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Shell and Tube Heat Exchanger Design Using Silica Nanofluid from Lemongrass Leaves

Baariq Fauzaan Andang Paryawan[1*], Asep Bayu Dani Nandiyanto[2]
Kimia, Fakultas Pendidikan Matematika dan Ilmu Pengetahuan Alam, Universitas Pendidikan Indonesia, Bandung, Indonesia
Corresponding author: baaryfauzaan@upi.edu

Abstract
One of the frequently used pieces of equipment in most industrial applications is heat exchanger. The design of a heat exchanger is very effective to reduce total production costs, compared to buying a ready-made exchanger. The purpose of this research is to design a shell and tube type heat exchanger (two passes) using silica nanofluid that produce from bimass lemongrass for nanosilica raw material. TEMA standard is used to obtain the dimensional specifications of the device. Then, these parameters are calculated manually using the Microsoft Excel application to evaluate the performance of the designed heat exchanger. The results of the study indicate that the designed heat exchanger has met the standard. The heat exchanger has 115 tubes with effectiveness around 75%. Although some parameters still do not meet the standards, so further research is needed to meet the permitted standards. A high value of tool effectiveness indicates that the heat exchanger has good performance. In future, we hope this planning and design can be used as a reference in the design of a heat exchanger to be more economical, effective, and has high reliability in the production activity.

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Key Words
Heat Exchanger, Shell and Tube , Effectivness, silica, nanofluid.

Introduction

Heat exchangers are used for transfering thermal energy between two or more fluids, or solid particulates and a fluid, at different temperature and in thermal contact. The essential principle of a heat exchanger is that it transfers the heat without transferring the fluid that carries the heat. In heat exchangers, there are no external thermal energy and work interactions. The heat transfer occurs mainly due to conduction and convection [1]. The modern human lifestyle is heavily dependent on energy and the transfer of energy. Heat exchangers are among the equipment typically utilized in the majority of industrial applications. [15].

Industrial processes frequently use heat exchangers to either heat or cool the working fluids. They drastically cut production costs by avoiding the need for an external heat source. High thermal conductivity heat exchangers are needed to keep up with the growing demands of industry. The goal of this work is to improve thermal energy transfer in a concentric tube heat exchanger by using nanofluid rather than a traditional fluid like water [2]. Until now, the heat exchanger system can be applied in life, especially in industrial sectors such as oil, gas, chemical industries [3,4], ventilation, air conditioning, battery cooling systems [5,6], and in nuclear, electrical power generation plant [7]. Previously, many papers reported the heat exchanger system conducted a design on the earth–air heat exchanger conducted an experimental investigation of the heat pump performance and ground temperature of a piled foundation heat exchanger system for a residential building [8].

Based on his research results, the overall change in coefficient of performance has not been significant from the beginning to the end of the heating season based upon the same running conditions. Conducted a dynamic modeling and control of plate heat exchanger. Results show that the performance of the fuzzy logic controller produces transient responses with less settling time and less oscillatory behavior compared to that under the conventional controller [10].

Heat exchange devices are essential components in complex engineering systems related to energy generation and energy transformation in industrial scenes. Modeling of shell and tube heat exchanger, for design and performance evaluation, is now an established technique in industrial fields [11]. The purpose of this study is to focus on the design of tube and shell typed heat exchanger using nanosilica fluida.

The current study's goal was to use silica nanofluid to increase the heat transfer rate in (two pass) pipe heat exchanger. Lemongrass leaves were used to prepare the silica nanoparticles by the use of the green synthesis method [12]. When nano fluid was employed, the output temperatures of the hot and cold fluids were compared to those obtained when using distilled water as the cool fluid [2]. For the assumption of the production process on an industrial scale, the heater on the laboratory scale is replaced by a heat exchanger.

Metode Penelitian

1. Synthesis of silica nanoparticles

Lemongrass leaves dried at 200℃ and powdered and filtered using a 34-mesh sieve. Lemongrass was leaching in 5M H_3PO_4 at 80 ℃, or 110 ℃. After being leached with warm distilled water, the treated lemon grass was left to dry overnight at 105°C in an oven. At 600 °C, dried lemon grass was calcined till it turned white ash [12].

