Effect of Tube Thickness for Shell and Tube Heat Exchanger in Portable Solar Water Distiller

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***Abstract*—Distillation is the process of water purification that utilized a source of heat to vaporize water and separate it from pollutants and other impurities. Then guide the vapor into a condenser to convert it to the liquid form. This process achieved by a simple device which uses solar energy as a heat source and called solar distiller. This study present experimental comparison and mathematical approach for pressure drop ΔP between four separate heat exchanger H.Es (A, B, C, and D) as a condenser in a portable solar water distiller. The heat exchangers were fabricating from stainless steel material with same physical specifications and different in thickness or length of tube. The H.Es able to be assembled and dismantled without tools. The experimental test show that all H.Es produce the same amount of distilled water as 22.8 liter/day. The experimental and mathematical results proved that the optimum design is heat exchanger D which has lower tube thickness and length. Further, reducing thickness of tube will increase pressure drop ΔP in shell and tube sides. While reducing length of tubes will lead to reduction of pressure drop in tube side and increasing ΔP in shell side with same number of baffles. However, the effect of reducing length of tube on reduction ΔP in tube side and increase ΔP in shell side is more than the effect of thickness reductions. However, the results proved that the systems able to dispensing on pumping power, because of lower value of ΔP. In addition, reducing thickness or length of tube will lead to reduce fabrication cost.**

Keywords—Heat Exchanger; Solar Distiller; Shell And Tube; Pressure Drope; Tube Thickness

I. INTRODUCTION

Several countries are suffered from lower drinkable water supply especially in flooded areas. Growth in agriculture, industrial, and increase the water source pollution by industry wastes will lead to increase required of fresh water in future [1–5]. Therefore, countries tended to use solar distiller to overcome this problem.

Solar distiller is the equipment convert fresh water to drinking water by using solar energy as a heat source [6,7]. The distilled process including two stages, evaporating water by using solar radiation and condensing the pure vapor by using Heat exchanger device.

Heat exchangers are the device use to transfer heat from one streaming fluid to another through solid partition separating these fluids [8–13]. It is important to reduce weight of H.E in order to investigate the condition of portable water distiller, with pay attention to pressure drop which should be small value, in order dispensing on pumping power. The system consist of the vapor source and stainless steel heat exchanger as a condenser with separate tank for accumulation production water. The Heat exchangers able to be assembled and dismantled without tools and will be easy cleaning [14].

The aims of this study is to achieve economic heat exchanger with higher heat efficiency by considering change in some dimensions. Another goal, is to study effect of the reduce thickness, length, or both to gather on pressure drop in both side shell and tube.

II. MATERIAL AND METHOD

A. Device Description

The study presented four (E type) shell and tube heat exchangers with, one pass parallel flow [15,16] and two 25 mm inside diameter tubes. The tubes was arrangement on a square pitch with pitch ratio (PR) 1.25. In each heat exchangers the vapor enters the tube side with 4 liter/hour as a volume flow rate. while 130 liter /hour as volume flow rate of coolant water flow in shell side which has 100 mm inside diameter with four segment baffles to guide the water flow cross the tube for enhancing heat transfer, as well as, supporting the tube bundle to prevent vibration [14,17,18]. Segmental baffle and H.E manufacture stages are shown in fig. 1.

The four heat exchangers which is used in this study are different with each other in thickness and length of tube with same produce distilled water. The heat exchangers (A) has two tubes with length and outside diameter as 0.4 m and 31 mm respectively. The heat exchangers (B) modified to 28 mm tubes outside diameter with 0.4 m tube length. The heat exchangers (C) fabricated with 31 mm and 0.3 m as tube outside diameter and length of tube respectively. The heat exchangers (D) was built with 28 mm outside tube diameter and 0.3 m as a tube length. The big inside tube diameter will help to ease cleaning and increase in surface area which will lead to increase heat transfer coefficient.

