Brine heat recovery unit for Multi Effect Desalination (MED) system based on Scheffler Concentrator using shell and coil heat exchanger

To cite this article: Parimal S. Bhambare et al 2018 IOP Conf. Ser.: Mater. Sci. Eng. 376 012009

View the article online for updates and enhancements.

Related content
- Decentralized stand-alone Multi Effect Desalination (MED) system using fixed focus type Scheffler concentrator for the remote and rural regions of Sultanate of Oman, Parimal S. Bhambare, M. C. Majumder and C. V. Sudhir
- A Numerical Analysis on a Compact Heat Exchanger in Aluminum Foam, B Buonomo, D Ercole, O Manca et al.
- Yeast fuel cell: Application for desalination, Unmy Mardiana, Christophe Innocent, Marc Crelin et al.
Brine heat recovery unit for Multi Effect Desalination (MED) system based on Scheffler Concentrator using shell and coil heat exchanger

Parimal S. Bhambare1*, M. C. Majumder2, Sudhir C. V.3
1Senior Lecturer, 2Professor, Mechanical and Industrial Engineering Department, Caledonian College of Engineering, Sultanate of Oman.
3Professor, Mechanical Engineering Department, National Institute of Technology, Durgapur.

* Corresponding Author: parimal.bhambare@gmail.com

Abstract: Three stage standalone solar thermal Multi Effect Desalination (MED) pilot plant for remote and arid rural regions of Sultanate of Oman using fixed focus type Scheffler concentrator producing 100 litres per day has been set up at author’s institute. Preliminary tests carried out on this plant yielded lower outputs of 60-65 litres per day. Heat loss through high temperature brine discharged at 85°C was observed as one of the reasons for lower yield. Literature reviewed and analysis shown that desalination outputs can be increased with increased feed water inlet temperature. To recover this heat and preheat the feed water supplied, brine heat recovery unit using counter flow type shell and coil heat exchanger has been proposed in this paper. Helically coiled heat exchanger has been selected for its compactness, higher heat transfer area per unit volume, higher inside heat transfer coefficient and lower fouling rates due to induced turbulence. The operating parameters of this heat exchanger will handle hot brine at 80-85°C and 1 bar pressure. In the present work, heat load, number of turns, coil length and pressure drop calculations are done for estimating the theoretical values based on heat transfer & fluid mechanics principles using numerical correlations from the literature reviewed. The steady state analysis has shown that that coil pitch,inside tube diameter, mass flow rate through the coil and shell affects the heat transfer rates. Practical dimensional constraints have been taken into consideration while finalising the dimensions of the heat exchanger. Overall effectiveness of 84.6% has been calculated for the proposed heat exchanger.

1. Introduction
Desalination is of prime importance for Sultanate of Oman for bridging the gap between the increasing fresh water demand and supply. All the plants installed in the country are uses fossil fuel for their operation. Sultanate of Oman, being tropical country, is blessed with abundance of solar energy with higher sky clearness index throughout the year. As per the literature reviewed and solar insolation measurements carried out at authors’ institute, Sultanate could be considered as one of the best destinations for solar thermal desalination. A pilot 100 litres/day three stage multi effect desalination plant operated using solar energy with Scheffler concentrators has been installed at author’s institute. Preliminary testing carried out on the plant shown lower yield between 60-65 litres/day. Investigations carried out indicated that heat lost through hot brine discharged between temperature 80-85°C is one of
the primary reasons for lower yield has been observed. With same heat input available, desalination output could be increased with increased feed water temperature using preheating. A brine heat recovery unit preheating the input feed water is designed for recovering this heat to enhance the overall desalination output from the plant. Counter - current type shell and coil heat exchanger has been considered for the application due to its compactness, higher heat transfer coefficients at lower flow rates expected in the system, higher heat transfer area per unit volume, lower cost and lower fouling rates due to induced turbulenceleading to comparatively lower cleaning requirements[1][2][15].Numerous researchers[1,3,4-8]have described the design procedure and empirical correlations for the shell and coil heat exchanger. Present paper describes a simplified step wise procedure for designing brine heat recovery unit using shell and coil heat exchanger. Effect of coil pitch, shell diameter, mass flow rate through the coil and shell for evaluating the effectiveness of the heat exchanger has been analysed in the paper. Based on the analysis and practical constraints dimensions have been finalized for the heat exchanger. Overall effectiveness of 84% has been calculated for the proposed the heat exchanger.

