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Abstract

Seawater cooling towers have been used since the 1970's in power generation and other industries, so as to reduce the consumption of freshwater. The salts in seawater are known to create a number of operational problems including salt deposition, packing blockage, corrosion, and certain environmental impacts from salt drift and blowdown return. In addition, the salinity of seawater affects the thermophysical properties which govern the thermal performance of cooling towers, including vapor pressure, density, specific heat, viscosity, thermal conductivity and surface tension. In this paper, the thermal performance of seawater cooling towers is investigated using a detailed model of a counterflow wet cooling tower. The model takes into consideration the coupled heat and mass transfer processes and does not make any of the conventional Merkel approximations. In addition, the model incorporates the most up-to-date seawater properties in the literature. The model governing equations are solved numerically and its validity is checked against the available data in the literature. Based on the results of the model, a correction factor that characterizes the degradation of the cooling tower effectiveness as a function of seawater salinity and temperature approach is presented for performance evaluation purposes.

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Introduction

Cooling towers are used in many applications to reject heat to the atmosphere. Heat rejection is accomplished within the tower by heat and mass transfer between the hot water droplets and ambient air. Seawater cooling towers have been used since the 1970's in facilities on the coast, as there is a potential to reduce fresh water consumption in power plants and other industries. In addition, the use of once-through cooling systems where hot water is rejected back into the sea caused many environmental problems. Therefore, seawater cooling towers have been found to be a competitive alternative in which seawater is recycled in a closed-loop cooling system [1]. The salts in the water create a number of engineering challenges including salt deposition, packing blockage, corrosion, potentially rising salt concentration, and salt emissions (drift). Moreover, the salts in seawater change the thermophysical properties with respect to freshwater, which in turn change the thermal performance of cooling towers.

The corrosion problems in seawater cooling towers can be avoided by appropriate selection of construction material and equipment. The use of plastic and asbestos for packing, pipes and water distribution system provided a practical and predictable solution for most of the corrosion problems. The use of exposed ferrous metal must be avoided and if it is necessary to use metal for specific requirements, monel or stainless steel should be selected. Coatings such as epoxy may also be used to cover special metal construction joints or sometimes galvanized rebar is used in critical areas. More details and material selection for seawater cooling towers can be found in Walston [2].

Obviously, all of these special materials add to the capital cost of the tower, which is beyond the scope of this paper.

The thermal design and performance of cooling towers have been abundantly discussed in the literature. The first cooling tower theory was developed by Merkel [3], and it included many approximations. The major assumptions in Merkel’s model are: the water loss by evaporation is neglected; Lewis factor is assumed to be unity; and the exit air is assumed to be saturated. Sutherland [4] found that using the Merkel model can result in undersizing the tower between 5 to 15%. A more accurate model was developed by Poppe and Rogener [5] without using any of Merkel’s approximations. The cooling tower characteristics or Merkel number determined by Poppe’s approach is approximately 10% higher than the Merkel number determined by the Merkel model [6]. Knowing that the effect of seawater properties on the cooling tower thermal performance may be small at lower salinities, it is intended in this paper to use an accurate cooling tower model that does not make any of the Merkel approximations.

The thermal performance of seawater cooling towers has not been studied carefully in the literature. The available data are mostly in technical reports, feasibility studies, or design guidance [7, 8]. General discussion about the effect of seawater properties on the thermal performance was given by Nelson [9] and Warner [10]. However, no detailed performance calculation was made. As a rule of thumb, cooling tower vendors recommend degrading the tower performance by approximately 1% for every 10,000 ppm of salts in the cooling water. In practice, most engineering contractors specify a 0.55-1.1 ^{o}C margin on the wet bulb temperature to account for salts in the cooling water [8]. The objective of this paper is to investigate the thermal performance of seawater cooling towers by using a detailed model and to provide a correction factor that relates the performance of the seawater to that of fresh water cooling tower that has the same size and operating conditions.

Seawater properties

The thermophysical properties of seawater are different from those of fresh water. This difference is sufficient to affect the heat and mass transfer processes in cooling towers. The literature contains many data for the properties of seawater, but only a few sources provide full coverage for all relevant thermophysical properties. A recent review and assessment of seawater properties is given by Sharqawy et al. [11]. The properties that most strongly affect the thermal performance of cooling tower are vapor pressure, density, and specific heat capacity. In addition, thermal conductivity, viscosity and surface tension affect the heat and mass transfer coefficients within the packing. In this section, correlations of seawater properties to be used in the cooling tower model are described. All liquid properties are given at 1 atm pressure.

The vapor pressure of seawater is less than that of fresh water which reduces the potential for water evaporation. The vapor pressure can be calculated using Raoult's law which states that the vapor pressure of seawater is equal to the product of water mole fraction in seawater and water's vapor pressure in the pure state. The mole fraction of water in seawater is a function of the salinity. Using these results, an equation for seawater vapor pressure based on Raoult's law is given by Eq. (1)

$$p_{v,w}/p_{v,sw} = 1 + 0.57357 \times \left(\frac{S}{1000 - S} \right) \tag{1}$$

where S is the seawater salinity in g/kg on the reference-composition salinity scale defined by Millero et al. [12] which is currently the best estimate for the absolute salinity of seawater. From Eq. (1), it is shown that the seawater vapor pressure decreases with the increase of salinity. This decrease reaches about 8% at salinity of 120 g/kg. Consequently, the humidity ratio and the enthalpy of saturated air above the water surface decrease. As will be seen later in the cooling tower model, the difference between the enthalpy of saturated humid air at water temperature and the enthalpy of humid air at air temperature and humidity is the driving force for water evaporation and mass transfer [13]. Therefore, the reduction of vapor pressure by salinity decreases the amount of water evaporated into the air stream and hence reduces the heat rejection capability.

The specific heat of seawater is less than that of freshwater which reduces the amount of sensible heat that can be transferred at the same temperature difference. The specific heat capacity can be calculated by using Eq. (2) given by Jamieson et al. [14] which fits the experimental measurements with an accuracy of ±0.3%. Equation (2) is valid for temperatures of 0 – 180 ^{o}C and salinities of 0 – 180 g/kg.

$$c_{p,sw} = A + BT + CT^2 + DT^3 \tag{2}$$

where $c_{p,sw}$ is in kJ/kg K, T in K, S in g/kg, and

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

Figure 1 shows the specific heat of seawater calculated from Eq. (2) as a function of temperature and salinity. It is shown that the specific heat of seawater is less than that of fresh water by about 12% at 120 g/kg salinity.

The density of seawater is higher than that of fresh water due to the salt content. This increases the mass flow rate of seawater for the same volumetric flow rate. Consequently increases the pumping power. The density of seawater can be calculated using Eq. (3) [11] which best fits the seawater density data measured by Isdale and Morris [15] and Millero and Poisson [16] and is in good agreement with IAPWS [17]. The fresh water density is given by Eq. (4) which best fits the fresh water density data extracted from the IAPWS [18] formulation of pure liquid water. Equation (3) has an accuracy of ±0.1% and valid for temperatures of 0 – 180 ^{o}C and salinities of 0 – 160 g/kg.

$$\rho_{sw} = \rho_w + S\left(b_1 + b_2\, t + b_3\, t^2 + b_4\, t^3 + b_5 S\, t^2\right) \tag{3}$$

$$\rho_w = \left(a_1 + a_2 t + a_3 t^2 + a_4 t^3 + a_5 t^4\right) \tag{4}$$

where ρ_{sw} and ρ_w are in kg/m^3, t in ^{o}C, S in g/kg, and

$$a_1 = 9.999\times10^{2}, a_2 = 2.034\times10^{-2}, a_3 = -6.162\times10^{-3},$$
$$a_4 = 2.261\times10^{-5}, a_5 = -4.657\times10^{-8}, b_1 = 0.8020,$$
$$b_2 = -2.001\times10^{-3}, b_3 = 1.677\times10^{-5}, b_4 = -3.060\times10^{-8},$$
$$b_5 = -1.613\times10^{-11}$$

Figure 2 shows the density of seawater calculated from Eq. (3) as it changes with temperature and salinity. It is shown in Fig. 2 that the density of seawater is higher than that of fresh water by about 10% at 120 g/kg salinity.