B. Select of Material

Several limitation such as hardness, yield stress, tensile stress, corrosion resistance, healthy, and cost, as well as, material properties like thermal conductivity k, should be considered to select type of material. Therefore, Stainless Steel (304) with (k= 14.9 W/m. K) [19] as thermal conductivity was used to fabricate heat exchangers as a healthy [20], corrosion resistance material, strong, and economic material [21,22].

C. Experimental set up

When the flow velocity in the nozzle of shell side exceeds a limit, the vibration happen in the tubes, specially, after reducing tubes thickness, the size of nozzle should be considered to solve this problem. Therefor Saunders (1990), calculated the minimum inside nozzle diameter of shell by [23]:

$$D_{ns} = \sqrt{\frac{\dot{m}}{(\pi/4)\rho_s V_{ns}}} \quad (1)$$

$$V_{ns} = \sqrt{\frac{2250}{\rho_s}} \quad (2)$$

Where:

D_{ns}: Minimum inside nozzle diameter in shell side (mm)
$\dot{m}$: Mass flow rate in shell side (kg/sec)
ρ_s: Fluid density in shell side (kg/m³)
V_{ns}: Nozzle flow velocity (m/sec)

Reynolds number is the criteria of flow, whether it is laminar or turbulent, and this will lead to select suitable equation for friction factor f and pressure drop, the Reynolds number in shell side was expressed as [15,23–26]:

$$Re_s = \frac{(G_s . D_e)}{\mu_s} \quad (3)$$

Where:

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

$$C = P_T - d_o \quad (5)$$

$$A_s = \frac{(D_s C B)}{P_T} \quad (6)$$

$$G_s = \frac{\dot{m}}{A_s} \quad (7)$$

Where:

Re_s: Reynolds number
D_e: Equivalent diameter (m) for square pitch layout
μ_s: Fluid dynamic viscosity in shell side (kg/m².s)
d_o: Outside tube diameter (m)
P_T: Tube pitch (m)
D_s: Shell diameter (mm)
C: Clearance between tubes (constant numbers)
B: Baffle space (m)

The pressure drop in shell side of H.E depends on number of baffles, number of tubes inside shell, and length of tube. Therefore, the shell side pressure drop can be calculation based on Kern method by following expression [15][23]

$$\Delta p_s = \frac{(f G_s^2 (N_b + 1). D_s)}{(2\rho D_e \emptyset_s)} \quad (8)$$

Where the relation between Reynolds number Re and friction factor f for laminar region is

$$f = e^{(0.576-0.19.ln(Re))} \quad (9)$$

Where:

$$\emptyset_s = \left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (10)$$

Where: N_b is number of baffles and $(N_b + 1)$ is number of times the shell fluid pass the tube bundle; μ_w is the dynamic viscosity at average wall temperature ($T_{w(average)}$) (K) [15].

$$T_{w(average)} = \frac{T_{b,h} + T_{b,c}}{2} \quad (11)$$

Where $T_{b,h}$ and $T_{b,c}$ is the hot and cold fluid balance temperature respectively

The number of baffles N_b can be calculated as [23,27]

$$N_b = \frac{L}{B} - 1 \quad (12)$$

Where L is length of tubes in (m), and the baffle space B is recommended between 0.4 - 0.6 of the shell diameter, the range can be increased more than 0.6, however it is not prefer to be less than 0.4 of shell diameter [15].

The tube side Reynolds number can be calculation by the expression [15,24,25]:

$$Re_t = \frac{\rho_t . u_m . d_i}{\mu_t} \tag{13}$$

Where:

$$u_m = \frac{\dot{m}_t}{\rho_t . A_i} \tag{14}$$

$$A_i = \left(\frac{\pi d_i^2}{4}\right) \times N_t \tag{15}$$

Where:

Re_t: Reynolds number in tube side
ρ_t: Fluid density in tube side (kg/m³)
u_m: Fluid velocity in tube side (m/s)
d_i: Tube inside diameter (m)
μ_t: Fluid dynamic viscosity (kg/m².s)
$\dot{m}_t$: Fluid mass flow rate inside tube (kg/s)
A_i: Inside tube cross sectional area (m²)
N_t: Number of tubes