2. Multi Effect Desalination (MED) pilot plant using Scheffler Concentrators
A three stage MED pilot desalination plant that uses solar energy and produces 100 litres per day has been designed and set up at author’s institute. Scheffler concentrators have been employed for harnessing the solar energy. Distilled water has been used in the primary circuit that constitutes the concentrators and the header. The pressurized steam generated at the header is utilized for heating the sea water at first stage. The condensate is recirculated to the receivers of the Scheffler concentrators. Heat of condensation of the steam evaporated in the first stage is then utilised for heating the sea water in the second stage. The condensation heat of the steam generated in second stage is then utilised for heating the sea water in the third stage. The steam generated in third stage is directly collected as the desalination output alongwith condensate of steam from first and second stage. The brine collected from all the three stages is stored in a large tank and after cooling is discharged to sewage. First and second stage operates above atmospheric pressure while third stage is operated in vacuum. Figure 1 below shows the pictorial view of the system.

3. Design procedure for brine heat recovery unit
Presently hot brine coming out at temperature between 80-85°C at an average flow rate of 0.0041 kg/s from first two stages of the plant working for seven hours per day is directly discharged to surroundings.

Figure 1 Pictorial view of the installed desalination plant
A brine heat recovery heat exchanger is to be designed to recover the heat lost to the surroundings. The outlet temperature of brine from the heat exchanger is taken at 40°C. The last stage in the MED plant is operated in vacuum and hence the brine water discharged from last stage is not considered for the analysis. Figure 2 shows the schematic of shell and coil heat exchanger.

Figure 2 Shell and coil heat exchanger schematic

Hot brine water passes through the coils of the heat exchanger while the feed water passes through the shell. Copper tubes have been considered for coil while shell is constructed of mild steel material.

3.1 Coil side calculation:

The velocity of brine water in the pipe ($V_{bi}$) is calculated as,

$$V_{bi} = \frac{m_b}{\rho_b x A_t}$$

Where,

$$A_t = \frac{\pi}{4} \times d_i^2$$

Brine density, dynamic viscosity, thermal conductivity and specific heat based on its salinity, temperature and pressure can be calculated by using the relations given in[3]. The Reynolds number ($Re_c$) could be calculated as,

$$Re_c = \frac{\rho_{bi} V_{bi} d_i}{\mu_{bi}}$$

The Dean number ($De$) for the coil side fluid is calculated as,

$$De = Re_c \left( \frac{d_i}{D_c} \right)^{0.5}$$

The critical Reynolds number ($Re_{cr}$) for transition of laminar to turbulent flow is the function of coil parameters and can be calculated by the following relation [4].

$$Re_{cr} = 2300 \left[ 1 + 8.6 \left( \frac{d_i}{D_c} \right)^{0.45} \right]$$

Dimension less pitch ($\gamma$) is calculated as,

$$\gamma = \frac{p}{2\pi R_c}$$
Nusselt number for the coil side heat transfer \((Nu_i)\) can be calculated as follows \([5]\),

\[
Nu_i = 0.152D_i^{0.431} Pr^{1.06} y^{-0.277}
\]

The coil side convective heat transfer coefficient \((h_i)\) is calculated as,

\[
h_i = \frac{Nu_i k_i}{d_i}
\]