The viscosity of seawater is higher than that of fresh water by about 40% at a salinity of 120 g/kg (see Fig. 3). It can be calculated using Eq. (5) given by Sharqawy et al. [11] which is valid for temperatures of 0 – 180 ^{o}C and salinities of 0 – 150 g/kg and has an accuracy of ±1.5 %. The pure water viscosity is given by Eq. (6) which fits data extracted from the IAPWS [19] release with an accuracy of ±0.05% and is valid for t = 0-180 ^{o}C.

$$\mu_{sw} = \mu_w \left(1 + A\,S + B\,S^2\right) \tag{5}$$

$$\mu_w = 4.2844\times10^{-5} + \left(0.157\left(t + 64.993\right)^2 - 91.296\right)^{-1} \tag{6}$$

where μ_{sw} and μ_w are in kg/m.s, t in ^{o}C, S in g/kg, and

$$A = 1.541\times10^{-3} + 1.998\times10^{-5}\ t - 9.52\times10^{-8}\ t^2$$

$$B = 7.974\times10^{-6} - 7.561\times10^{-8}\ t + 4.724\times10^{-10}\ t^2$$

The surface tension of seawater is higher than that of fresh water by about 1.5% at salinity of 40 g/kg (see Fig. 4). Unfortunately the available data and correlations for seawater surface tension are limited to temperatures of 40 ^{o}C and salinities of 40 g/kg [11]. Surface tension can be calculated using Eq. (7) which is valid for temperatures of 0 – 40 ^{o}C and salinities of 0 – 40 g/kg with an accuracy of ±0.2%. Pure water surface tension is given by Eq. (8) [20] which is valid for t = 0 – 370 ^{o}C and has an accuracy of ±0.08%.

$$\frac{\sigma_{sw}}{\sigma_w} = 1 + \left(0.000226 \times t + 0.00946\right) \ln\left(1 + 0.0331 \times S\right) \tag{7}$$

$$\sigma_w = 0.2358\left(1 - \frac{t + 273.15}{647.096}\right)^{1.256}\left[1 - 0.625\left(1 - \frac{t + 273.15}{647.096}\right)\right] \tag{8}$$

where σ_{sw} and σ_w are in N/m, t in ^{o}C and S in g/kg.

The thermal conductivity of seawater is less than that of fresh water by about 1% at 120 g/kg (see Fig. 5). It can be calculated using Eq. (9) given by Jamieson and Tudhope [21] which is valid for temperature of 0 – 180 ^{o}C and salinities of 0 – 160 g/kg with an accuracy of ±3%.

$$\log_{10}\left(k_{sw}\right) = \log_{10}\left(240 + 0.0002\,S\right) + 0.434\left(2.3 - \frac{343.5 + 0.037S}{t + 273.15}\right)\left(1 - \frac{t + 273.15}{647 + 0.03S}\right)^{0.333} \tag{9}$$

where k_{sw} is in mW/m.K, t in ^{o}C and S in g/kg.

Cooling Tower Model

A schematic diagram of the counterflow cooling tower is shown in Fig. 6, including the important states and boundary conditions. The assumptions that are used to derive the modeling equations are as follows:

- Negligible heat transfer between the tower walls and the external environment.
- Constant mass transfer coefficient throughout the tower.
- The Lewis factor that relates the heat and mass transfer coefficients is not unity.
- Water mass flow lost by evaporation is not neglected.
- Uniform temperature throughout the water stream at any horizontal cross section.
- Uniform cross-sectional area of the tower.
- The atmospheric pressure is constant along the tower and equal to 101.325 kPa.

A steady-state heat and mass balances on an incremental volume leads to the following differential equations [6]

Energy balance on moist air:

$$\frac{dh_a}{dz} = MR \times Me \times \left[Le\left(h_{s,w} - h_a\right) + \left(1 - Le\right)\left(\omega_{s,w} - \omega\right)h_v\right] \tag{10}$$

Mass balance on water vapor

$$\frac{d\omega}{dz} = MR \times Me \times \left(\omega_{s,w} - \omega\right) \tag{11}$$

Energy balance on seawater:

$$\frac{dt_{sw}}{dz} = \left[\frac{1}{MR - \left(\omega_o - \omega\right)}\right] \times \left[\frac{1}{c_{p,sw}}\frac{dh_a}{dz} - \left(t_{sw} - t_{ref}\right)\frac{d\omega}{dz}\right] \tag{12}$$

Mass balance on salts

$$\frac{dS}{dz} = \left[\frac{-S}{MR - (\omega_o - \omega)} \right] \times \frac{d\omega}{dz} \tag{13}$$

where

$$Le = h_c / h_D c_{p,a} = 0.865^{0.667} \left(\frac{\omega_{s,w} + 0.622}{\omega_{s,a} + 0.622} - 1 \right) \bigg/ \ln \left(\frac{\omega_{s,w} + 0.622}{\omega_{s,a} + 0.622} \right) \tag{14}$$

$$MR = \dot{m}_{w,i} / \dot{m}_a \tag{15}$$

$$Me = h_D a V / \dot{m}_{w,i} \tag{16}$$

It is important to mention that in the cooling tower literature, the mass flow rate ratio (*MR*) is usually referred by L/G (liquid-to-gas flow rate ratio) and Merkel number (*Me*) is usually referred by KaV/L where K is the mass transfer coefficient and *L* is water mass flow rate. However, in recent studies [6, 22] these symbols have been replaced by the ones used in this paper. In addition, the multiplication of the mass flow rate ratio and Merkel number (*MR* x *Me*) is referred in the literature as the number of transfer units, *NTU* (Braun et al. [23]).

For a given number of transfer units (*NTU*), mass flow rate ratio (*MR*) and inlet conditions ($t_{w,i}$, S_i, $t_{a,i}$, ω_i). Equations (10) – (13) can be solved numerically to find the exit conditions for both air and seawater streams. The solution is iterative with respect to the outlet air humidity, outlet seawater temperature and outlet seawater salinity (ω_o, $t_{w,o}$, S_o). In this solution, seawater properties are calculated along the tower length using the equations presented in the previous section. The Lewis factor is calculated using Eq. (14) given by Bosnjakovic [24] and the moist air properties are calculated using the correlations provided by Klopper [25]. In addition, seawater vapor pressure, Eq. (1), is used to determine the humidity ratio and enthalpy of the saturated moist air at seawater temperature.

In the above cooling tower model, the heat and mass transfer coefficients are related by Lewis factor based on Chilton-Colburn analogy. However, the mass transfer coefficient (h_D) should be determined in order to know the number of transfer units. Unfortunately, general correlations for the mass transfer coefficient in terms of physical properties and packing specifications do not exist for cooling towers. For that reason, experimental measurements are normally carried out to determine the transfer characteristics for different packing types. It is, however, important to note that in the present work an empirical correlation given by Djebbar and Narbaitz [26] is used to calculate the change in number of transfer units of a particular packing when seawater properties are used instead of fresh water properties. This equation is a modified form of Onda's correlation (Onda et al., [27]) and has an average error of ±26% relative to experimental data. A comparison between the packing characteristic (Merkel number) calculated using the Djebbar and Narbaitz's correlation and the experimental given by Narbaitz et al. [28] is shown in Fig. 7a.