According to [15], for laminar flow inside circular tubes, the pressure drop can be calculated by using the relationship between Reynolds number (Re) and training friction factor f, independent of the surface roughness.

$$f = \frac{16}{Re} \tag{16}$$

For both flows laminar and turbulent, the pressure drop can be calculated by:

$$f = \frac{\Delta P_t}{\left[4\left(\frac{L}{d_i}\right) . \left(\frac{\rho u_m^2}{2}\right)\right]} \tag{17}$$

III. RESULTS AND DISCUSSIONS

A. Experimental test

In design the forth exchangers, it should be considered the coolant water nozzle size which is extremely depends on change in fluid properties. By using the equation (1) and (2), the nozzle size of coolant water for each heat exchangers is 12.5 mm.

In the beginning the forth heat exchangers are connected separately with vapor source. The vapor entered the heat exchanger with volume flow rate of 4 liter/hour at 96 °C as average inlet temperature. The heat exchanger convert the vapor to the distilled water by heat transfer process from high temperature vapor to low temperature of coolant water which is flow inside the shell at 130 liter/hour volume flow rate.

Digi-Sense 91000-00 Type K thermocouple thermometer is used to measurement the temperature in four point of H.E (inlet and outlet vapor and coolant water) by using four thermocouple wires. Type K Ten-Channals Switch-Box is used as a connection between thermometer device and thermocuple wires as shown in Fig 2: (d), (e), and (f). Data collected from experimantl test are listed in Table 1 to 4. the experimantel test proved that the water productivity for each heat exchangers is 3.8 liter/hour.

B. Matheematical results

The data collection from experimatel test used to achieved the hydraulic calculation. By using the Equations (3) and (12) which is represent the Reynolds number equation in shell and tube sides, the results show that the flow in two sides of each heat exchangers is lamener as shown in Table 5. By considering the heat exchangers phisical dimension and data collection from experimental test with using equations (7) and (16), the hydrulic calculation was achived as shown in Table 1 to 4.

In Table 1 to 4, the symbol T refer to the temperature and the subscript *h, c, i, and o* are mean hot, cold, inlet, and outlet respectively. Table 5 presented the comparison between four H.Es with maximum presser drop in tube side.

Fig. 3 and fig. 4 show the comparing between four heat exchangers with respect to pressure drope ΔP in both sides shell and tube.

Although, increase in pressur drope in heat exchanger as aresult of midification tubes, the pressure drop will be negelected because of its low value.

Fig.5 show the effect of change physical dimention on weight of heat exchangers (A, B, C, D)

Table. 1: Data collection with physical design and hydraulic calculation results from heat exchanger (A)

Time	$T_{h,i}$	$T_{h,o}$	$T_{c,i}$	$T_{c,o}$	$\dot{V}_h$	$\dot{V}_c$	d_o	d_i	L	D_s	D_{ns}	N_t	N_b	B	Δp_s	ΔP_t
min	°C	°C	°C	°C	L/h	L/h	mm	mm	m	mm	mm			m	Pa	Pa
0	101	45.8	31.5	32.3	4	130	31	25	0.4	100	12.5	2	4	0.08	1.91486	0.00883
10	100.5	45.5	31.5	32.2	4	130	31	25	0.4	100	12.5	2	4	0.08	1.91458	0.00883
20	99.5	43.4	31.5	32.3	4	130	31	25	0.4	100	12.5	2	4	0.08	1.91486	0.00936
30	99.4	42.3	30.8	32.3	4	130	31	25	0.4	100	12.5	2	4	0.08	1.91403	0.00936
40	99.6	42.7	30.8	32.3	4	130	31	25	0.4	100	12.5	2	4	0.08	1.91403	0.00936
50	99.5	38.5	30.8	32.3	4	130	31	25	0.4	100	12.5	2	4	0.08	1.91403	0.00936
60	99.5	37.9	30.8	32.3	4	130	31	25	0.4	100	12.5	2	4	0.08	1.91403	0.00936
70	98.4	37.6	30.8	32.3	4	130	31	25	0.4	100	12.5	2	4	0.08	1.91403	0.00936
80	98.6	39.4	30.8	31.6	4	130	31	25	0.4	100	12.5	2	4	0.08	1.91257	0.00936
90	97.6	38.8	30.8	31.6	4	130	31	25	0.4	100	12.5	2	4	0.08	1.91257	0.01031

