International Food Research Journal 23(Suppl): S15 - S22 (December 2016)

Journal homepage: *http://www.ifrj.upm.edu.my*

Design and fabrication of a heat exchanger for portable solar water distiller system

Abduhamed, A.J., *Adam, N. M., Hairuddin, A. A. and Kareem, H. K.

Department of Mechanical Engineering, Faculty of Engineering, University Putra Malaysia, 43400 UPM Serdang, Selangor, Malaysia

Article history

Received: 15 October 2016
Received in revised form: 25 October 2016
Accepted: 28 October 2016

Keywords

Heat exchanger
Shell and tube
Condenser
Portable distiller
Solar distiller

Abstract

Heat exchanger H.X.s is the equipment used to transfer the thermal energy between two or more fluids at varying temperatures. The nature of this paper is an experimental study of the optimum design for shell and tube heat exchanger as a condenser with high productivity of drinking water for portable solar water distiller. The elaboration covers the aspects of considerations, design, fabrication, and test of the shell and tube H.X. as a portable condenser for solar water distiller system. The system consists of a portable stainless steel condenser, which is able to be dismantled and assembled without tools. The experimental result establishes that the condenser is able to produce 3.8 liter /hour of distilled water from vapor at 99.7˚C of inlet temperature and 4 liter/hour vapor flow rate, with 130 liter /hour as a condenser coolant water flow rate. The heat efficiency of the condenser can be increased by means of minimizing the tube's thickness and vapor inlet pressure. There is no back pressure effect on the system and the pressure drop in both sides of the condenser is reportedly of very low value and negligible, therefore, no need for a pressure pump is to be eliminated.

Introduction

Human life not only relies on food but also on water, oxygen, and shelter. Several countries have this growing issue with drinkable water supply especially in flood-stricken areas. In times ahead, the fresh water as a bare necessity will increase resulting from an enhanced living standard, climate change, and population growth as also a wide range of agriculture and industrial expansion activities (El-Bahi and Inan, 1999; Rijsberman, 2006). Sathyamurthy *et al*. (2014). The lack of drinking water is ongoing in spite of the fact that more than 70% of the earth is covered by water, because of about 97% of the earth's water is characteristically salty and impure (Moh'd A, 2015). To address this matter, the countries are using water antiseptic and solar distiller system both of which, are the widely utilized techniques in producing fresh water especially in countries with excellent source of solar energy.

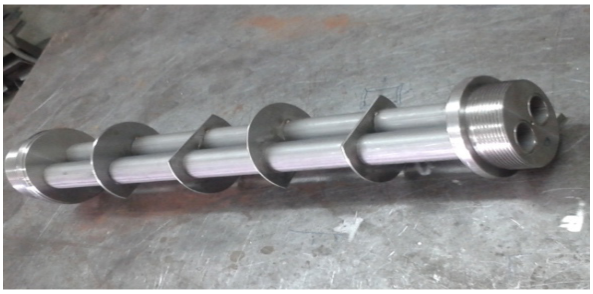
Solar distiller is an easy-to-handle device that utilise the solar irradiation to convert available brackish water into drinking water (Abdallah and Badran, 2008; Sathyamurthy *et al*., 2015). External condenser is helpful in enhancing the distilled water by reducing the heat loss Monowe *et al*. (2011); Kabeel *et al*. (2014), combine H.X. as a condenser with active solar still (parabolic trough collector) further leading to an increase the system efficiency. H.X.s is the equipment that transfers the thermal energy between two or more fluids at varying temperatures. To examine the portable condition for distiller, the most usual choice being the shell and tube H.X. which known to boast off flexibility, accuracy and high efficient heat transfer between two fluids at unrelated temperature. H.X. physical size is very significant in a number of applications. Therefore, the use of the shell and tube type is more suitable for small place, high heat transfer, and high capacity although the H.X. like double pipe type can be cheaper (Hewitt and Pugh 2007; Zhou *et al*., 2015). Furthermore, shell and tube H.X. is also known for its ease of maintenance, upgrading possibility, and powerful construction geometry. In this study, the system consists of stainless steel H.X. as a condenser to convert steam into water with the tank for distiller water accumulation.