Despite the deviation between Djebbar and Narbaitz's model and the experimental measurements, the effect of physical property variation with salinity on the mass transfer coefficient is very small as shown in Fig. 7b. In this figure, the number of transfer units decreases by about 7% at a salinity of 120 g/kg. This reduction agrees well with the data presented by Ting and Suptic [29]. They recommended rating the cooling tower as if it was using freshwater and then increasing the water flow rate to compensate for the reduction in the number of transfer units by applying a mass flow rate correction factor. This correction factor method can be used in a design stage of the cooling tower. However, for rating of cooling towers if we assume a seawater cooling tower working at the same number of transfer units as of a fresh water tower, it is important to calculate the reduction in the cooling tower effectiveness. Therefore, it is assumed in the following analysis that the number of transfer units is the same as for a fresh water cooling tower and the reduction in the effectiveness is calculated subsequently.

The mathematical model given by equations (10) – (13) subject to the boundary conditions showed on Fig. 6 were transferred to finite difference equations and solved by a successive over-relaxation method followed a procedure outlined by Patrick et al. [30]. A convergence criterion of $(t^{n+1} - t^{n}) \leq 10^{-5}$ was used for the present computations where n is the number of iterations. Numerical solutions for the air and water temperature distribution along the tower as well as the air humidity and seawater salinity were obtained at different inlet conditions.

Results and discussion

To illustrate the results of the present work, the air effectiveness of the cooling tower is calculated at different inlet conditions. The air effectiveness is defined as the ratio of the actual to maximum possible air-side heat transfer that would occur if the outlet air stream was saturated at the incoming water temperature (Narayan et al. [31]), given by

$$\varepsilon_a = \frac{h_{a,o} - h_{a,i}}{h_{s,w,i} - h_{a,i}} \tag{17}$$

To examine the validity of the numerical solution, the results at zero salinity were compared to those given by Braun et al. [23] who solved the same set of equations for Lewis factor of unity and constant properties. The comparison is achieved by making the necessary adjustments to the present model to suit Braun's assumptions. Figure 8 shows a comparison between the cooling tower air effectiveness from the present work and from Braun et al. [23]. The numerical solution of the present work is in excellent agreement with that of Braun. In addition, Fig. 8 shows the numerical results using Merkel assumptions. The Merkel assumption solution differs by 1-3% at these particular conditions, however at higher water temperatures (40-60 ^{o}C) the amount of water evaporation increases and the difference may reach 10-15%.

Figures 9 through 11 show the air effectiveness of the cooling tower as it changes with the number of transfer units at different mass flow ratio and seawater salinity. In these figures, the dry and wet bulb temperatures of the inlet air are 30^{o}C and 25^{o}C respectively. In Fig. 9, the inlet water temperature is 40^{o}C and the salinity of the inlet seawater is taken as 0 (fresh water), 40 and 80 g/kg. As shown in this figure, the air effectiveness decreases as the salinity increases. The decrease in the effectiveness is a weak function of *NTU* and *MR*. The air effectiveness decreases by about 5% at salinity of 40 g/kg and by about 10 % at salinity of 80 g/kg.

To examine whether the reduction of air effectiveness depends on the seawater inlet temperature, numerical results are obtained at different seawater inlet temperatures. Figures 10 and 11 show the air effectiveness versus *NTU* at seawater inlet temperatures of 60 and 80^{o}C, respectively. It is found that the average reduction in the effectiveness is about 5% at a salinity of 40 g/kg and about 10% at a salinity of 80 g/kg. This salinity-dependent reduction is the same when the water inlet temperature is 40^{o}C.

From Fig. 9 – 11, it is clear that the air effectiveness of the cooling tower decreases with an increase in the seawater salinity. This reduction is a linear function of the salinity as shown in Fig. 12. However, the slope of this linear relationship depends on the approach (App) which is the difference between the outlet water temperature and the inlet air wet bulb temperature given by Eq. (18).

$$App = T_{w,o} - T_{wb,i} \tag{18}$$

Figure 12 shows that at a lower approach, the reduction in the effectiveness is higher than at higher approaches for the same seawater salinity. This is because at lower approaches the potential for water evaporation decreases (the difference between saturated air enthalpy at water temperature and air enthalpy is lower). Therefore, the effect of reducing the vapor pressure due to the salts becomes significant on the effectiveness. This is found to be true for the range of *NTU*, *MR* and $t_{w,i}$ studied in this paper. Therefore, a simple expression is obtained for the reduction in the air effectiveness. The slope of the linear relationship between the salinity and effectiveness reduction is plotted versus the approach in Fig. 13, and a best fit equation is obtained as shown on this figure. Consequently, a relationship between the air effectiveness reduction as a function of salinity and approach can be expressed as,

$$1-\frac{\varepsilon_a}{\varepsilon_a^o}=\left(0.1324-0.0033\times App\right)\times S \tag{19}$$

where ε_a^o and *App* are the air effectiveness and approach at zero salinity, respectively. Equation (19) can be rewritten in the form of a correction factor (CF) for the air effectiveness. This correction factor is the ratio between the air effectiveness at any salinity and that at zero salinity, written as,

$$CF=\frac{\varepsilon_a}{\varepsilon_a^o}=1-\left(0.1324-0.0033\times App\right)\times S \tag{20}$$

It is important to note that Eq. (20) estimates the reduction of the cooling tower air effectiveness within ±2% from that calculated using the full numerical solution. This can be considered as an accurate estimation at higher salinities where the reduction in the air effectiveness is high (14 to 18%). However, at lower salinities, it is recommended to solve the governing equations numerically to get a better estimate. In addition, this correction factor assumes that the number of transfer units is independent of the salinity which is an approximation with the following accuracy: The *NTU* decreases by a maximum of 7% at a salinity of 120 g/kg (for the particular packing shown in Fig. 7), which in turn reduces the effectiveness by an additional 3%. However, for typical seawater salinity of 40 g/kg, the reduction of *NTU* is about 2% which reduces the effectiveness by about 0.85%. It is somewhat difficult to combine the effect of salinity on the *NTU* and the effectiveness, since this calculation must be carried out for a particular packing with known specifications. Therefore, further reduction in the effectiveness should be considered when using Eq. (20) to account for the effect of salinity on the *NTU*. This reduction ranges from 0.85% at typical seawater salinity (40 g/kg) to 3% for salinity of 120 g/kg.

Conclusion

The thermal performance of a seawater cooling tower is investigated in this paper. The thermophysical properties of seawater that affect the thermal performance are discussed and given as a function of salinity and temperature. A detailed numerical model for a counterflow cooling tower is developed and numerical solution for the air effectiveness is obtained. It is found that an increase in salinity decreases the air effectiveness by 5 to 20% relative to fresh water cooling tower. A correction factor correlation is obtained that relates the effectiveness of the seawater cooling tower with that of fresh water cooling tower for the same tower size and operating conditions. This correction factor equation is valid up to salinity of 120 g/kg and is accurate within ±2% with respect to the present numerical results.

