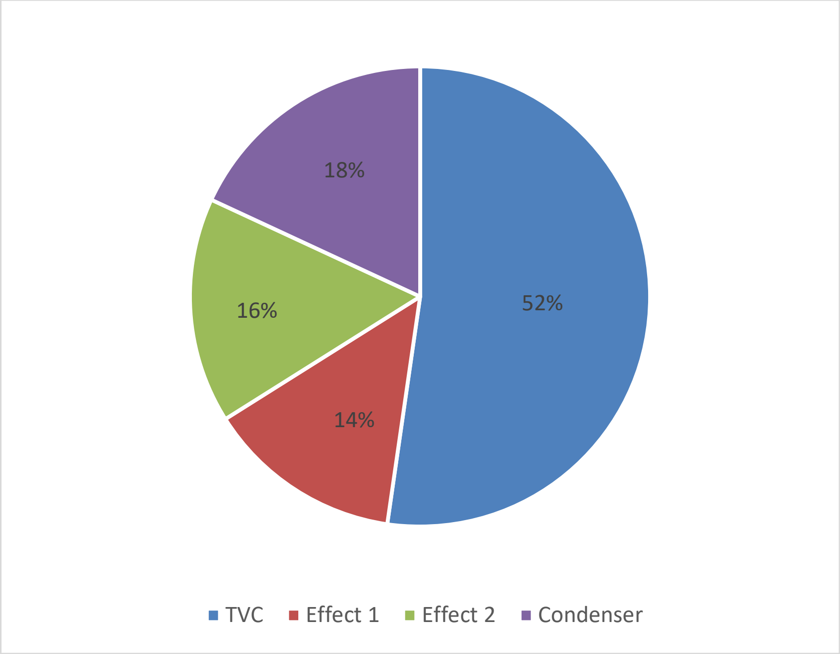

Figure 2: Distribution of the exergy destruction among the components of the plant.

The very important fraction of exergy destruction occurring in the thermo-compressor gives an order of magnitude of the “thermodynamic cost” to pay in return for the improvements made by this component. Let remind that in the absence of a thermo-compressor, the *GOR* would not exceed 2 (which means that we will consume 2.5 times the actual consumption of motive steam) and it would not be possible to operate at low temperature. The thermo-compressor has a low exergy efficiency (about 77%). The simultaneous occurrence of several complicated processes, i.e. expansion of motive steam, entrainment of vapor, a mixture of the two streams, compression of the resulting stream and eventual friction, can explain this low value.

The contributions of the three exchangers are roughly the same and have an average value of 16%. Here, less irreversible processes are occurring (only heat transfer and separation). We note that the exergy efficiencies of the two effects are roughly equal to 93% which is a good value that is principally explained by the small temperature difference, of the order of only 5°C, in these two exchanges. Again, this low-temperature difference is possible only in the presence of the thermo-compressor. The mean temperature difference in the condenser is about 3 times that in the evaporators which explains why the exergy efficiency of the condenser is lower.

For the entire plant, the ratio of the exergy outlet to the exergy inlet is about 32% which means that 68% of the exergy is initially contained in the seawater and in the motive steam is destructed within the plant. Recalling that the distillate should be theoretically at the temperature of the seawater (we seek pure water and not for hot pure water [11]) and that the brine should be sent back to the sea, the exergy outlet of the plant may be interpreted as an additional loss of exergy.

5. Conclusion

In this paper, we have established a thermodynamic analysis based on the second-law of thermodynamics for an existing double effect desalination plant with vapor thermal compression. In addition to the gain output ratio GOR , a second-law based performance criterion is adopted: the second-law efficiency [11]. The main results are as follows:

- The gain output ratio of the plant is about 5. This value would not be possible in the absence of thermo-compression. In that, the thermo-compressor makes it possible to operate at low temperature, with low-temperature differences and to consume less steam to produce the same distillate flow rate.

- The second-law efficiency is of the order of 2.4. This very low value is expected and is due to the simultaneous occurrence of several irreversible processes in the plant: heat-transfer, separation and mixture, expansion and compression. It is important to note that the exergy destruction is equally distributed between the heat-transfer components (the two effects and the condenser) on the one hand and the thermo-compressor on the other.
- The three exchangers present roughly the same part of exergy destruction. Nevertheless, the exergy efficiency of the condenser is clearly less than that of the two evaporators, 72% vs. 93%, which is due to the important temperature difference in the condenser.

As mentioned in the introduction, the physical meaning of the results is to say us what is the extent of the *imperfection* we could eventually overcome and in which locations is that imperfection concentrated. The analysis showed that there are serious chances of thermodynamic enhancement of MED-TVC plants. The major effort should be assigned to reduce the irreversibility within the thermo-compressor. Indeed, for an existing plant, the chances of introducing serious improvements are low since the geometric and operating parameters are fixed in a so tight manner that they could not be easily altered. The results may be used as a guide towards better designs.

Appendix: Thermodynamic Properties of Pure Water and Seawater

Thermodynamic properties of pure water and seawater are calculated using equations from [2, 19]. In all the equations, the temperature is in °C.