### 3.2 Shell side calculation:

The hydraulic or the equivalent diameter of the shell \((D_h)\) is calculated as\([5]\),

\[
D_h = \frac{D_s^2 - 2\pi R_s d_o^2 y^{-1}}{D_s + 2\pi R_s d_o y^{-1}}
\]

Shell side fluid velocity \((V_s)\) and Reynolds number \((Re_s)\),

\[
V_s = \frac{m_s}{\rho_s x A_s}
\]

Where,

\[
A_s = \frac{\pi}{4} x D_h^2
\]

\[
Re_s = \frac{\rho_s V_s D_h}{\mu_s}
\]

The Nusselt of shell side \((Nu_s)\) is calculated as \([5]\),

\[
Nu_s = 19.64 Re_s^{0.513} Pr^{0.129} y^{0.938}
\]

The shell side convective heat transfer coefficient \((h_s)\) is calculated as,

\[
h_s = \frac{Nu_s k_s}{D_h}
\]

### 3.3 Heat exchanger analysis:

The heat transfer on shell and coil side is,

\[
Q_s = m_s c_p s (T_{so} - T_{si}) = Q_c = m_c c_p c (T_{bi} - T_{bo})
\]

And the overall heat transfer is,

\[
Q = U_o A_{co} LMTD
\]

Where,

\[
\frac{1}{U_o A_{co}} = \frac{1}{A_{ci} h_i} + \frac{\ln(d_o/d_i)}{2\pi k_m L_c} + \frac{1}{A_{co} h_o}
\]

\[
A_{co} = \pi d_o L_c; \hspace{1cm} A_{ci} = \pi d_i L_c
\]

For cross flow of fluids in the tube and shell LMTD can be calculated as,

\[
LMTD = \frac{(T_{bo} - T_{fi}) - (T_{bi} - T_{fo})}{\ln \left( \frac{T_{bo} - T_{fi}}{T_{bi} - T_{fo}} \right)}
\]
From above equation the overall heat transfer coefficient can be calculated as,

\[
U_o = \frac{Q}{A_{co}LMTD}
\]

The number of turns for the coil (N) and the minimum length of the shell can be calculated as[6],

\[
L_c = N\sqrt{(2\pi R_c)^2 + p^2}
\]

The minimum height of the shell can be calculated from the following relation,

\[
L_s = N \cdot p
\]

3.4 Effectiveness calculation:
Heat transfer effectiveness is calculated using the following relation,

\[
\varepsilon = \frac{m_2C_{ps}(T_{so} - T_{si})}{(mC_p)_{min}(T_{ct} - T_{si})}
\]

3.5 Pressure drop calculation:
Friction factor [7] and pressure losses can be calculated as,

\[
f_c = 0.029 + 0.324 \times \left[ Re \left( \frac{d_i}{D_c} \right)^{2} \right]^{-0.25} \times \left( \frac{D_c}{d_i} \right)^{-0.5}
\]

\[
\Delta P_L = \frac{\rho f_c L V_c^2}{2 d_i}
\]

4. Effect of design parameters
Effect of different parameters: the coil pitch, inside coil diameter, mass flow rate through the coil and shell has been considered for the analysis. These parameters affect the overall performance of the heat exchanger. Final dimensions have been finalised based on the analysis and the practical constraints.

![Figure 3](image-url)  
**Figure 3** Effect of coil pitch on coil length and number of turns
4.1 Effect of coil pitch:
Coil pitch (p) affects the heat transfer parameters along with number of turns and coil length of the heat exchanger. Figure 3-5 shows the effect of coil pitch on coil length, number of turns, Nusselt number on coil & shell side and the overall heat transfer coefficient. The flow rates on coil and shell side have been kept constant along with the coil and shell diameters. Hot brine at 85°C enters the heat exchanger and leaves at 40°C while feed water enters the shell at 32°C from the preheater.