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Nomenclature

a	effective surface area for heat and mass transfer per unit volume	$m^2\ m^{-3}$
App	cooling tower approach given by Eq. (18)	K
c_p	specific heat at constant pressure	$J\ kg^{-1}\ K^{-1}$
CF	correction factor given by Eq. (20)	
h	specific enthalpy	$J\ kg^{-1}$
h_c	convective heat transfer coefficient	$W\ m^{-2}\ K^{-1}$
h_D	mass transfer coefficient (also K)	$kg\ m^{-2}\ s^{-1}$
h_v	specific enthalpy of water vapor	$J\ kg^{-1}$
k	thermal conductivity	$W\ m^{-1}\ K^{-1}$
Le	Lewis factor defined by Eq. (14)	
$\dot{m}$	mass flow rate (also L)	$kg\ s^{-1}$
MR	inlet water to air mass flow ratio	
n	number of iterations	
NTU	number of transfer units defined by Eq. (16)	
P_v	vapor pressure	Pa
S	seawater salinity	$g\ kg^{-1}$
t	temperature	^oC
T	temperature	K
t_{ref}	reference temperature taken as 0 ^oC	^oC
V	volume of cooling tower	m^3
z	dimensionless height of packing in the cooling tower	

Greek Symbols

ε	effectiveness	
ρ	density	$kg\ m^{-3}$
μ	dynamic viscosity	$kg\ m^{-1}\ s^{-1}$
σ	surface tension	$N\ m^{-1}$
ω	humidity ratio	$kg\ kg^{-1}$

Subscripts

a	moist air
i	inlet
o	outlet
s	saturated
sw	seawater
w	pure water
wb	wet bulb

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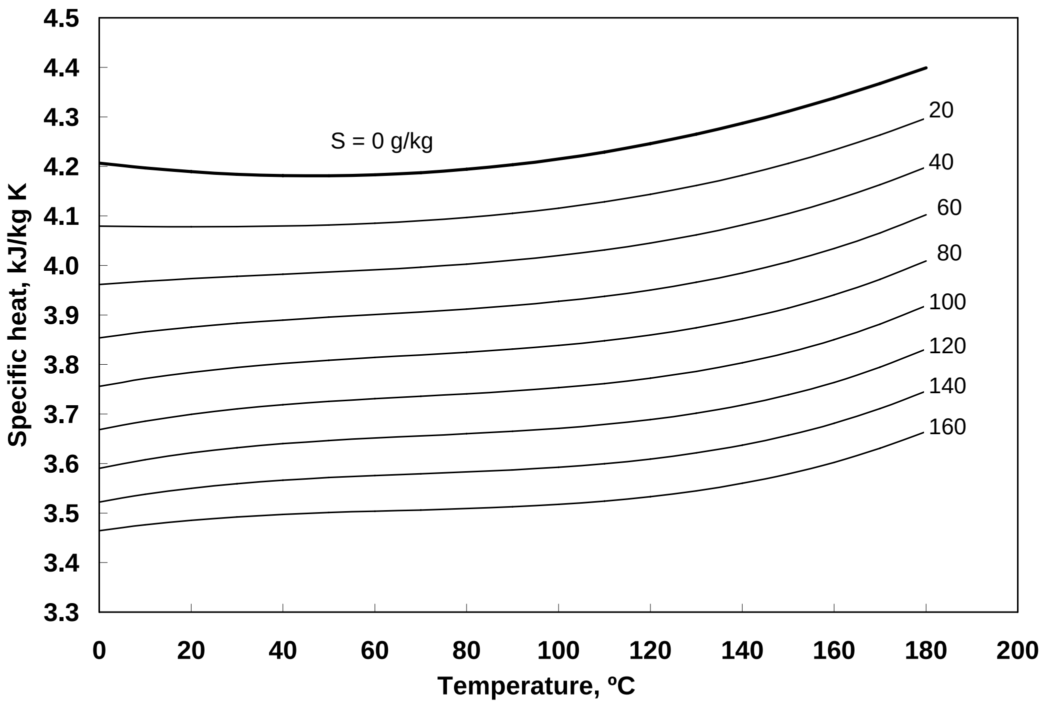
Fig. 1



S = 0 g/kg
20
40
60
80
100
120
140
160
Specific heat, kJ/kg K
Temperature, ºC
4.5
4.4
4.3
4.2
4.1
4.0
3.9
3.8
3.7
3.6
3.5
3.4
3.3
0
20
40
60
80
100
120
140
160
180
200

Fig. 2

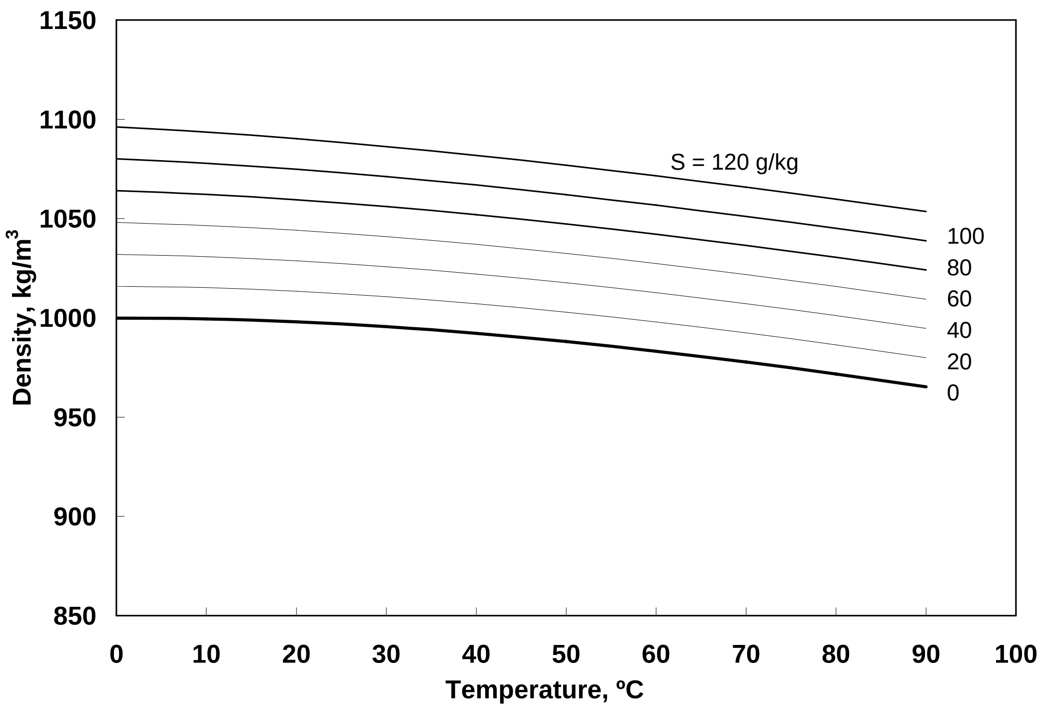

1150
1100
1050
1000
950
900
850
Density, kg/m³
0
10
20
30
40
50
60
70
80
90
100
Temperature, ºC
S = 120 g/kg
100
80
60
40
20
0

