Shell and Tube Heat Exchanger Design: Utilization of Wasted Energy in Air Conditioning Systems

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Abstract—**This research aims to design a simple Shell and Tube (one pass) heat exchanger for applications in recovering wasted heat from air conditioning systems in the hospitality industry. The Tubular Exchanger Manufacturers Association (TEMA) standard is used as a reference in the design of heat exchangers in order to obtain the dimensional specifications of the device. In addition, mathematical calculations based on the technical properties of the working fluid are carried out with the help of the Microsoft Excel application to obtain values from other parameters in the design. The results showed that the designed heat exchanger met the standards in terms of device effectiveness. The heat exchanger has 152 tubes with an effective value of more than 90% with an impurity factor of 0.014. A high tool effectiveness value indicates a good heat exchanger performance. The results of this planning and design are expected to be a reference in designing an effective heat exchanger with high reliability.**

Keywords— Effectiveness, heat exchanger, heat recovery, Shell and Tube, TEMA standar.

I. INTRODUCTION

The rapid growth of the industrial world (especially in the tourism sector) has resulted in an increase in the need for energy. Hotels are one of the supporting parts of this sector and are the largest energy users, especially in terms of the use of electrical energy. Most of the electricity consumption is spent on lighting and air conditioning systems. Without energy efficiency efforts, the energy crisis will become a real threat to the sustainability of the tourism industry. Many researchers try to mitigate this threat by reducing energy losses at each stage of the process. Several alternative approaches have been proposed to achieve this target, including using conversion, conservation, and energy recovery systems [1]. Specifically for the energy recovery approach in the hotel industry, one of the potentials that can be utilized is wasted heat energy from the air conditioning system which can be used as additional energy to heat water. In this case, the heat exchanger (HE) plays a very crucial role.

The most fundamental problem in the process of making HE for specific applications on certain systems is the problem of technical dimensions. Determination of flow patterns, selection of material types and construction types, suitability of heat transfer criteria and pressure drop throughout the process, are the main challenges in HE design apart from the physical dimensions of the tool [2-6]. In addition, several basic parameters can be used as a reference in HE design calculations, including: low flow rate, temperature at the inlet (and at least one at the outlet), and heat transfer rate [7].

Efforts to optimize the HE designs process have been proposed for a long time and reported in many literatures. However, this HE designs problem is still a topic that is widely discussed. The proposed designs generally differ and depend heavily on selecting the objective function, quantity, and type of measurement parameter, to the numerical method used in the calculation [7]. In several literatures, there have been many reports of different HE designs for different uses. Nandiyanto et al., Hidayah & Nandiyanto, and Aprilia, et al. proposed an HE design for the production process of chemicals such as Titanium Dioxide particles, Carbon, nano-Zeolite and Fe3O4 nanoparticles [8-11]. Meanwhile, Milcheva et al. and Erdogan et al. proposed an HE design for an Organic Rankine Cycle (ORC) system [12-13]. On another occasion, Luo et al., Nandiyanto et al., and Masoumpour et al. successively proposed HE designs for automotive thermoelectric generator systems, laboratory scale water baths, and recovery systems at fluid mass flow rates [14-16].

According to the results of the several reports mentioned above, each HE designs proposal has different specifications and performance. This presents an exciting challenge for researchers in terms of the design and construction of HE systems. Based on this, this study focuses on the proposed Shell and Tube type HE designs used in the heat recovery process of the air conditioning system in hotels. The heat that is removed is later used to heat water for bathing purposes. The hope is that by optimizing this energy, the hotel industry will be able to take advantage without reducing the quality of service.