![Figure 4](image3.png)  
**Figure 4** Effect of coil pitch on tube side (Nui) and shell side (Nuo) Nusselt numbers

![Figure 5](image4.png)  
**Figure 5** Effect of coil pitch on overall heat transfer coefficient

Number of turns and the coil length increases sharply till coil pitch 0.06 m with the coil pitch. Nusselt number on shell side and the overall heat transfer coefficient increases with the coil pitch. This is due to the increased heat transfer rates due to increased contact between the shell side fluid and tube surface. The curvature of the coil increases with decreasing the coil pitch. This leads to increased secondary flow [8][9] in the tube which increases the Nui with decrease in the coil pitch. Overall heat transfer coefficient decreases at lower pitch values due to decreasing Nui. But as coil pitch increases, effect of increase in Nuo increases the overall heat transfer coefficient. This also affects the number of
turns and the coil length, which reduces with increase in coil pitch after initial rise up to coil pitch of 0.06 m.

4.2 Effect of inside diameter of the tube:
The coil pitch, shell diameter and mass flow rates through the coil & shell are kept fixed for the analysis. Figures 6 and 7 shows the variations in Nui, Nuo and overall heat transfer coefficient with inside diameter of the tube. Higher Nusselt number on coil side (Nui) is observed at lower tube diameters due to increased Reynolds number. As tube diameter increases, Nui decreases due to the reduction in the fluid velocity and subsequent Reynolds number for the fixed flow rate under consideration. Nusselt number on shell side (Nuo) increases with tube diameter. This is due to the increased surface area for heat transfer with increased tube diameter. The overall heat transfer coefficient increases with the inside tube diameter due to increased Nuo.

![Figure 6](image1.png)

**Figure 6** Effect of inside coil diameter on tube side (Nui) and shell side (Nuo) Nusselt numbers

![Figure 7](image2.png)

**Figure 7** Effect of inside coil diameter on overall heat transfer coefficient

4.3 Effect of brine water and feed water flow rates:
The coil pitch, inside coil diameter and shell diameter are kept fixed for the analysis. For fixed mass flow rate through the shell, as mass flow rate of water through the tube increases, Reynolds number of the flow in the tube increases leading to increased coil side Nusselt number, Nui and the overall heat
transfer coefficient. This increase the overall heat transfer inside the tube. For fixed mass flow rate through the tube, as mass flow rate of water through the shell increases, Reynolds number of the flow in the shell increases leading to increased shell side Nusselt number, Nu and the overall heat transfer coefficient. This increases the overall heat transfer coefficient on the shell side. This has been shown in Figures 8-11.

**Figure 8** Effect of brine water flow rate on tube side Nusselt number (Nui)

**Figure 9** Effect of brine water flow rate on overall heat transfer coefficient
5. Final design

As per the data collected, the practical constraints on the site and the formulae & procedure stated in the above sections, if the hot brine at an average mass flow rate of 0.0081 kg/s for seven hours of daily operation is to be cooled from 80-85°C to 40°C during heat recovery process, 1.524 kW is the total heat energy available for the recovery. This heat is absorbed by the feed water in the Brine heat recovery unit. Final dimensions of the heat exchanger are finalized based on the analysis and the practical constraints. Following table shows the final design details of the heat exchanger,

| Table 1 Final design details of the heat exchanger |
|--------------------------------------------------|
| Shell diameter, $D_s$ | 0.6 m |
| Tube inside diameter, $d_i$ | 25.3 mm |
| Tube outside diameter, $d_o$ | 28.6 mm |
| Diameter of helical coil, $D_c$ | 0.4 m |
| Number of turns, $N$ | 10 |
| Coil pitch, $p$ | 0.1 m |
| Tube length, $L_c$ | 14.8 m |
| Shell height, $L_s$ | 1.2 m |
| Curvature ratio, $d/D_c$ | 0.06325 |
| Parameter                              | Value       |
|----------------------------------------|-------------|
| Mass flow rate of brine water through the coil, $m_b$ | 29.24 kg/hr |
| Mass flow rate of feed water through the coil, $m_w$     | 56.93 kg/hr |
| Pressure losses                         | 20 Pa       |
| Overall heat transfer coefficient, $U$  | 85.163 W/m²K|
| Effectiveness                           | 84.6 %      |