The study intent is to fabricate the optimum design for H.X. as a condenser with high productivity drinking water for portable solar water distiller. The system can also be dismantled and assembled without tools (as shown in Figure 1). The condenser length should not surpass 0.6 meter and maximum weight not more than 8 kg (including the inside water). The

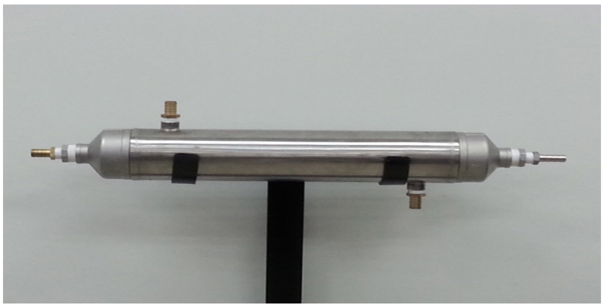
*Corresponding author.
Email:*mariah@upm.edu.my*

limitations worth considering would be the type of material, device dimension, mechanism of heat transfer, arrangement of flow, safety, geometry and economic.

(a)



(b)



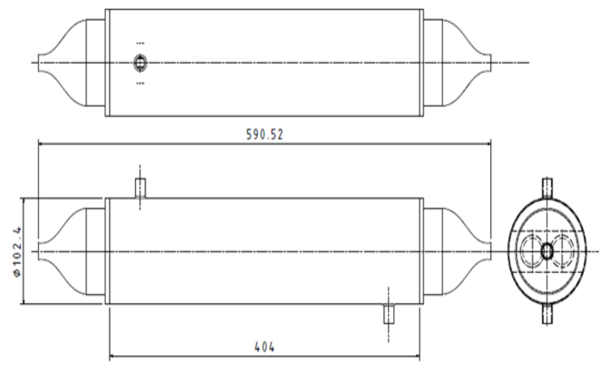
(c)

Figure 1. Two head and baffles of shell tube heat exchanger

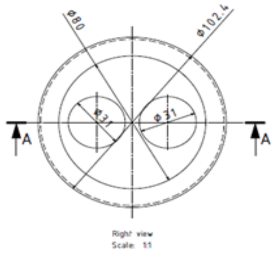
Materials and Methods

Device description

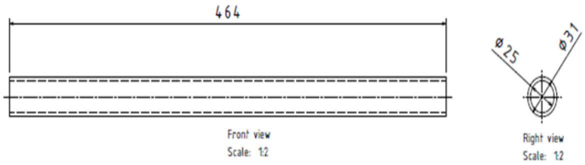
To be specific, the device is (E type) shell and tube condenser with one pass parallel flow Kakac (2012). The tube length (L) measures 40 cm and the whole condenser length is 0.59 m (see Figure 2), it has weigh of 7.5 kg included inside water. The water vapour inside the tube has inside and outside with the diameter of 25 mm and 31 mm, respectively. The big tube diameter will help facilitate the cleaning and increase the surface area which will lead to the increasing Nusselt number. The shell diameter is 100 mm and it has four segment baffles to force the water to flow cross the tube and make the turbulence to perform the heat transfer (as shown in Figure 1). Moreover, it supports the tube bundle to prevent vibration, according to Kakac (2012). The coolant water flowing inside shell is brackish water. In the shell and tube condenser, segment baffles are very much responsible for the higher fouling rate. It will lead to the accumulation of the fouling film on the heat transfer surface. Therefore, slowly and steadily, the H.X. performance will drop. In this study, the water fouling resistance was selected to be 0.000528 m^2/K. W for water velocity < 0.9 (m/s) and heating temperature < 115 (Kakac, 2012).



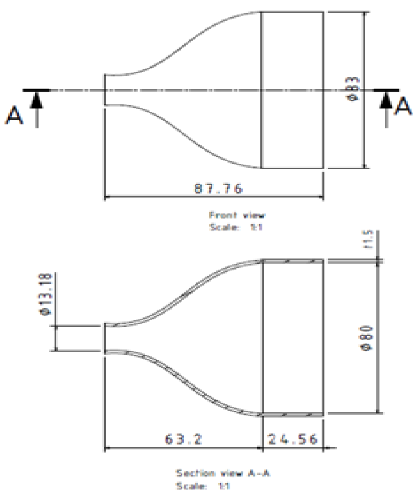
(a)



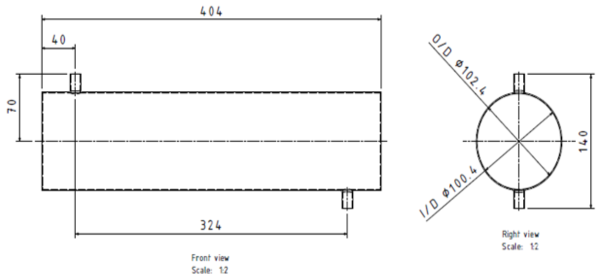
(b)