Table. 2: Data collection with physical design and hydraulic calculation results from heat exchanger (B)

Time	$T_{h,i}$	$T_{h,o}$	$T_{c,i}$	$T_{c,o}$	$\dot{V}_h$	$\dot{V}_c$	d_o	d_i	L	D_s	D_{ns}	N_t	N_b	B	Δp_s	ΔP_t
min	°C	°C	°C	°C	L/h	L/h	mm	mm	m	mm	mm			m	Pa	Pa
0	100.5	39.2	31.2	34.6	4	130	28	25	0.4	100	12.5	2	4	0.08	2.1319	0.00936
10	99.8	38.2	31.2	27.2	4	130	28	25	0.4	100	12.5	2	4	0.08	2.1361	0.00936
20	99.6	38	31.2	34.3	4	130	28	25	0.4	100	12.5	2	4	0.08	2.1610	0.00971
30	99.6	38.3	30.7	33.2	4	130	28	25	0.4	100	12.5	2	4	0.08	2.1629	0.00973
40	98.4	38.7	30.7	33.2	4	130	28	25	0.4	100	12.5	2	4	0.08	2.1719	0.00978
50	98.3	38.5	30.7	33.2	4	130	28	25	0.4	100	12.5	2	4	0.08	2.1719	0.00978
60	97.7	37.9	30.7	33.2	4	130	28	25	0.4	100	12.5	2	4	0.08	2.1719	0.00987
70	97.3	37.7	30.7	33.4	4	130	28	25	0.4	100	12.5	2	4	0.08	2.1707	0.00987
80	96.5	37.4	30.7	33.4	4	130	28	25	0.4	100	12.5	2	4	0.08	2.1707	0.00987
90	95.8	37.2	30.7	33.5	4	130	28	25	0.4	100	12.5	2	4	0.08	2.1707	0.00992

Table. 3: Data collection with physical design and hydraulic calculation results from heat exchanger (C)

Time	$T_{h,i}$	$T_{h,o}$	$T_{c,i}$	$T_{c,o}$	$\dot{V}_h$	$\dot{V}_c$	d_o	d_i	L	D_s	D_{ns}	N_t	N_b	B	Δp_s	ΔP_t
min	°C	°C	°C	°C	L/h	L/h	mm	mm	m	mm	mm			m	Pa	Pa
0	99.8	46.9	31.6	32.2	4	130	31	25	0.3	100	12.5	2	4	0.06	3.1852	0.00662
10	99.7	46.5	31.6	32.2	4	130	31	25	0.3	100	12.5	2	4	0.06	3.2052	0.00662
20	99.4	46	31.6	32.3	4	130	31	25	0.3	100	12.5	2	4	0.06	3.2287	0.00681
30	99	45.3	30.7	31.4	4	130	31	25	0.3	100	12.5	2	4	0.06	3.2281	0.00681
40	98.6	45.1	30.7	31.4	4	130	31	25	0.3	100	12.5	2	4	0.06	3.2416	0.00681
50	97.5	44.5	30.7	31.4	4	130	31	25	0.3	100	12.5	2	4	0.06	3.2416	0.00706
60	97.1	44.3	30.7	31.4	4	130	31	25	0.3	100	12.5	2	4	0.06	3.2416	0.00702
70	96.4	43.7	30.7	31.4	4	130	31	25	0.3	100	12.5	2	4	0.06	3.2416	0.00702
80	96.6	43.2	30.7	31.4	4	130	31	25	0.3	100	12.5	2	4	0.06	3.2416	0.00706
90	95.3	43.4	30.7	31.4	4	130	31	25	0.3	100	12.5	2	4	0.06	3.2416	0.00706