Fig. 3

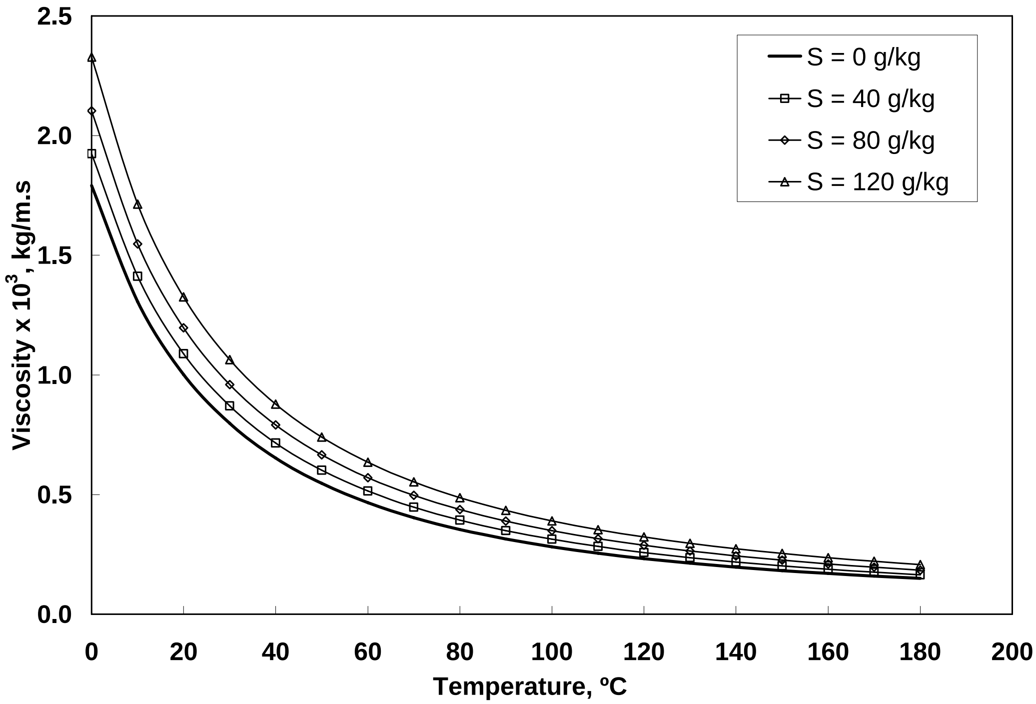
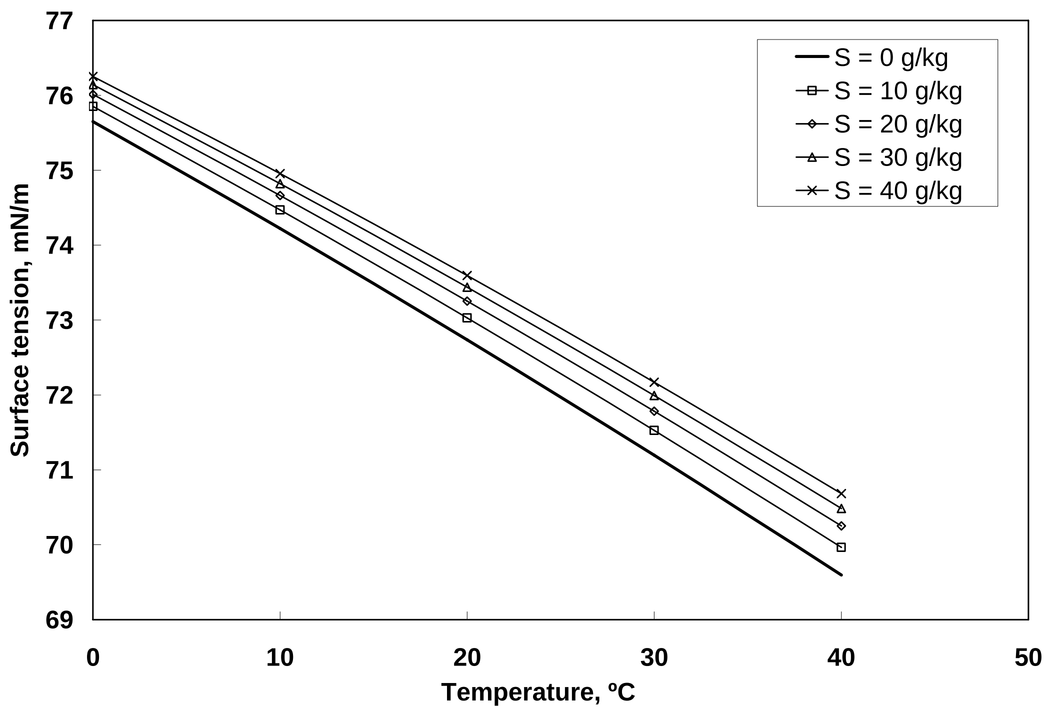

S = 0 g/kg
S = 40 g/kg
S = 80 g/kg
S = 120 g/kg
Viscosity x 10^3, kg/m.s
Temperature, ºC
2.5
2.0
1.5
1.0
0.5
0.0
0
20
40
60
80
100
120
140
160
180
200

Fig. 4



77
76
75
74
73
72
71
70
69
0
10
20
30
40
50
Surface tension, mN/m
Temperature, ºC
S = 0 g/kg
S = 10 g/kg
S = 20 g/kg
S = 30 g/kg
S = 40 g/kg

Fig. 5

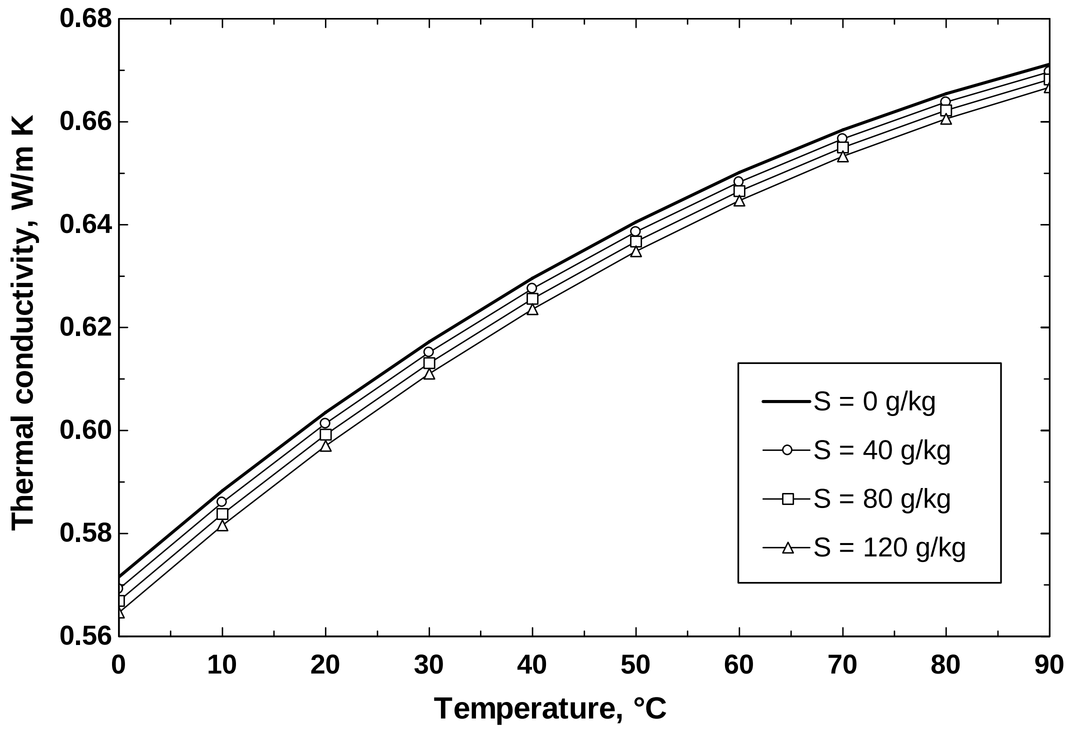
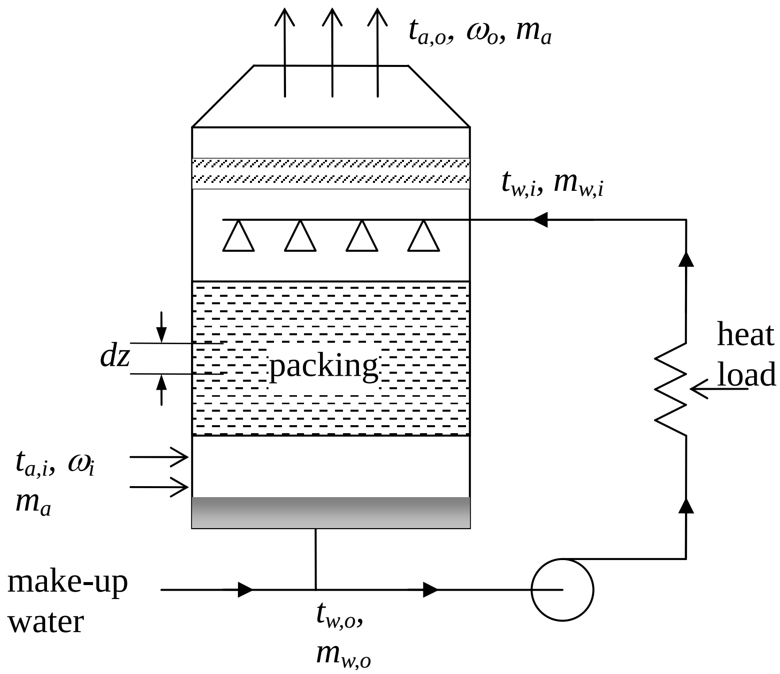
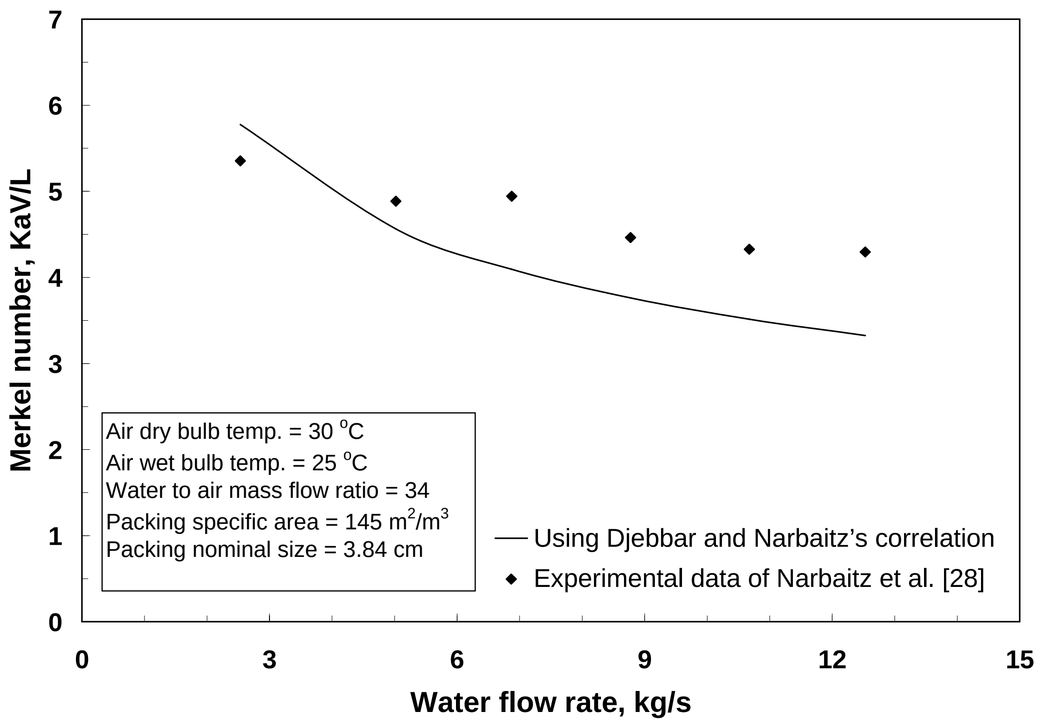