2. Preparation of silica nanofluid

A two-step procedure was used to create the nanofluid. The primary fluid, distilled water, was used to disseminate the produced silica nanoparticles. Making nanofluids requires more than just adding nanoscale particles to the host fluid; it also requires making sure that the nanoparticles and fluid have a long-lasting dispersion without any chemical imbalance. High shear blending equipment, such as a sonicator, is used to combine nanopowders with base fluids. Sonication must be used often in order to minimize the formation of particle clusters [16].

Preparation a silica nanofluid with a volume concentration of 0.05%, 1.325 g of manufactured silica nanoparticles were added to the base fluid. The main beverage that was consumed was distilled water. First, the digital weighing equipment was used to measure the silica pieces that were nanosized. They were then mixed with the basic liquid in a glass beaker and sonicated for three to four hours using a Probe sonicator. The result of sonication was a stable nanofluid [2].

2. Method analysis of heat exchanger

When gathering information regarding the model and size of the equipment, we consult the Tubular Exchanger Manufacturers Association Standard (TEMA). The thermal analysis then takes the shape of an overall calculation heat transfer coefficient (U), LMTD method, heat transfer (Q), and pressure drop is also manually computed using basic Microsoft Excel after the equipment dimensions are designed in accordance with the standard. Table 1 presents the basic design calculations with operating circumstances to estimate the apparatus heat exchanger's performance.

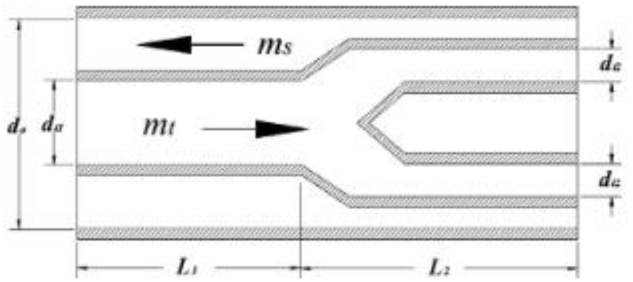

Figure 1. Constructal theory-based heat exchanger with a three-shaped structure [17].

Table 1. Solid properties [2]

Solid Properties	Density ($\rho\ kg/m^3$)	Thermal Conductivity (K) (W/m K)	Specific Heat (Cp) (J/kg K)
Cooper	8980	387,5	380
Iron	7850	344,8	460,54

Table 2. Fluid properties [2]

Fluid Properties	Density ($\rho\ kg/m^3$)	Thermal Conductivity (K) (W/m K)	Specific Heat (Cp) (J/kg K)
Silica nanofluid	1014	0,63	4105,29
Water	1000	0,59	4179,6

Table 3. Heat exchanger parameter calculation.

Section	Parameter	Equation	Eq
Basic parameters	The energy transferred (Q)	$Q_{in} = Q_{out}$ $m_c \times Cp_c \times \Delta T_c = m_h \times Cp_h \times \Delta T_h$ Where, Q : the energy transferred (Wt) m : the mass flow rate of the fluid (Kg/s) Cp : the specific heat ΔT : the fluid temperature difference (°C).	(1)
	Logarithmic mean temperature differenced (LMTD)	$LMTD = \dfrac{(T_{hi} - Tc_i) - (T_{ho} - Tc_o)}{ln\dfrac{(T_{hi} - Tc_i)}{(T_{ho} - Tc_o)}}$ Where, T_{hi} : temperature of the hot fluid inlet (°C) T_{ho} : temperature of the hot fluid outlet (°C) Tc_i : temperature of the cold fluid inlet (°C) Tc_o: temperature of the cold fluid outlet (°C)	(2)
	Correction factor	$R = \dfrac{T_{hi} - T_{ho}}{T_{co} - T_{ci}}$	(3)
		$S = \dfrac{T_{co} - T_{ci}}{T_{hi} - T_{ci}}$	(4)
		$F = \dfrac{\sqrt{R^2+1}\ \ln[\dfrac{1-P}{1-PR}]}{(R-1)\ln\left(\dfrac{2-P(R+1-\sqrt{R^2+1}}{2-P(R+1+\sqrt{R^2+1}}\right)}$	(5)