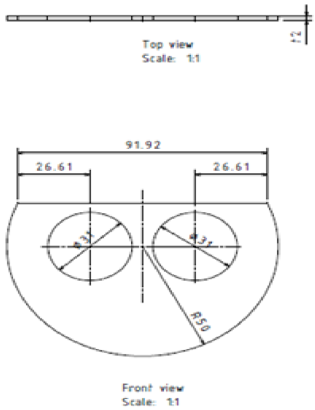
(c)



(d)



(e)



(f)

Figure 2. Schematic diagram for H.X. with all dimensions

Select of material

The material chosen to fabricate the condenser should have high heat transfer properties (high thermal conductivity) for example copper, aluminium and stainless steel. In this study, several limitations were considered upon choosing the material because of the exclusive use of the condenser for drinking water. Stainless steel (304) material with thermal conductivity (k= 14.9 W/m.K) was used to fabricate the condenser as a safe, strong, economic and corrosion-resistant material (Disegi and Eschbach, 2000;. Reynolds *et al*., 2003; Pugh *et al*., 2009; Nie *et al*., 2011).

Experimental set up

When it comes to devising a shell and tube condenser for portable solar water distiller system, the volume flow rate (V_c) and inlet temperature of coolant water ($T_{c,i}$) and vapour ($T_{h,i}$) are the important aspects to be considered. The outlet temperature of coolant water ($T_{c,o}$) and distilled water ($T_{h,o}$) were estimated with energy balance from thermodynamics first law for the open system under the steady state condition, a steady flow with negligible kinetic and potential energy changes (Kakac, 2012).

As the initial design for the condenser, some hypothetical data have been considered based on the study results of parabolic trough collector (PTC) obtained earlier. The condenser inlet vapour temperature and flow rate (V_h) are 100˚C and 1.6 litre/hour respectively. While the condenser coolant water inlet temperature was considered as 31˚C based on the average temperature in the city pipe line flow which was between 28-33˚C (Rahman *et al*., 2011) with the volume flow rate of 216 litre/hour. The coolant water flew in shell side while the vapour flew in the tube side. The primary design follows the log mean temperature different (LMTD) method based on tumbler exchanger manufacture association (TEMA) slandered.

Two ultimate individual coefficients of heat transfer should be estimated. According to Kakac (2012), the heat transfer coefficient for coolant water h_o is estimated at 5000 W/m².K for boiling heat transfer and for the vapour h_i at 9000 W/m².K for the condensation heat transfer. Furthermore, the tubes were developed on a square pitch with the ratio of pitch (PR) of 1.25. The shell side has been four segmental baffles spacing (B) which is approximately 0.8 of the shell diameter and 25% as baffle cut (B_c). The tube count calculation constant (CTP) was chosen as 0.93 for one tube pass, and tube layout constant (CL) was suggested to be equal one for 90˚ and 45˚ (Tan and Fok, 2006; Kakac, 2012)

Determination of heat flux (Q)

In heat exchangers (condenser) the fundamental thermal analysis equation under the condition of steady state can be written as Bhardwaj *et al*. (2013)

$$Q = mC_p(T_{h1} - T_{h2}) + mL_v = mC_p(T_{c2} - T_{c1}) \qquad (1)$$

Determination of overall heat transfer coefficient (U)

To determine the overall heat transfer coefficient, two individual coefficient of heat transfer should be estimated and the overall heat transfer coefficient (U) was calculated as Kakac (2012)

$$\frac{1}{u_f} = \frac{1}{h_o} + \frac{r_o}{r_i} \times \frac{1}{h_i} + R_f + \frac{r_o}{k} \times \ln\left(\frac{r_o}{r_i}\right) \qquad (2)$$

$$\frac{1}{u_f} = \frac{1}{h_o} + \frac{r_o}{r_i} \times \frac{1}{h_i} + \frac{r_o}{k} \times \ln\left(\frac{r_o}{r_i}\right) \qquad (3)$$

Determination number of tube

There are two different conditions that must be considered for having reasonable design: Pugh *et al*. (2009)

- Ideal condition to calculate inner surface area (A_c)

In this case the condensing process happens in single phase (vapour); without any fouling resistance factor for coolant water.

- Actual condition to calculate inner surface area (A_f)

In this case the fouling resistance factor for coolant water is deliberated and the condensing process takes place in two phase (vapour, water).

Most importantly, the reasonable design of condenser can be modified through the comparison between two different condition results.