0.68
0.66
0.64
0.62
0.60
0.58
0.56
Thermal conductivity, W/m K
0
10
20
30
40
50
60
70
80
90
Temperature, °C
S = 0 g/kg
S = 40 g/kg
S = 80 g/kg
S = 120 g/kg

Fig. 6



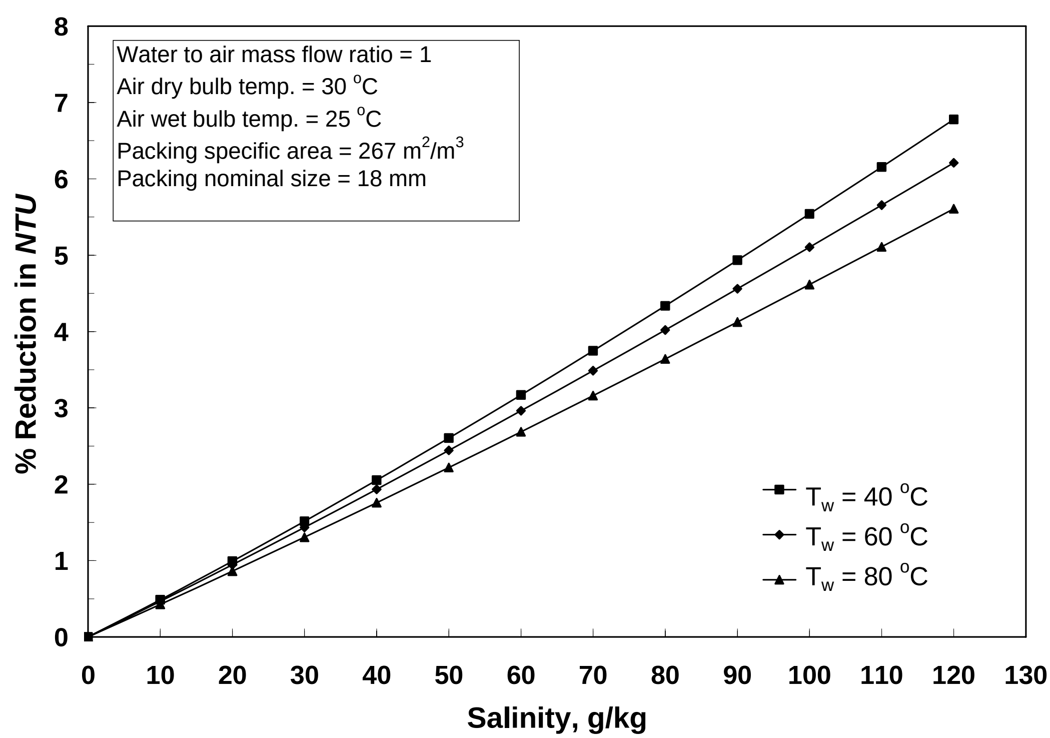
$t_{a,o}$, ω_o, m_a
$t_{w,i}$, $m_{w,i}$
heat
load
dz
packing
$t_{a,i}$, ω_i
m_a
make-up
water
$t_{w,o}$,
$m_{w,o}$

Fig. 7a



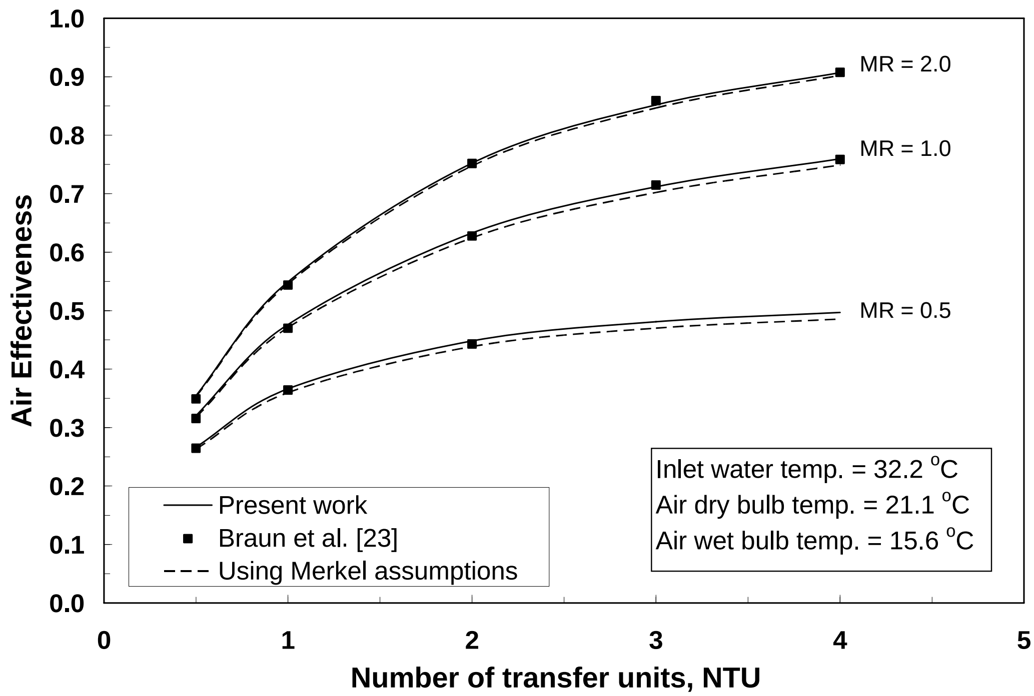
Merkel number, KaV/L
Water flow rate, kg/s
Air dry bulb temp. = 30 °C
Air wet bulb temp. = 25 °C
Water to air mass flow ratio = 34
Packing specific area = 145 m²/m³
Packing nominal size = 3.84 cm
Using Djebbar and Narbaitz's correlation
Experimental data of Narbaitz et al. [28]
0
1
2
3
4
5
6
7
0
3
6
9
12
15