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		$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$$ $$\Delta T_m = F_t\,\Delta T_{lm}$$	(6)
	Heat Transfer Field Area (A)	$$A = \frac{Q}{U \times LTMD}$$ Where, Q : the energy transferred (W) U : the overall heat transfer coefficient *LMTD* : the logarithmic mean temperature difference.	(7)
	Number of Tubes (Nt)	$$Nt = \frac{A}{\pi \times D_o \times l}$$ Where, *N* : the number of tubes *A* : the area of the heat transfer area (m^2), π : 3.14 D_o : tube diameter (m) l : length diameter (m).	(8)
Tube	Surface Area of Total Heat Transfer in Tube (a_t)	$$a_t = N_t \frac{a'_t}{n}$$ Where, a_t : the total heat transfer surface area in the tube (m^2) N_t : the number of tubes a'_t : the flow area in the tube (m^2) *n* : the number of passes.	(9)
	Mass Flow Rate of Water in Tube (Gt)	$$Gt = \frac{m_h}{a_t}$$ Where, Gt : the mass flow in the tube (kg/m^2s) m_h : the mass flow rate of the hot fluid (Kg/s) a_t : the flow area tube (m^2)	(10)
	Reynold number (Re,t)	$$Re_t = \frac{di_t \times Gt}{\mu}$$ Where, Re_t : the Reynolds number in tube di_t : the inner tube diameter (m), Gt : the mass flow of water in the tube (m^2) μ : the dynamic viscosity (Kg/ms).	(11)
	Prandtl Number (Pr,t)	$$Pr = \left(\frac{C_p \times \mu}{K}\right)^{\frac{1}{2}}$$	(12)

		Where, *Pr* : Prandtl number *Cp* : the specific heat of the fluid in the tube μ : the dynamic viscosity of the fluid in the tube (Kg/ms) *K* : the thermal conductivity of the tube material (W/m°C)	
	Nusselt number (Nu,t)	$Nu = 0.023 \times Re_t^{0.6} \times Pr^{0.33}$	(13)
	Inside coefficient (h_i)	$hi = \frac{Nu \times K}{d_i, t}$ Where, hi : the convection heat transfer coefficient in the tube (W/m^2 °C) *K* : the thermal conductivity of the material (W/m °C) d_i, t : the inner tube diameter (m).	(14)
Shell	Shell flow area (A_s)	$A_s = \frac{d_s \times C \times B}{P_t}$	(15)
		$D_b = d_o \left(\frac{N_t}{k_1}\right)^{\frac{1}{n_1}}$ Where, d_s : shell diameter (m) *C* : clearance (P_t-d_o) *B* : a shell bundle P_t : tube pitch (1.25× d_o) (m)	(16)
	Mass Flow Rate of Water in Shell (Gs)	$Gs = \frac{m_c}{A_s}$ m_c : the mass flow rate of the cold fluid (Kg/s) A_s : the shell flow area (m^2).	(17)
	Equivalent diameter (d_e)	$d_e = \frac{4\left(\frac{Pt}{2} \times 0.87\ Pt - \frac{1}{2}\pi \frac{d_{o,t}}{4}\right)}{\frac{1}{2}\pi d_{o,t}}$ Where, P_t : tube pitch (1.25× d_o) (m) π : 3.14 $d_{o,t}$: tube outside diameter (m)	(18)
	Reynold number (Re,s)	$Re_s = \frac{di_t \times Gt}{\mu}$ Re_s : Reynold number di_s : inner tube diameter (m) Gs : the mass flow of water in the shell (Kg/m^{2}s) μ : the dynamic viscosity (Kg/ms)	(19)
	Prandtl Number (Pr,s)	$Pr = \left(\frac{C_p \times \mu}{K}\right)^{\frac{1}{2}}$ Pr_s : Prandtl number *Cp* : specific heat capacity (kJ/kg°C) μ : dynamic fluid viscosity (Kg/ms) *K* : thermal conductivity (W/m°C)	(20)