According to Kakac (2012), A_f and A_c can be calculate from these equations (Mohanty and Singru, 2011; Kakac, 2012).

$$A_f = \frac{Q}{(U_f \,.\, \Delta T_m)} \tag{4}$$

$$A_c = \frac{Q}{(U_c \,.\, \Delta T_m)} \tag{5}$$

This enables the temperature difference ΔT_m in (°C or K) can be calculated from Kakac (2012)

$$\Delta T_m = F.\Delta T_{1m} \tag{6}$$

Where: the correction factor $F = 1$ in case of parallel flow and (ΔT_{1m}) indicates the logarithm-mean temperature difference. (ΔT_{1m}) can be calculated from these equation Kakac (2012); Fettaka *et al*. (2013); Zhou *et al*. (2015)

$$\Delta T_{lm} = \frac{(\Delta T_1 - \Delta T_2)}{\ln(\Delta T_1 - \Delta T_2)} \tag{7}$$

As well

$$\Delta T = T_{h1} - T_{c1} \tag{8}$$

$$\Delta T = T_{h2} - T_{c2} \tag{9}$$

Last but not least, the diameter of shell side and number of tube inside the shell can be calculate from (Tan and Fok, 2006; Kakac, 2012).

$$D_s = 0.637\left(\frac{CL}{CTP}\right)^{0.5} .\, A_f .\left(\frac{(PR)^2 d_o}{L}\right)^{0.5} \quad 10)$$

$$N_t = 0.785\frac{\frac{CTP}{CL} .\, D_s^2}{((PR)^2 . d_o^2)} \quad 11)$$

Determination of modified heat transfer coefficient (h_m)

The quiescent vapour was treated in case of laminar film condensation by modified Nusselt equation In the horizontal laminar flow (Louahlia-Gualous and Mecheri, 2007; Jassim *et al*., 2008).

$$h_m = \Omega\left[\frac{\left(k_l^3 \,.\, \rho_l\left(\rho_l - \rho_g\right) . g . i_{fg}\right)}{(\mu_l . d_o \,.\, \Delta T)}\right]^{0.25} \quad 12)$$

Where: ($\Delta T = T_{sat} - T_w$) .

Jassim, Kosky and Jaster have shown further that the coefficient (Ω) has a link with the vapour ( ) void fraction (Jaster and Kosky, 1976; Jassim *et al*., 2008).

$$\Omega = 0.728\,\alpha_g^{\left(\frac{3}{4}\right)} \tag{13}$$

As well

$$\alpha_g = \frac{1}{\left\{1 + \left[\frac{(1-x)}{x}\right] . \left(\frac{\rho_g}{\rho_l}\right)^{\frac{2}{3}}\right\}} \tag{14}$$

Pressure drop in shell side

To select an appropriate pressure drop equation, the Reynolds number should be counted from the equation below (Patel and Rao, 2010; Kakac, 2012; Yang and Fan, 2014; Yang and Oh, 2014).

$$Re = \frac{(G_s . D_e)}{\mu} \tag{15}$$

Where:

$$D_e = 4\frac{\left(P_T^2 - \frac{\pi\, d_o^2}{4}\right)}{(\pi\, d_o)} \tag{16}$$

$$C = P_T - d_0 \tag{17}$$

$$A_s = \frac{(D_s\, CB)}{P_T} \tag{18}$$

$$G_s = \frac{\dot{m}}{A_s} \tag{19}$$

The pressure drop in the shell side is determined by the length of tube, number of tubes inside shell, and number of baffles. Leaning on Kern method, the shell side pressure drop is calculated using the following expression (Tan and Fok, 2006; Kakac, 2012).

$$\Delta p_s = \frac{(f\, G_s^2 (N_b + 1) . D_s)}{(2\, \rho\, D_e \emptyset_s)} \tag{20}$$

Where the relation between friction factor *f* and Reynolds number Re correlated in the laminar region is as follows:

$$f = e^{(0.576 - 0.19.\ \ln(Re))} \qquad (21)$$

And

$$N_b = \frac{L}{B} - 1 \qquad (22)$$

$$\emptyset_s = \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (23)$$

Where (N_b + 1) is number of times that the shell fluid pass the tube bundle; μ_w is the dynamic viscosity at the average wall temperature (K).

$$T_{w(average)} = \frac{T_{b,h} + T_{b,c}}{2} \qquad (24)$$

Pressure drop in tube side

Reynolds number is prime value to the definition of the pressure drop equation. The Reynolds number for the tube side can be calculated using the expression: (Kakac, 2012; Yang and Fan, 2014; Yang and Oh, 2014).