II. METHODS
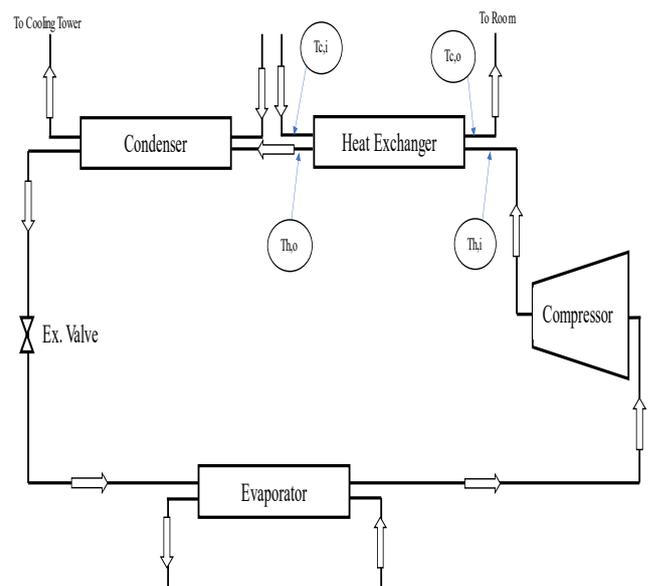
Fig. 1. Schematic heat exchanger

TABLE 1. ASSUMPTION FOR FLUID PROPERTIES WORKING ON HEAT EXCHANGER

Parameters	Shell Side	Tube Side
	Hot Fluid (T_h)	Cold Fluid (T_c)
Inlet temperature, Ti (°C)	88	27
Outlet temperature, To (°C)	28	50
Fluid velocity, ν (m/s)	0.17	0.1
Density, ρ (kg/m³)	980.204	993.185
Thermal conductivity, k (W/m.°C)	0.0977	0.556
Heat specific, Cp (kJ/kg.°C)	1.02332	4.178
Dynamic viscosity, μ (kg/m.s)	0.4869 x 10⁻³	0.536 x 10⁻³

In this work, the chiller machine at the XYZ hotel was used as an object in the design of the Shell and Tube-type HE system. Some of the main components in the chiller system include: evaporator, compressor, condenser, and expansion valve. The designed HE system is placed between the compressor and the condenser. Fig. 1 shows a system schematic of the designed HE placement in the chiller system.

Initial data in heat energy (Q, kW) discharged from the condenser in the chiller system is used as a reference in HE design. Several physical dimensions of the HE (eg. tube diameter, tube thickness, shell thickness) are adjusted to the existing conditions in the condenser, including: inner tube diameter ($D_{T,i}$ = 0.028 m); tube thickness (T_T = 0.001 m) and shell thickness (T_S = 0.01 m). The tube material is made of copper (thermal conductivity, k = 386 W/m.°C), and the shell is made of carbon steel material. The hot fluid used in the HE system is R-134A refrigerant, while the cold fluid is water sourced from the local Regional Drinking Water Company (PDAM). Table 1 shows several assumptions related to the characteristics of each working fluid used in the designed HE system.

The HE system design process (in this work), especially data related to specifications, refers to the Standard Tubular Exchanger Manufacturers Association (TEMA). Manual calculations related to thermal analysis using the help of the Microsoft Excel application based on Equations (1-25). The details of the heat exchange parameters calculated in the HE system design are shown in Table 2.

TABLE 2. HEAT EXCHANGER PARAMETER CALCULATION

Parameters	Equations	Eq.	Ref.
Basic Parameters:			
Energy transferred (Q)	$$Q = C_h(T_{h,i} - T_{h,o}) = C_c(T_{c,o} - T_{c,i})$$ where: Q = energy transferred (W) C = heat capacity (kJ/s.°C) $T_{h,i}$ = inlet temperature (hot fluid) (°C) $T_{h,o}$ = outlet temperature (hot fluid) (°C) $T_{c,i}$ = inlet temperature (cold fluid) (°C) $T_{c,o}$ = outlet temperature (cold fluid) (°C)	(1)	[6]
Logarithmic mean temperature differenced (LMTD)	$$LMTD = \frac{(T_{h,i} - T_{h,o}) - (T_{h,o} - T_{c,i})}{ln\frac{(T_{h,i} - T_{c,o})}{(T_{h,o} - T_{c,i})}}$$ where: LMTD = logarithmic mean temperature differenced (°C)	(2)	[17]
Correction factor (F)	$$R = \frac{(T_{h,i} - T_{h,o})}{(T_{c,o} - T_{c,i})}$$	(3)	[8]
	$$P = \frac{(T_{c,o} - T_{c,i})}{(T_{h,i} - T_{c,i})}$$	(4)	
	$$F = \frac{\sqrt{R^2 + 1}\ ln(\frac{1 - P}{1 - PR})}{(R - 1)\ ln(\frac{2 - P(R + 1 - \sqrt{R^2 + 1}}{2 - P(R + 1 + \sqrt{R^2 + 1}}}$$ where: R, P, and F = correction factor	(5)	
Heat transfer field area (A)	$$A = \frac{Q}{U_o\ x\ LMTD}$$ where: A = heat transfer area (m²)	(6)	[8]