6. Conclusion
In the present study design procedure for the brine heat recovery unit has been discussed. Effect of design parameters namely the coil pitch, inside coil diameter, mass flow rate through the coil and shell on the heat transfer parameters has been also presented in the paper. From the results obtained, following conclusions can be cited:

- As coil pitch is increased, the number of turns and coil length increases initially up to 0.06 m and decreases thereafter due to increasing overall heat transfer coefficient and Shell side Nusselt number.
- At lower coil pitch, higher coil side Nusselt number is observed which decreases due to the decrease in coil curvature with increasing coil pitch.
- By increasing the tube diameter, shell side Nusselt number and overall heat transfer coefficient increases while coil side Nusselt number decreases.
- Both the Nusselt numbers, $N_{ui}$ and $N_{uo}$ increases with increasing mass flow rates inside the tube and through the shell. The overall coefficient increases with the mass flow rates. This is due to the increased Reynolds number of the flow through the shell and the tube.
- Based on the above discussion, the maximum rate of heat transfer on both coil and shell side could be gained by highest possible coil pitch, inside tube diameter and the maximum mass flow rates through the shell & the tube.

Final dimensions for the brine heat recovery unit have been finalized to maximize the rate of heat transfer confirming to the practical limitations on the site. Overall effectiveness has been calculated to be 84.6% for the heat exchanger.

Acknowledgement
The research leading to these results has received Project Funding from The Research Council of the Sultanate of Oman under Research Agreement No. ORG/CCE/El/13/009. Authors would acknowledge the support and encouragement received from The Research Council as well as Caledonian College of Engineering, Muscat Sultanate of Oman.

References:
[1] B. W. Shende, P. K. Ghosh & Ramachandra K. Patil 1982 Designing a helical coil heat exchangerChem. Eng. 85–88.
[2] James R. Lines 2012 Helically Coiled Heat Exchangers [Online] Available: http://lms.i-know.com/pluginfile.php/28801/mod_resource/content/91/Helically Coiled Heat Exchangers offer Advantages.pdf.
[3] K. G. Nayar, M. H. Sharqawy, L. D. Banchik, and J. H. Lienhard2016 Thermophysical properties of seawater: A review and new correlations that include pressure dependence Desalination390 1–24.
[4] J. S. Jayakumar, S. M. Mahajani, J. C. Mandal, P. K. Vijayan, and R. Bhoi 2008 Experimental and CFD estimation of heat transfer in helically coiled heat exchangersChemical Engineering Research and Design86(3) 221–232.
[5] M. R. Salimpour 2009 Heat transfer coefficients of shell and coiled tube heat exchangers Exp. Therm. Fluid Sci.33(2), 203–207.
[6] G. Bonafoni and R. Capata 2015 Proposed Design Procedure of a Helical Coil Heat Exchanger
for an Orc Energy Recovery System for Vehicular Application Mech. Mater. Sci. &Eng. MMSE Journal.

[7] A. Siddique, A. Ghias, S. V. Ananth, M. Dev Anand, and G. Glan Devadhas 2016 Experimental Study of Thermal Performance of Coil in Shell Heat Exchanger Indian J. Sci. Technol. 9(13).

[8] L. Tang, Y. Tang, and S. Parameswaran 2016 A numerical study of flow characteristics in a helical pipe Adv. Mech. Eng. 8(7) 1–8.

[9] N. Jamshidi, M. Farhadi, D. D. Ganji, and K. Sedighi 2013 Experimental analysis of heat transfer enhancement in shell and helical tube heat exchangers Appl. Therm. Eng. 51(1–2) 644–652.

[10] Haraburda, and Scott 1995 Three-phase flow? Consider helical-coil heat exchangers Chemical Engineering 102 149-151.