- For pure water, the saturation pressure (bar) [20] and the latent heat of evaporation (kJ/kg) are:

$$\ln(P^0) = \left[23.1698 - \frac{3799.8871}{226.346 + T}\right] 10^{-5}$$

$$\lambda_i = 2501.689845 - 2.407064037 T_i + 1.192217\,10^{-3} T_i^2 - 1.5863\,10^{-5} T_i^3$$

- The specific enthalpies (kJ/kg) of saturated liquid and saturated vapor are:

$$h_w = \left(141.355 + 4202.07T - 0.535T^2 + 0.004T^3\right) 10^{-3}$$

$$h_v = 2501.689845 + 1.806961015T + 5.987717\,10^{-4} T^2 - 1.221\,10^{-5} T^3$$

- The specific entropies (kJ/kgK) of saturated liquid and saturated vapor are:

$$s_w = \left(0.1543 + 15.383T - 2.996\,10^{-2} T^2 + 8.193\,10^{-5} T^3 - 1.37\,10^{-7} T^4\right) 10^{-3}$$

$$s_v = s_w + \frac{\lambda}{(T + 273.15)}$$

- The boiling point elevation (°C) of seawater is a function also of the salinity (ppm)

$$BPE = x\left(B + xC\right) 10^{-3}$$

$$B = \left(6.71 + 6.34\,10^{-2} T + 9.74\,10^{-5} T^2\right) 10^{-3}$$

$$C = \left(22.238 + 9.59\,10^{-3} T + 9.42\,10^{-5} T^2\right) 10^{-8}$$

- The specific heat (kJ/kg°C) is a function of the temperature (°C) and the salinity (g/kg):

$$C_p = \left(A + BT + CT^2 + DT^3\right) 10^{-3}$$

$$A = 4260.8 - 6.6197x + 1.2288\,10^{-2}\,x^2$$

$$B = -1.1262 + 5.4178\,10^{-2}\,x - 2.2719\,10^{-4}\,x^2$$

$$C = 1.2026\,10^{-2} - 5.3566\,10^{-4}\,x + 1.8906\,10^{-6}\,x^2$$

$$D = 6.87774\,10^{-7} + 1.517\,10^{-6}\,x + 4.4268\,10^{-9}\,x^2$$

- The specific enthalpy (kJ/kg) and the specific entropy (kJ/kgK) of saturated liquid as functions of the temperature (°C) and the salinity (kg/kg) are respectively

$$h_{sw} = h_w - x\begin{pmatrix} 2.438\,10^{4} + 3.152\,10^{5}\,x + 2.803\,10^{6}\,x^2 - 1.446\,10^{7}\,x^3 + 7.826\,10^{3}\,T - 4.417\,10^{1}\,T^2 \\ + 2.139\,10^{-1}\,T^3 - 1.991\,10^{4}\,Tx + 2.778\,10^{4}\,Tx^2 + 9.728\,10^{1}\,T^2x \end{pmatrix}10^{-3}$$

$$s_{sw} = s_w - x\begin{pmatrix} -4.231\,10^{2} + 1.463\,10^{4}\,x - 9.88\,10^{4}\,x^2 + 3.095\,10^{5}\,x^3 + 2.562\,10^{1}\,T - 1.443\,10^{-1}\,T^2 \\ + (5.879\,10^{-4}\,T^3 - 6.111\,10^{1}\,Tx + 8.041\,10^{1}\,Tx^2 + 3.035\,10^{-1}\,T^2x \end{pmatrix}10^{-3}$$

- The specific Gibbs energy is defined as:

$$g_{sw} = h_{sw} - Th_{sw}$$

And the chemical potential of water and salt in the solution as functions of the salinity (kg/kg) are respectively:

$$\mu_w = g_{sw} - x\frac{\partial g_{sw}}{\partial x} \qquad \mu_s = g_{sw} + (1-x)\frac{\partial g_{sw}}{\partial x}$$

Nomenclature

B	Brine flow rate (t/h)
BPE	Boiling point elevation (°C)
c	Specific heat capacity (kJ/kgK)
D,D′	Distillate flow rate (t/h)
e	Specific exergy (kJ/kg)
GOR	Gain output rate
h	Specific enthalpy (kJ/kg)
h_f	Specific enthalpy of liquid (kJ/kg)
h_g	Specific enthalpy of vapor (kJ/kg)
NEA	Non-equilibrium allowance (°C)
P	Pressure (bar)
P^0	Saturation pressure (bar)
Pr	Performance ratio
Ra	Entrainment ratio
s	Specific entropy (kJ/kgK)
s_f	Specific entropy of liquid (kJ/kg)

s_g	Specific entropy of vapor (kJ/kg)
T	Temperature (°C)
W	flow rate (t/h)
W_{sep}	Work of separation (kW)
x	Salinity (ppm, g/kg or kg/kg)

Greek Symbols

η	Efficiency
μ	Chemical potential (kJ/kg)
λ	Latent heat of evaporation (kJ/kg)

Subscripts

av	Average
c	Cooling water
D	Distillate
d	Discharge steam
ex	exergy
ev	Entrained vapor
J	Feed
1,2	effects
m	Motive steam
s	Salt
sw	Seawater
v	vaporization
w	Water

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