$$Re = \frac{\rho . u_m . d_i}{\mu} \qquad (25)$$

Where:

$$u_m = \frac{\dot{m}_t}{\rho_t . A_i} \qquad (26)$$

$$A_i = \left(\frac{\pi d_i^2}{4}\right) \times N_t \qquad (27)$$

According to Kakac (2012), the pressure drop inside the circular tubes, can be calculated by using the relationship between Reynolds number (Re) and training friction factor (f) for laminar flow, which is free from the surface roughness.

$$f = \frac{16}{Re} \qquad (28)$$

The expression of the pressure drop below can be used for both laminar and turbulent flow:

$$f = \frac{\Delta P_t}{\left[4\left(\frac{L}{d_i}\right) . \left(\frac{\rho\, u_m^2}{2}\right)\right]} \qquad (29)$$

Results and Discussion

The energy balance equation (1) serves to calculate the balance temperature (Tb). Water properties can be estimated as we refer to the inlet vapour and coolant water temperature not to mention the balance temperature (Tb) as show in Table 1. Thermal calculation (heat flux *Q* and over all heat transfer coefficient *U*) and mechanical design including shell diameter (D_s) and number of tubes (N_t) were obtained in this study. Also, Reynolds number (Re) is calculated for two sides shell and tube of condenser to make identification of the suitable pressure drop equation, friction factor, and the type of flow laminar or turbulent as illustrated in Table 1. The experimental test shows that the fluid flow in both shell and tube side is laminar.

The calculation in Table 1 considered the vapour flow inside tube as one phase (gas) only. This hypothesis is imprecise and should be adjusted as new condition. The actual condition weighed upon the flow inside tubes as two phase (gas-liquid),

Table 1. Primary design (thermal and physical design)

$T_{h,i}$ (°C)	$T_{c,i}$ (°C)	T_b (°C)	V_h (L/h)	V_c (L/h)	ρ_h (kg/m^3)	ρ_c (kg/m^3)	m_h (kg/s)	m_c (kg/s)
100	31	35.5	1.667	216	957.9	995.6	0.000443	0.0597

$C_{p,h}$ (kJ/ kg. K)	$C_{c,i}$ (kJ/ kg. K)	L_v (kJ/ kg)	Q (kW)	h_i $(W/m^2.K)$	h_0 $(W/m^2.K)$
4.217	4.178	2260	1.1228	9000	5000

R_f $(m^2.K/W)$	D_0 (mm)	D_i (mm)	k (W/m. K)	U_f $(W/m^2.K)$	U_c $(W/m^2.K)$
0.000528	31	25	14.9	894.28	1694.31

ΔT_1 (°C)	ΔT_2 (°C)	ΔT_{lm} (°C)	F	ΔT_m (°C)	A_f (m^2)	A_c (m^2)	B (m)
64.5	4.49	22.5	1	22.5	0.0557	0.0294	0.08

N_B	D_0 (m)	CTP	CL	L (m)	D_s (m)	N_t
4	0.0123	0.93	1	0.4	0.054259	1.4313

therefore, the modified Nusselt equation (Louahlia-Gualous and Mecheri, 2007). Jassim *et al*. (2008) for a quiescent vapour inside horizontal tube in case of laminar film condensation Equation (12) had functioned to determine the vapour individual heat transfer (h_m) with new number of tube and shell diameter for variable vapour quality (x). 99.7 Temperature of vapour, 4 litre/hour vapour flow rate and condenser coolant water flow rate in Table 2 are the actual data gathered through the experimental test, then it used the shell and tube H.X. equations to gauge on the thermal, mechanical and hydraulic designs for comparison with the primary design. It is found that the flow is still laminar in both side shell and tube and the calculations illustrate a big approach with primary design with regard to the mechanical design as shown in Table 2.

Table 2. Experimental and modified result for different vapour quality

$T_{h,i}$ (°C)	$T_{h,o}$ (°C)	$T_{c,i}$ (°C)	$T_{c,o}$ (°C)	V_h (L/h)	V_c (L/h)	L_v (kJ/kg)	Q (W)
99.7	38.3	31	32.5	4	216	2260	1.139

x	h_m (W/m².K)	h_o W/m².K)	R_f (m².K/W)	D_o (mm)	D_i (mm)	k (W/m. K)
1	8961.69	5000	0.000528	31	25	14.9
0.9	8959.12	5000	0.000528	31	25	14.9
0.8	8955.9	5000	0.000528	31	25	14.9
0.7	8951.7	5000	0.000528	31	25	14.9
0.6	8946.25	5000	0.000528	31	25	14.9
0.5	8938.56	5000	0.000528	31	25	14.9
0.4	8927.04	5000	0.000528	31	25	14.9
0.3	8907.9	5000	0.000528	31	25	14.9
0.2	8869.9	5000	0.000528	31	25	14.9
0.1	8758.37	5000	0.000528	31	25	14.9