Shell and Tube	Nusselt number (Nu,s)	$Nu_s = 0.023 \times {Re_s}^{0.6} \times Pr^{0.33}$ Re_s : Reynold number Pr : Prandtl number	(21)
	Convection Heat Transfer Coefficient (ho)	$ho = \frac{Nu \times K}{d_e}$ ho : convection heat transfer coefficient (W/m^2°C) K : thermal conductivity (W/m°C) d_e : equivalent diameter (m).	(22)
	Actual Overall Heat Transfer Coefficient (U_{act})	$U_{act} = \frac{1}{\frac{1}{h_i} + \frac{\Delta r}{k} + \frac{1}{h_o}}$ Where, hi : inside heat transfer coefficient (W/m^2°C) ho : outside heat transfer coefficient (W/m^2°C), Δr : wall thickness (m) K : thermal conductivity(W/m°C)	(23)
Heat rate	Hot Fluid Rate (C_h)	$C_h = m_h . Cp_h$ Where, C_h : hot fluid rate (W/°C) Cp_h : specific heat capacity (J/Kg°C) m_h : mass flow rate of hot fluid (Kg/s).	(24)
	Cold Fluid Rate (C_C)	$C_C = m_c . Cp_c$ C_c : cold fluid rate (W/°C), Cp_h : specific heat capacity (J/Kg°C), m_c : mass flow rate of cold fluid (Kg/s). $Q_{max} = C_{min}(T_{h,i} - T_{c,i})$ Q_{max} : maximum heat transfer (W) C_{min} : minimum heat capacity rate (W/°C) $T_{h,i}$: temperature of the hot fluid inlet (°C) $T_{c,i}$: temperature of the cold fluid inlet (°C).	(25)
Effectiveness	Heat Exchanger Effectiveness (ε)	$\varepsilon = \frac{Qact}{Qmax} \times 100\%$ Where, Q_{act} : actual energy transferred (W) Q_{max} : maximum heat transfer (W)	(26)
	Number of Transfer Unit (NTU)	$NTU = \frac{U \times A}{C_{min}}$ Where, U : overall heat transfer coefficient (W/m^2°C) A : heat transfer area (m^2) C_{min} : minimum heat capacity rate (W/°C).	(27)
	Fouling factor (Rf)	$Rf = \frac{U_a - U_{act}}{U_a \times U_{act}}$ Where Rf : fouling factor U_a : overall heat transfer coefficient (W/m^2°C) U_{act} : actual overall heat transfer coefficient (W/m^2°C)	(28)

Result and Discussion

The assumptions are used to design and estimate the performance on heat exchanger, include:

1. The heat exchanger design is a shell and tube (two-pass) type
2. The material for the design of the heat exchanger is cooper - iron
3. The fluid used of the heat exchanger is a nanosilica fluid-water fluid system.
4. The hot fluid is assumed to be on the shell side and the cold fluid is assumed to be on the tube side.
5. The specifications for the type of the heat exchanger respectively are AEW
6. Heat losses to or from the surronding are negligible
7. Overall coefficient (U) for hot and cold fluids water is 800 W/m^2°C
8. The baffle type is single segmental with orientation perpendicular
9. Hot fluid is located in tube side and cold fluid is located in shell side

Geometric specifications and dimensions of the heat axchanger apparatus based on the TEMA standard are listed in table 4.

Table 4. Dimensional specifications of the heat exchanger apparatus based on the TEMA standard.

Parameters	Specification
Conductivity Material (W/m°C)	114,85
Tube Outer Diameter (m)	0.018
Tube Inner Diameter (m)	0.016
Wall Thickness (m)	0.0019
Tube Length (m)	4.0
Tube arrangements	Triangular
Pitch Tube (m)	0.025
Tube-side passes	2 pass
Tube Characteristic Angle (°)	30
Shell Outer Diameter (m)	0.152
Shell Inner Diameter (m)	0.136
Baffle Cut	25%

Table 5. Specifications of hot and cold fluids

Parameters	The specification in Tube Side	The specification in Shell Side
Inlet Temperature ($T_{h,in}$; °C)	90	-
Outlet Temperature ($T_{h,out}$; °C)	60	-
Inlet Temperature ($T_{c,in}$; °C)	-	30
Outlet Temperature ($T_{c,out}$; °C)	-	80
Fluid Flow Rate (kg/s)	0,133 kg/s	0,027 kg/s
Density (kg/m^3)	8980	7850
Viscosity (N.s/m^2)	0,000315	0,000798
Thermal Conductivity (W/m.K)	387.5	344.8
Heat Spesific (J/kg.K)	380	460.54
Operating Pressure (bar)	1.013	1.013