Table. 4: Data collection with physical design and hydraulic calculation results from heat exchanger (D)

Time	$T_{h,i}$	$T_{h,o}$	$T_{c,i}$	$T_{c,o}$	$\dot{V}_h$	$\dot{V}_c$	d_o	d_i	L	D_s	D_{ns}	N_t	N_b	B	Δp_s	ΔP_t
min	°C	°C	°C	°C	L/h	L/h	mm	mm	m	mm	mm			m	Pa	Pa
0	99.7	40.6	31.2	33.5	4	130	28	25	0.3	100	12.5	2	4	0.06	3.5894	0.007026
10	100	40.8	31.2	33.3	4	130	28	25	0.3	100	12.5	2	4	0.06	3.6119	0.007026
20	99.4	40.1	31.2	33.1	4	130	28	25	0.3	100	12.5	2	4	0.06	3.6384	0.007113
30	99.1	40.3	31.1	32.2	4	130	28	25	0.3	100	12.5	2	4	0.06	3.6425	0.007131
40	98.6	39.7	30..8	32.2	4	130	28	25	0.3	100	12.5	2	4	0.06	3.6577	0.007165
50	98.2	39.5	30.6	32.2	4	130	28	25	0.3	100	12.5	2	4	0.06	3.6577	0.007270
60	97.5	39.6	30.6	32.2	4	130	28	25	0.3	100	12.5	2	4	0.06	3.6577	0.007322
70	96.8	38.9	30.6	32.1	4	130	28	25	0.3	100	12.5	2	4	0.06	3.6587	0.007391
80	96.3	38.7	30.6	32.1	4	130	28	25	0.3	100	12.5	2	4	0.06	3.6587	0.007409
90	95.1	38.4	30.6	32.1	4	130	28	25	0.3	100	12.5	2	4	0.06	3.6587	0.007444

(a) (b) (c)

Fig. 1: (a) , (b), and (c) fabrication stage of Heat exchanger

(d) (e) (f)

Fig. 2: (d) Digi-Sense 91000-00 Type K thermocouple thermometer (e) Type K Ten-Channals Switch-Box (f) Type K thermocouple wires

TABLE 5: COMPARING BETWEEN HEAT EXCHANGERS (A, B, C, D) WITH RESPECT TO PRESSUR DROP AND WEIGHT

Parameters	H.E (A)	H.E (B)	H.E (C)	H.E (D)
Shell diameter (mm)	100	100	100	100
Number of tubes	2	2	2	2
Parameters	H.E (A)	H.E (B)	H.E (C)	H.E (D)
Length of tubes (m)	0.4	0.4	0.3	0.3
ID of tubes (mm)	25	25	25	25
OD of tubes (mm)	31	28	31	28
H.E weight (kg)	7.5	6.75	6.25	5.6
Vapor flow rate (L/h)	4	4	4	4
Coolant water flow rate (L/h)	130	130	130	130
Water productivity (L/h)	3.8	3.8	3.8	3.8
Reynolds number (shell)	894.29	818.77	1181.1	1066.7
Reynolds number (tube)	62.3	64.8	68.15	64.7
Flow type (shell side)	Laminar	Laminar	Laminar	Laminar
Flow type (shell side)	Laminar	Laminar	Laminar	Laminar
ΔP in shell side (Pa)	1.9148	2.1707	3.2416	3.65871
ΔP in tube side (Pa)	0.010319	0.009925	0.007061	0.007444