U_f (W/m².K)	U_c (W/m².K)	ΔT_1 (°C)	ΔT_2 (°C)	ΔT_{lm} (°C)	F	ΔT_m (°C)	A_f (m²)	A_c (m²)
917.313	1778.92	61.4	1.5	16.13	1	16.13	0.076957	0.039683
917.279	1778.79	61.4	1.5	16.13	1	16.13	0.076959	0.039686
917.237	1778.63	61.4	1.5	16.13	1	16.13	0.076963	0.039679
917.184	1778.43	61.4	1.5	16.13	1	16.13	0.076968	0.039694
917.112	1778.16	61.4	1.5	16.13	1	16.13	0.076974	0.039700
917.011	1777.78	61.4	1.5	16.13	1	16.13	0.076982	0.039708
916.861	1777.22	61.4	1.5	16.13	1	16.13	0.076995	0.039721
916.610	1776.28	61.4	1.5	16.13	1	16.13	0.077016	0.039742
916.110	1774.40	61.4	1.5	16.13	1	16.13	0.077058	0.039784
914.618	1768.81	61.4	1.5	16.13	1	16.13	0.077183	0.039910

CTP	*CL*	*L* *(m)*	D_s *(m)*	N_t	ΔP_t *(Pa)*	ΔP_s *(Pa)*
0.93	1	0.4	0.063765	1.97686	0.0039494	6.9478
0.93	1	0.4	0.063766	1.97693	0.0039492	6.9477
0.93	1	0.4	0.063767	1.97702	0.0039490	6.9476
0.93	1	0.4	0.063769	1.97713	0.0039488	6.9474
0.93	1	0.4	0.063772	1.97729	0.0039485	6.9472
0.93	1	0.4	0.063775	1.97750	0.0039481	6.9469
0.93	1	0.4	0.063780	1.97783	0.0039474	6.9464
0.93	1	0.4	0.063789	1.97837	0.0039463	6.9457
0.93	1	0.4	0.063806	1.97945	0.0039442	6.9441
0.93	1	0.4	0.063859	1.98268	0.0039378	6.9395

The comparison drawn between the primary design and experimental test is used to verify the output results. By contemplating on the conditions such as the system accuracy, condition of working and product standard that have some powerful effects on the error in design, the comparison points to a big approach between mechanical design and experimental result with error less than 15%.

Figures 3 (a), (b), and (c) show that the increase in the vapour quality (x) leads to an increase in value of the modified heat transfer coefficient, and a decrease in the shell diameter and number of tube, and yet, the effect of change in vapour quality on this value is very minimal.

Finally, hydraulic calculations suggest that the pressure drop for both side shell and tube increased with high vapour quality (see Figures 3 (d) and (e)) as well as the fact that the maximum pressure drop in shell and tube sides was $\Delta P_s = 6.9478$ Pa and $\Delta P_t = 0.0039494$ Pa respectively. The pressure drop for both the side shell and tube are in acceptable rang (less than 20 Pa), and the value is found to be small and negligible (Kakac, 2012).

Conclusion

From the experimental results, the E-type stainless steel material shell and tube heat exchanger (condenser) with two tubes at 25 mm inside diameter, 100 mm shell diameter, and 400 mm length of tube can well produce 3.8 litre/ hour of distilled water from vapour at 99.7˚C of inlet temperature in the atmospheric pressure and 4 litre/hour of inlet vapour volume flow rate with 216 litre/hour of coolant water flow rate. The experimental test further demonstrates that reducing the condenser coolant water flow rate to 130 litre/hour does not leave any impact on the system productivity. The heat efficiency of condenser can be increased by trying to reduce the tubes thickness and vapour inlet pressure. Still depending on the experimental results, the flow in both sides shell and tube of heat exchanger are laminar flow. Additionally, it is not back pressure effect on system and the pressure drop in two sides of heat exchanger have very low value and negligible, therefore, it removes the need for a pressure pump.