	U_o = overall heat transfer coefficient (W/m².°C)		
Overall heat transfer coefficient (U_o)	$$U_o = \left[\frac{1}{h_a} x \frac{D_{T,o}}{D_{T,i}} + \frac{D_{T,o} \, ln \frac{D_{T,o}}{D_{T,i}}}{2k} + \frac{1}{h_r} \right]^{-1}$$ where: h_a = heat transfer coefficient in tube (W/m².°C) h_r = heat transfer coefficient in shell (W/m².°C)	(7)	[17]
Tube:			
Surface area of total heat transfer in tube (A_T)	$$A_T = N_T \frac{A'_T}{n}$$ where: A_T = surface area of total heat transfer in tube (m²) A'_T = flow area in tube (m²) N_T = number of tube n = number of passes	(8)	[8]
Reynold number in tube ($Re_{,T}$)	$$Re_{,T} = \frac{\rho_a x v_a x D_{T,i}}{\mu_a}$$ where: ρ_a = density of water in tube (kg/m³) v_a = the velocity of water in tube (m/s) $D_{T,i}$ = inner tube diameter (m) μ_a = the dynamic viscosity of water (kg/m.s)	(9)	[18]
Prandtl number in tube ($Pr_{,T}$)	$$Pr_{,T} = \left(\frac{C_p x \mu_a}{k} \right)^{1/2}$$ where: C_p = specific heat of the fluid in tube (kJ/kg.°C) k = thermal conductivity of the tube material (W/m.°C)	(10)	[8]
Nusselt number in tube ($Nu_{,T}$)	$$N_{u,T} = 0,023 \; x \; Re_{,T}{}^{0,6} \; x \; Pr_{,T}{}^{0,33}$$	(11)	[8]
Heat transfer coefficient in tube (h_a)	$$h_a = \frac{N_u \; x \; k}{D_{T,i}}$$ where: k = thermal conductivity of the fluid in tube (kJ/kg.°C)	(12)	[18]
Number of tubes (N_T)	$$N_T = \frac{A_T}{\pi \; x \; D_{T,i} \; x \; l}$$ where: l = tube length (m)	(13)	[8]
Shell:			
Shell flow area (A_S)	$$A_S = \frac{D_S \; x \; C \; x \; D_b}{P_t}$$	(14)	
	$$D_b = D_{T,o} \left(\frac{N_T}{k} \right)^{1/n}$$	(15)	
	$$P_t = 1,25 \; x \; D_{T,o}$$ where: A_s = shell flow area (m²) D_s = shell diameter (m) C = clearance (P_t - $D_{T,o}$) D_b = a shell bundle P_t = tube pitch $D_{T,o}$ = tube outside diameter (m)	(16)	[8]
Equivalent diameter (D_e)	$$D_e = \frac{4 \left(\frac{P_t}{2} x \, 0,87 \, P_t - \frac{1}{2} \pi \frac{D_{T,o}}{4} \right)}{\frac{1}{2} \pi D_{T,o}}$$	(17)	[8]

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	where: $\pi = 3.14$		
Reynold number in shell (Re,s)	$$R_{e,S} = \frac{\rho_R \; x \; v_R \; x \; D_{S,i}}{\mu_R}$$ where: ρ_R = density of refrigerant in shell (kg/m³) v_R = the velocity of refrigerant in shell (m/s) $D_{S,i}$ = inner shell diameter (m) μ_R = the dynamic viscosity of refrigerant (kg/m.s)	(18)	[18]
Prandtl number in shell (Pr,s)	$$Pr_S = \left(\frac{C_p x \mu_R}{k}\right)^{1/2}$$ where: C_p = specific heat of the fluid in shell (kJ/kg.°C) k = thermal conductivity of the shell material (W/m.°C)	(19)	[8]
Nusselt number in shell (Nu,s)	$$N_{u,S} = 0{,}023 \; x \; Re_S{}^{0,6} \; x \; Pr_S{}^{0,33}$$	(20)	[8]
Heat transfer coefficient in shell (h$_r$)	$$h_r = \frac{N_u \; x \; k}{D_e}$$ where: k = thermal conductivity of the fluid in shell (kJ/kg.°C)	(21)	[8]
Shell and Tube:			
Actual overall heat transfer coefficient (U$_{act}$)	$$U_{act} = \frac{1}{\frac{1}{h_a} + \frac{\Delta r}{k} + \frac{1}{h_r}}$$ where: Uact = aktual overall heat transfer coefficient (W/m².°C) Δr = wall thicness (m)	(22)	[8]
Heat Rate:			
Hot fluid rate (C$_h$)	$$C_h = m_h \; x \; C_{P,h}$$ where: C_h = hot fluid rate (W/°C) m_h = mass flow rate of hot fluid (kg/s) $C_{p,h}$ = specific heat capacity of hot fluid (J/kg.°C)	(21)	[8]
Cold fluid rate (C$_c$)	$$C_c = m_c \; x \; C_{P,c}$$ where: C_c = cold fluid rate (W/°C) m_c = mass flow rate of cold fluid (kg/s) $C_{p,c}$ = specific heat capacity of cold fluid (J/kg.°C)	(22)	[8]
Maximum heat transfer (Q$_{max}$)	$$Q_{max} = C_{min}(T_{h,i} - T_{c,i})$$ where: C_{min} = minimum heat capacity rate (W/°C)	(23)	[8]
Effectiveness:			
Heat exchanger effectiveness (ε)	$$\varepsilon = \frac{Q_{act}}{Q_{max}} \; x \; 100\%$$	(24)	[8]
Fouling factor (Rf)	$$Rf = \frac{U_o - U_{act}}{U_o \; x \; U_{act}}$$	(25)	[8]