Fig. 7b



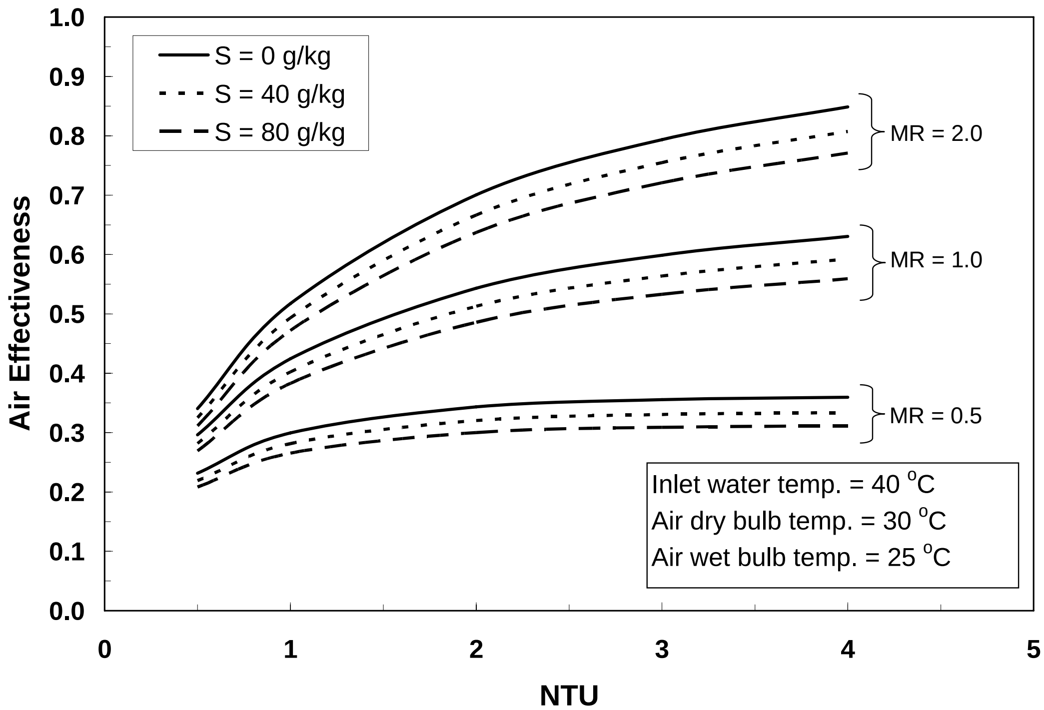
% Reduction in NTU
Salinity, g/kg
Water to air mass flow ratio = 1
Air dry bulb temp. = 30 °C
Air wet bulb temp. = 25 °C
Packing specific area = 267 m²/m³
Packing nominal size = 18 mm
Tw = 40 °C
Tw = 60 °C
Tw = 80 °C
0
1
2
3
4
5
6
7
8
0
10
20
30
40
50
60
70
80
90
100
110
120
130

Fig. 8



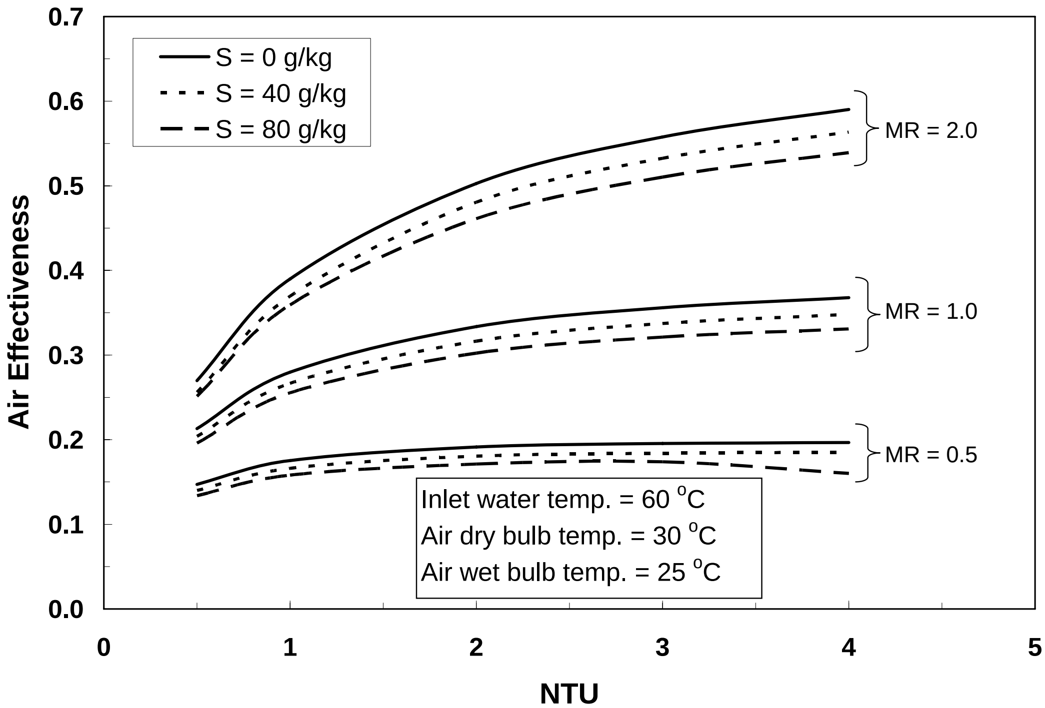
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
Air Effectiveness
0
1
2
3
4
5
Number of transfer units, NTU
MR = 2.0
MR = 1.0
MR = 0.5
Present work
Braun et al. [23]
Using Merkel assumptions
Inlet water temp. = 32.2 °C
Air dry bulb temp. = 21.1 °C
Air wet bulb temp. = 15.6 °C