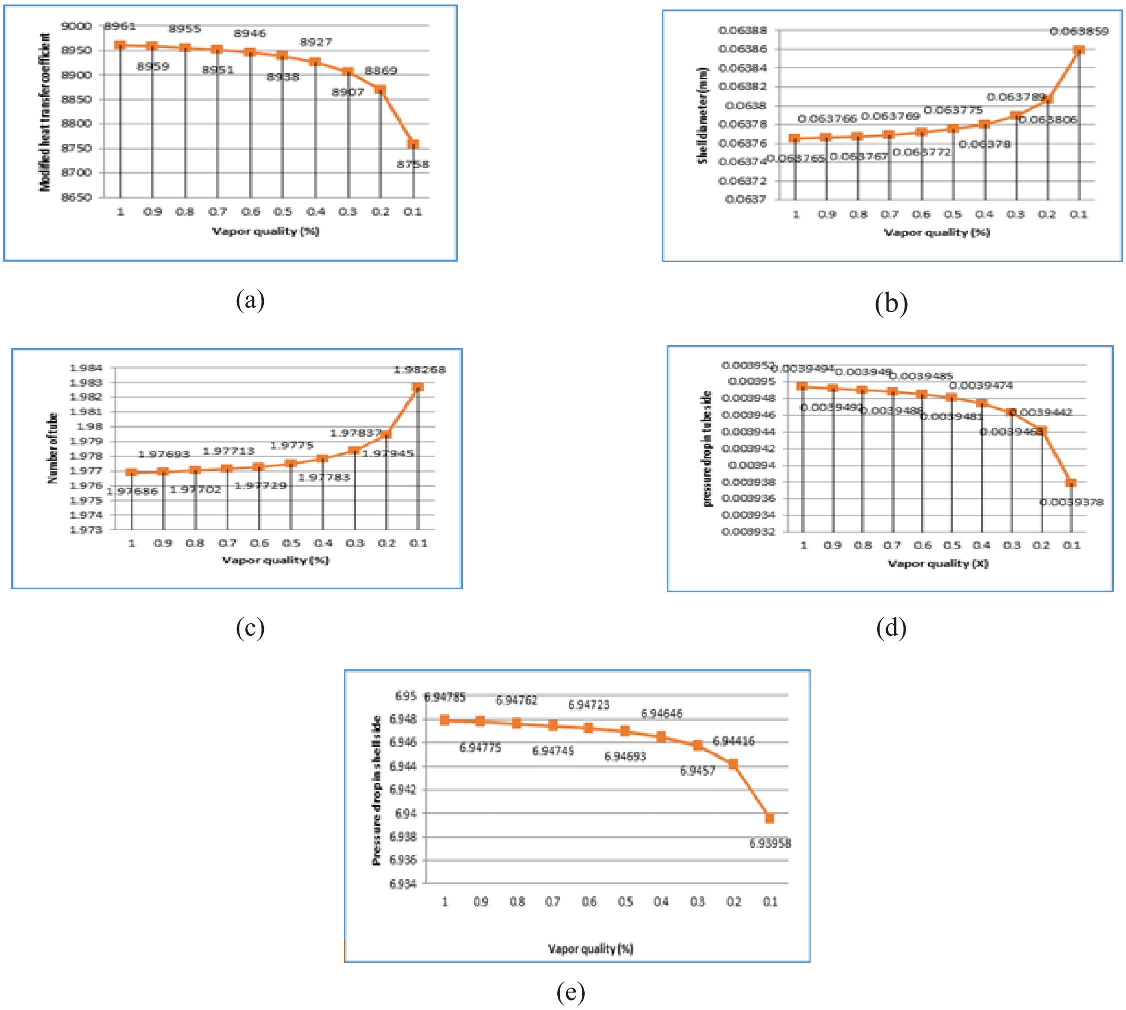


Figure 3. (a) Effect of vapour quality on vapour individual heat transfer; (b) Effect of vapour quality on shell diameter; (c) Effect of vapour quality on number of tube; (d) Effect of vapour quality on pressure drop in tube side; (e) Effect of vapour quality on pressure drop in shell side

Acknowledgement

The financial support by Putra grant for post graduate (IPS) is highly acknowledged.

References

Abdallah, S. and Badran, O.O. 2008. Sun tracking system for productivity enhancement of solar still. Desalination 220(1-3): 669–676.

Bhardwaj, R., Kortenaar, M.V. and Mudde, R.F. 2013. Influence of condensation surface on solar distillation. Desalination 326: 37–45.

Disegi, J. A. and Eschbach, L. 2000. Stainless steel in bone surgery. Injury 31(Suppl. 4).

El-Bahi, A. and Inan, D. 1999. Analysis of a parallel double glass solar still with separate condenser. Renewable Energy 17(4): 509–521.

Fettaka, S., Thibault, J. and Gupta, Y. 2013. Design of shell-and-tube heat exchangers using multiobjective optimization. International Journal of Heat and Mass Transfer 60(1): 343–354.