III. Results and Discussion

Several process assumptions are applied in the HE design mathematical calculations for the heat recovery process from the air conditioning system, including i) HE, which is designed to be a Shell and Tube one-pass process type; ii) the working fluid used is refrigerant R-134A – water; iii) counter flow system; iv) the working fluid on the hot side is refrigerant and on the cold side is water; v) no heat loss during the process; vi) the orientation of the HE is horizontal; vii) the heat rate (Q) obtained from the measurement of wasted heat from the condenser in the air conditioning system which is used as a reference for calculations is 944 kW; and viii) the evaluation is carried out based on TEMA standards.

TABLE 3. DIMENSIONAL SPECIFICATIONS OF THE HEAT EXCHANGER APPARATUS BASED ON THE TEMA STANDARD [9]

Parameters	Specifications
Conductivity materials (W/m.°C)	206
Tube diameter (m)	0.016
Wall thicness (m)	0.0012
Tube length (m)	1.83
Tube arrangements	Triangular
Pitch tube (m)	0.0064
Tube-side passes	2 pass
Tube characteristic angle (°)	30
Shell diameter (m)	0.15
Baffle cut	20 %
Baffle spacing (m)	0.0675

Mathematical calculations for the design of HE devices are intended to theoretically obtain technical performance data from HE, including heat rate (Q), LMTD, heat transfer field area (A), overall heat transfer coefficient (UA), number of tubes (NT), and HE effectiveness. (ε) and the fouling factor (Rf) which refers to the TEMA standard as shown in Table 3.

Based on the assumptions in Table 1 and Table 3, the detailed calculation results of the designed HE performance evaluation (calculated using the equations in Table 2) are shown in Table 4. The values of design parameters such as heart rate (Q)), LMTD, heat transfer field area (A), overall heat transfer coefficient (U_A), number of tubes (N_T), HE effectiveness (ε) and fouling factor (Rf) respectively produced 944 kW; 23.77 °C; 18.378 m^2; 2160.97 W/m^2.°C; 152; 98.36%; and 0.14 °C.m^2/W. The proposed HE design has a high effectiveness value (>90%), in which, this is in line with the proposed HE design reported by Aprilia et al. [11]. The value of HE effectiveness depends on the magnitude of the difference in fluid temperature between the inlet and outlet [9], which means that the effectiveness of HE and the magnitude of the temperature difference are directly proportional.

TABLE 4. PERFORMANCE PARAMETERS OF HEAT EXCHANGERS DESIGNED BASED ON MATEHEMATICAL CALCULATIONS

No	Parameters	Results
1	Heat rate (Q)	944 kW
2	Hot fluid rate (C_h)	15.733 kW/°C
3	Cold fluid rate (Cc)	37.76 kW/°C
4	LMTD	23.77 °C
5	Overall heat transfer coefficient (U_A)	2160.97 W/m^2.°C
6	Heat transfer field area (A)	18.832 m^2
7	Number of tube (N_T)	152
8	Total heat transfer surface area in tube (A_T)	13.832 m^2
9	Reynold number in tube (Re,$_T$)	5095.632
10	Prandtl number in tube (Pr,$_T$)	4.028
11	Nusselt number in tube (Nu,$_T$)	6.107
12	Convection heat transfer coeficient in tube (h$_i$)	123.449 W/m^2.°C
13	Total heat transfer surface area in shell (A_S)	0.008 m^2
14	Reynold number in shell (Re,$_S$)	164273.25
15	Prandtl number in shell (Pr,$_S$)	0.071
16	Nusselt number in shell (Nu,$_S$)	12.941
17	Convection heat transfer coeficient in shell (h$_o$)	158.09 W/m^2.°C
18	Overall actual heat transfer coefficient (U_{act})	69.444 W/m^2.°C
19	Heat exchanger effectiveness (ε)	98.36 %
20	Fouling factor (Rf)	0.014 °C.m^2/W

IV. CONCLUSION

Based on the standard TEMA, the designed HE Shell and Tube meet the specific requirements in terms of tool effectiveness (>90%). In general, the results of this design can be used as a reference for HE design which is intended for the heat recovery process of air conditioning systems as additional energy for heating water.

ACKNOWLEDGMENT

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