Fig. 9



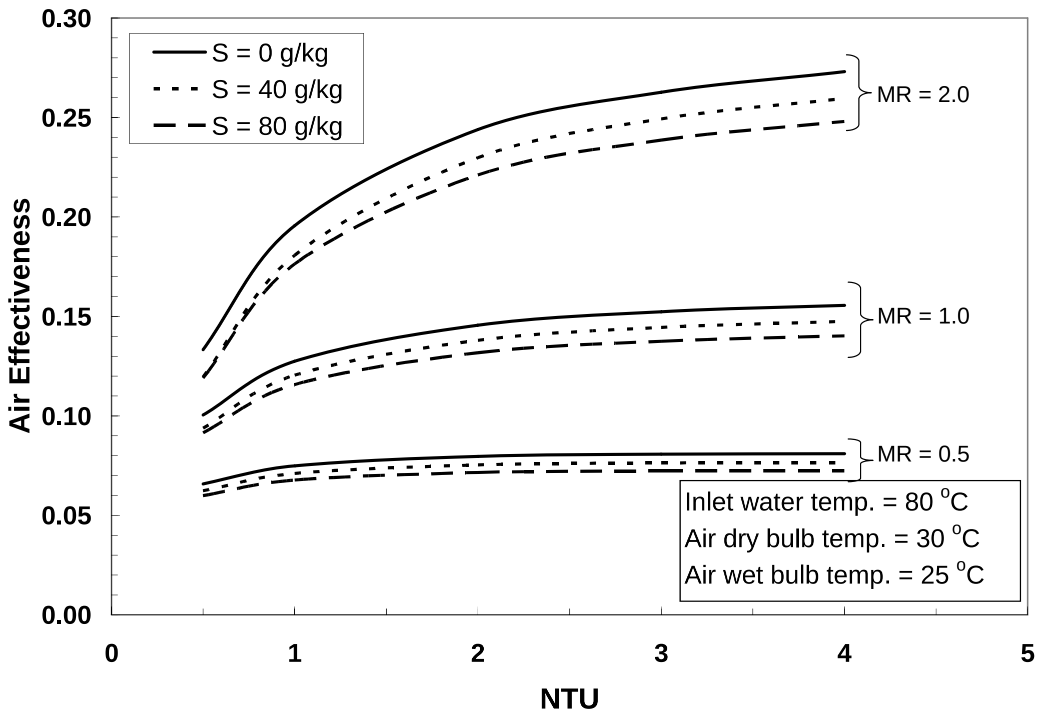
S = 0 g/kg
S = 40 g/kg
S = 80 g/kg
MR = 2.0
MR = 1.0
MR = 0.5
Inlet water temp. = 40 °C
Air dry bulb temp. = 30 °C
Air wet bulb temp. = 25 °C
Air Effectiveness
NTU
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
0
1
2
3
4
5

Fig. 10



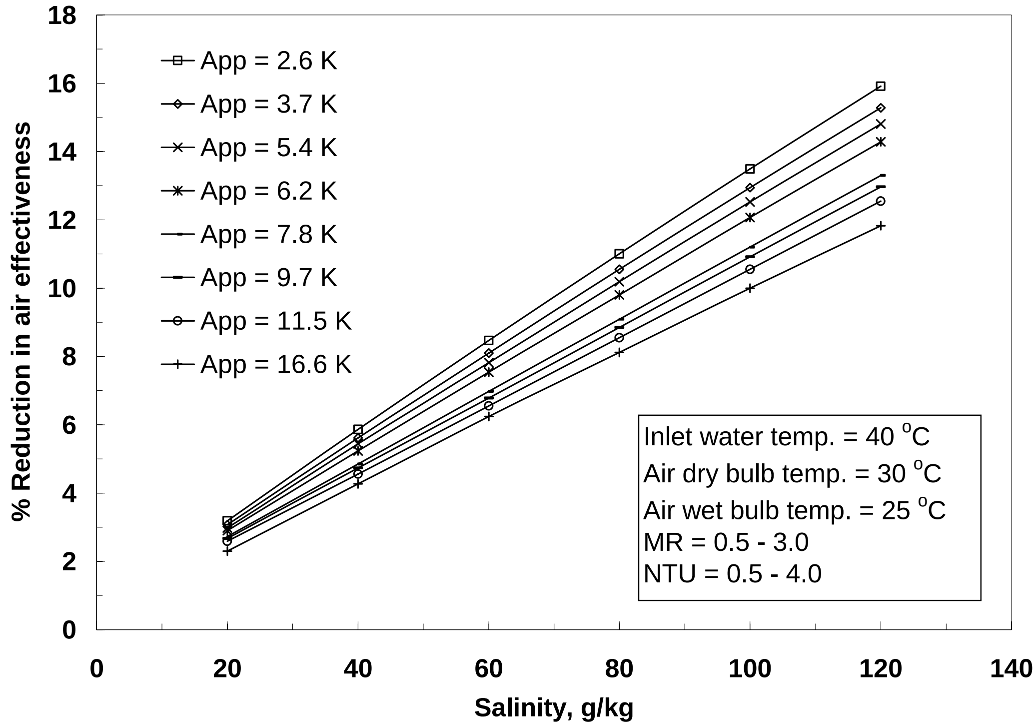
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
Air Effectiveness
S = 0 g/kg
S = 40 g/kg
S = 80 g/kg
MR = 2.0
MR = 1.0
MR = 0.5
Inlet water temp. = 60 °C
Air dry bulb temp. = 30 °C
Air wet bulb temp. = 25 °C
0
1
2
3
4
5
NTU

Fig. 11



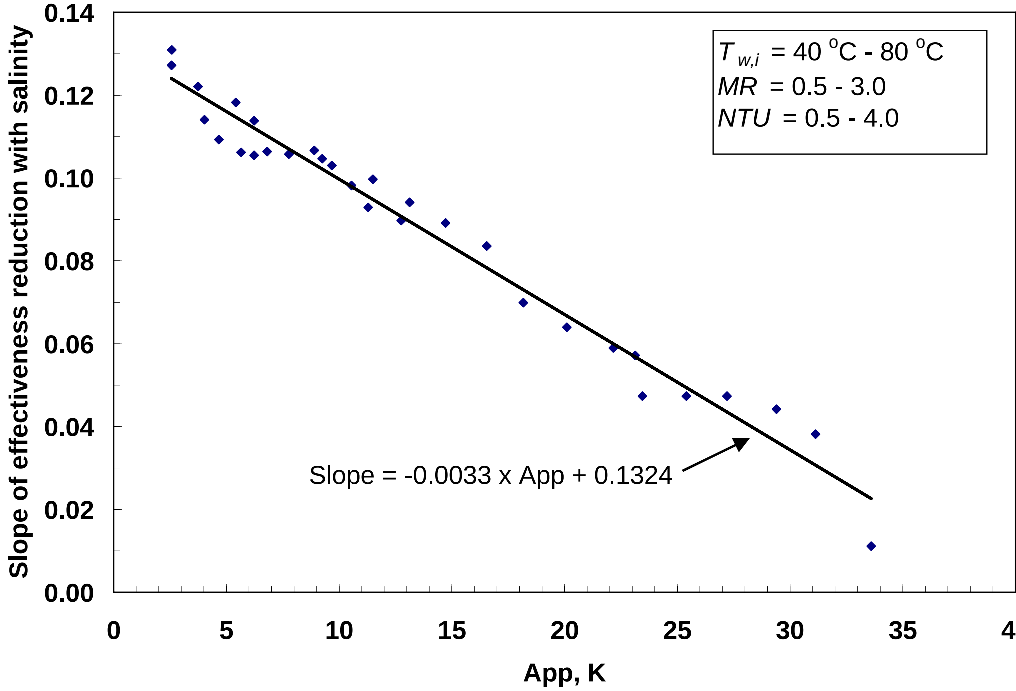
0.30
0.25
0.20
0.15
0.10
0.05
0.00
Air Effectiveness
0
1
2
3
4
5
NTU
S = 0 g/kg
S = 40 g/kg
S = 80 g/kg
MR = 2.0
MR = 1.0
MR = 0.5
Inlet water temp. = 80 °C
Air dry bulb temp. = 30 °C
Air wet bulb temp. = 25 °C

Fig. 12



18
16
14
12
10
8
6
4
2
0
0
20
40
60
80
100
120
140
Salinity, g/kg
% Reduction in air effectiveness
App = 2.6 K
App = 3.7 K
App = 5.4 K
App = 6.2 K
App = 7.8 K
App = 9.7 K
App = 11.5 K
App = 16.6 K
Inlet water temp. = 40 °C
Air dry bulb temp. = 30 °C
Air wet bulb temp. = 25 °C
MR = 0.5 - 3.0
NTU = 0.5 - 4.0

Fig. 13



0.14
0.12
0.10
0.08
0.06
0.04
0.02
0.00
Slope of effectiveness reduction with salinity
0
5
10
15
20
25
30
35
40
App, K
$T_{w,i}$ = 40 °C - 80 °C
MR = 0.5 - 3.0
NTU = 0.5 - 4.0
Slope = -0.0033 x App + 0.1324

List of Figures