The data used to model the heat exchanger have been defined as in Tables 4 and 5 which respectively show the apparatus dimension specifications and fluid specifications. Based on the analysis of the calculation of the assumption data, the designed heat exchanger follows the specifications in Table 5. The dimension specifications of the apparatus refer to the standards of The Tubular Exchanger Manufacturers Association (TEMA). Based on the assumptions in Tables

4 and 5, Table 6 shows the results of calculations to evaluate the performance of the designed apparatus heat exchanger.

Table 6. Performance parameters of heat exchangers designed based on calculations

No	Parameter	Results
1	Initial Heat Transfer Rate (*Q*)	378450 W
2	Logarithmic Mean Temperature Difference (*LMTD*)	18,2°C
3	Assumed Overall Fluid Heat Coefficient of Water and Nanosilica (U_a)	800 W/m°C
4	R	0,6
5	S	0,83
6	Ft	1,85
7	ΔTm	33,66°C
8	Area of Heat Transfer (*A*)	25,98 m^2
9	Number of Tube (*Nt*)	115
10	Total Heat Transfer Surface Area in Tube (a_t)	0,023 m^2
11	Mass Flow Rate of Water Fluid in Tube (Gt)	130,50 m/s
12	Reynold Number in Tube (*Re, t)*	6628,72
13	Prandtl Number in Tube (*Pr, t*)	0,98
14	Convection Heat Transfer Coefficient in the Tube (h_i)	188,31 W/ m^2·K
15	Bundle Shell (*Db*)	3,76 m
16	Total Heat Transfer Surface Area in Shell (A_s)	0,178 m^2
17	Mass Flow Rate of Water Fluid in Shell (*Gs*)	11,23 m/s
18	Equivalent Diameter (*De*)	0,98 m
19	Reynold Number in Shell (*Re, s*)	225,24
20	Prandtl Number in Shell (*Pr, t*)	2,73
21	Nusselt Number in Shell (*Nu, t*)	0,74
22	Convection Heat Transfer Coefficient in Shell (h_o)	81,42 W/ m^2.C
23	Overall Heat Transfer Coefficient Actual (U_{act})	48,29 W/ m^2.C
24	HE Effectiveness (*ε*)	75%
25	Number of Transfer Unit (*NTU*)	2,47
26	Fouling Resistance	0,019 ^{o}C. m^2/W

Stated differently, Table 5 shows that the following values were obtained for the planned heat exchanger: 18,2°C, 25,98 m2, 115 pcs, 48.29 W/m2.K, and 75%, respectively, for the LMTD, surface area, number of tubes, and overall heat exchanger transfer coefficient. Heat transmission from the heated fluid to the cold fluid is facilitated by a high overall heat exchanger transfer coefficient value [11]. According to the findings, the intended heat exchanger's efficacy is greater than 70%. The heat exchanger's heat transfer properties are solely represented by the efectiveness (ε), which is unaffected by pressure losses [13]. If there is a significant temperature differential between the input and the output, the value is high. Laminar flow is seen in the fluid flow on the tube side.

In the meantime, turbulent flow is visible in the fluid flow on the shell side. The Reynolds value (Re) determines the flow of a fluid. The turbulent flow is followed by the fluid flow if Re is less than 2300. Conversely, the fluid flow follows laminar flow if Re > 2300 [14]. Further research is required to satisfy the permissible requirements, even though the effectiveness value is rather good and some metrics still do not reach the norms. The TEMA standard states that the water fluid's fouling resistance value is 0.019 °C.m^2/W.

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Conclusion

The heat exchanger has been successfully developed using the Shell and Tube (two-pass) type with the AEW type furnished with as many as 115 pieces of tube, based on the TEMA standard-based design. With laminar flow on the shell side and turbulent flow on the tube side, the device generates a heat transfer rate of 378450 watts. The heat exchanger design's efficacy is greater than 75%. As a result, the heat exchanger's performance is good.

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