Hewitt, G.F. and Pugh, S.J. 2007. Approximate Design and Costing Methods for Heat Exchangers. Heat Transfer Engineering 28(2): 76–86.

Jassim, E.W., Newell, T. and Chato, J. 2008. Prediction of two-phase condensation in horizontal tubes using probabilistic flow regime maps. International Journal of Heat and Mass Transfer 51(3-4): 485–496.

Jaster, H. and Kosky, P.G. 1976. Condensation Heat Transfer in a Mixed Flow Regime. International Journal of Heat and Mass Transfer 19: 95–99.

Kabeel, A. E., Omara, Z.M. and Essa, F. 2014. Enhancement of modified solar still integrated with external condenser using nanofluids: An experimental approach. Energy Conversion and Management 78: 493–498.

Kakac, S. 2012. In Liu, H. and Pramuanjaroenkij, A. (Eds). Heat exchangers: selection, rating, and thermal design. 3rd ed., p. 1-615 Boca Raton: CRC Press.

Khwaji, A.D., Kutubkhanah, I.K. and Wie, J.M. 2008. Advances in seawater desalination technologies. Desalination 221(1-3): 47–69.

Louahlia-Gualous, H. and Mecheri, B. 2007. Unsteady steam condensation flow patterns inside a miniature tube. Applied Thermal Engineering 27(8-9): 1225–1235.

Moh'd A. 2015. Modeling of a novel concentrated solar still enhanced with a porous evaporator and an internal condenser. Solar Energy 114: 8–16.

Mohanty, D.K. and Singru, P.M. 2011. Use of C-factor for monitoring of fouling in a shell and tube heat exchanger. Energy 36(5): 2899–2904.

Monowe, P., Masale, M. and Nijegorodov, N. 2011. A portable single-basin solar still with an external reflecting booster and an outside condenser. Desalination 280(1-3): 332–338.

Nie, F.L., Wang, S.G., Wang, Y.B, Wie, S.C. and Zheng, Y.F. 2011. Comparative study on corrosion resistance and in vitro biocompatibility of bulk nanocrystalline and microcrystalline biomedical 304 stainless steel. Dental materials : official publication of the Academy of Dental Materials 27(7): 677–683.

Patel, V.K. and Rao, R.V., 2010. Design optimization of shell-and-tube heat exchanger using particle swarm optimization technique. Applied Thermal Engineering 30(11-12): 1417–1425.

Pugh, S.J., Hewitt, G.F. and Muller, S.H. 2009. Fouling During the Use of "Fresh" Water as Coolant—The Development of a "User Guide." Heat Transfer Engineering 30(10-11): 851–858.

Rahman, A., Malek, A., Khalid, H.L. and Yusup, Y. 2011. Optimizing Wind Power for Energy Efficient Building Design in Tropical Hot-humid Climate of Malaysia. Journal of Sustainable Development 4(2): 217–224.

Reynolds, A., Tang, W., Gnaupel, H.T. and Prask, H. 2003. Structure, properties, and residual stress of 304L stainless steel friction stir welds. Scripta Materialia 48(9): 1289–1294.

Rijsberman, F.R. 2006. Water scarcity: Fact or fiction? Agricultural Water Management 80(1): 5–22.

Sathyamurthy, R., Nagarajan, P.K., Subramani, J. and Vijayakumar, D. 2014. Effect of water mass on triangular pyramid solar still using phase change material as storage medium. Energy Procedia 61: 2224–2228.

Sathyamurthy, R., El-Agouz, S.A. and Dharmaraj, V. 2015. Experimental analysis of a portable solar still with evaporation and condensation chambers. Desalination 367: 180–185.

Tan, F.L. and Fok, S.C. 2006. An educational computer-aided tool for heat exchanger design. Computer Applications in Engineering Education 14(2): 77–89.

Yang, J. and Fan, A. 2014. Optimization of shell-and-tube heat exchangers conforming to TEMA standards with designs motivated by constructal theory. Energy Conversion and Management 78: 468–476.

Yang, J. and Oh, S.R. 2014. Optimization of shell-and-tube heat exchangers using a general design approach motivated by constructal theory. International Journal of Heat and Mass Transfer 77: 1144–1154.

Zhou, J. F., Wu, S.W., Chen, Y. and Shao, C.I. 2015. Semi-numerical analysis of heat transfer performance of fractal based tube bundle in shell-and-tube heat exchanger. International Journal of Heat and Mass Transfer 84: 282–292.