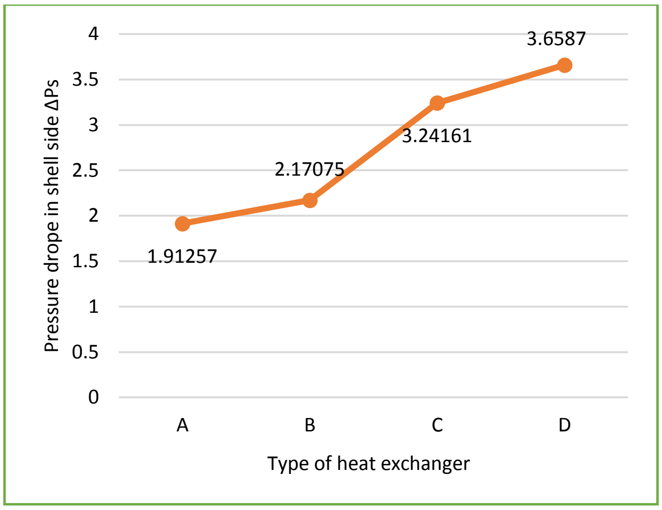

Fig.3 Effect of physical dimention on shell side pressure drop in heat exchangers (A, B, C, D)

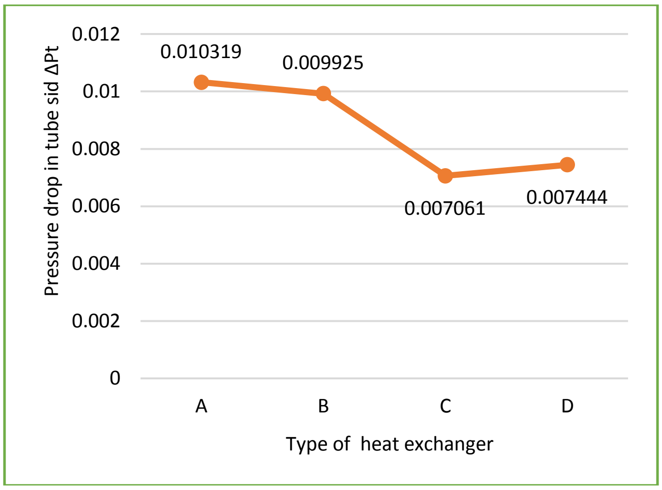

Fig.4 Effect of change physical dimention on tube side pressure drop in heat exchangers (A, B, C, D)

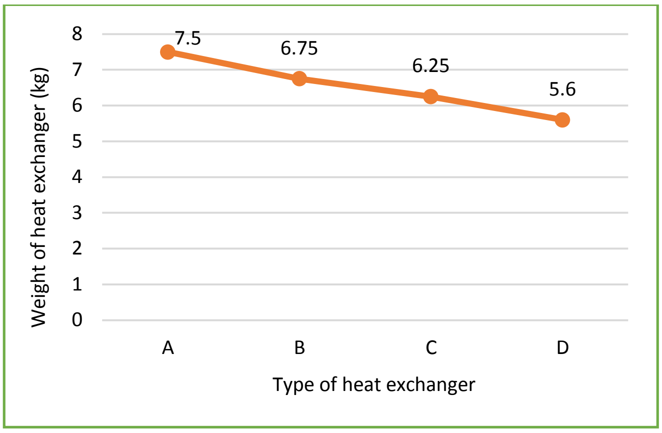

Fig.5 Effect of change physical dimention on weight of heat Exchangers (A, B, C, D)

IV. CONCLUSION

Firstly, the device is able to produce distilled water with no cost of any energy source. However, the maintenance cost still have to be determined. By studying the effect of changing length, outside tube diameter, or both modification between four heat exchangers (A, B, C, and D), the experimental and mathematical results show that:

1) The heat exchanger D is the optimum design which has 100 mm shell diameter and two tubes with 25 mm, 28 mm, and 0.3 m as inside, outside, and length of tube respectively.
2) Reducing thickness or length of tube will lead to reduce the heat exchanger weight and fabrication cost.
3) Reducing thickness of tube will increase pressure drop (ΔP) in both shell and tube sides.
4) While reducing length of tubes will lead to reduction of pressure drop in tube side and increasing ΔP in shell side with same number of baffles.
5) The effect of reducing length of tube on reduction ΔP in tube side and increasing ΔP in shell side is more than effect of thickness reductions.

Finally, the experimental test show that the four heat exchangers (A, B, C, and D) produce the same amount of distilled water as 22.8 liter/day and can be reduce length and thickness of tube to a certain extent in order to reduce the weight for portable heat exchanger.

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