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Design of Heat Exchanger for The Production of SnO_2 Nanoparticle

Yohanes Ivan Benaya Parlindungan Nainggolan, Asep Bayu Dani Nandiyanto

Department of Chemical Education, Universitas Pendidikan Indonesia, Indonesia
nandiyanto@upi.edu

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Abstract

This study's objective is to examine the design of a heat exchanger for the production of SnO_2 nanoparticles by calculating the shell and tube heat exchanger's dimensions. In order for the design to be well-directed, several steps must be taken, including determining the heat exchanger's dimensions and material requirements using standards, computing the main shell and tube components, and computing the heat exchanger's performance. Microsoft Excel is used for heat exchanger data processing. One shell and two tubes were used in the design of the heat exchanger. The results of the designed HE have a shell diameter of 0.84 m, a tube length of 7.315 m, and an inside diameter of 0.021 m. The flow type of HE is turbulent, with a heat transfer efficiency of 48.05 percent and a fouling factor of 0.0015195 $m^2 \cdot K/W$, the device generates a heat transfer rate of 958198.91 W. This study demonstrates that the heat exchanger has a successful, high-performing design. This design can be used as a model for creating a heat exchanger that is more cost-efficient, efficient, and reliable.

Keywords: Heat exchanger; Shell and tube; Effectiveness; Performance

1. Introduction

The process of changing temperature and phase takes place by utilizing heat transfer from a high to low temperature fluid [1]. A heat transfer device that moves heat energy from one medium to another is called a heat exchanger. Two fluids with different temperatures will be separated on the hot side or the cold side by a separating medium in order to achieve the ideal thermal in the heat transfer process. An advantage of a heat exchanger is that it is affordable and has high thermal efficiency [2]. The thin plates that make up the heat exchanger are connected by a frame that extends parallel to the plates [3].

The heat exchanger device is essential for industrial process operations, and many different industries use this device, including the natural gas industry, chemical and petrochemical factories, oil refineries, refrigeration [1], [4], power plants, nuclear reactors, industrial processes, and as fuel [4]–[7]. There are many different types of heat exchanger, including wire on tube [8], crossflow type plate heat exchanger [9], type plate heat exchanger [10], type ground heat exchanger [11], and plate-fin heat exchanger using a new type of vortex generators [12]. The shell and tube heat exchanger is one of the most common heat exchangers [13].

A precise and cost-effective design plan is required for the manufacture of the heat exchanger because it is an apparatus that is crucial to the process activities industry and whose maximum support can influence a process to become more optimal. As a result, it can be efficiently operated and maximized in accordance with the design that have been planned.

SnO_2 nanoparticles are one of the materials that are produced on an industrial scale. SnO_2 is a highly stable oxide semiconductor with a wide n-type band gap of 3.6 eV, and it has good optical and electrical properties in addition to being low cost and non-toxic. It also has a high carrier density, a high level of surface defects, and a high level of oxygen vacancy [14], [15]. SnO2 has been used in a wide range of applications, such as solar cells [16], [17], catalytic support materials [18], photocatalysts [19], solid-state chemical sensors [20], and transparent electrodes [21].

The objective of this research is to design a heat exchanger for the production of SnO_2 nanoparticles and to determine the fundamental measurements of the devices, such as the heat transfer surface area (A), which depends on other factors, including thermal load (Q), overall heat transfer coefficient (U), and logarithmic mean temperature difference (ΔTlm).

2. Method

2.1. Manufacturing of SnO_2 Nanoparticle

Synthesis of SnO_2 nanoparticles in conventional batch reactor approach [22] was carried out by adding 1 mmol (0.351 g) of $SnCl_4.5H_2O$ and dissolving it in 10 mL of water in a beaker. To create a 1:1 basic mixture, 10 mL of ethanol and 0.8 g of NaOH were dissolved in 10 mL of water. After that, 6 mL of this basic mixture was continuously stirred into the $SnCl_4.5H_2O$ solution. When the pH was acidic, a white, cloudy suspension slowly started to form; however, as the pH increased to around 11-12, the suspension disappeared. The resulting mixture was then refluxed for 5 mins at 90 ^{0}C. The contents were then centrifuged (at a speed of 6000 rpm for 10 minutes), washed with water twice and three times, and dried for 24 hours under vacuum.

2.2. Mathematical Models for Designing Heat Exchanger

In this study, we designed a heat exchanger with a single shell and two tubes passes. The apparatus was created through a number of steps, including calculating the main shell and tube parts and the heat exchanger's performance. The Tubular Exchanger Manufacturers Association (TEMA) standard was the basis for the apparatus' design. Furthermore, performance calculations were done manually. Microsoft Excel is used in the HE design to process the data. Equations 1-29 in **Table 1** provide the fundamental design calculations with operating conditions to estimate the performance of the heat exchange.

Table 1. HE parameter calculation

Section	Parameter	Equation	Eq. no.
Basic Parameter	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 = Energy transferred (W) m = Mass flow rate of the fluid (kg/s) Cp = Specific heat of fluid (J/kg·K) ΔT = Fluid temperature difference (°C)	(1)
	Logarithmic Mean Temperature Difference ($LMTD$)	$LMTD = \dfrac{(T_{hi} - T_{ci}) - (T_{ho} - T_{co})}{\ln \dfrac{(T_{hi} - T_{ci})}{(T_{ho} - T_{co})}}$ Where: $LMTD$ = Logarithmic Mean Temperature Difference (℃) T_{hi} = Temperature of the hot fluid inlet (℃) T_{ho} = Temperature of the hot fluid outlet (℃) T_{ci} = Temperature of the cold fluid inlet (℃)	(2)

Section	Parameter	Equation	Eq. no.
		T_{co} = Temperature of the cold fluid outlet (°C)	
	Correction factor	$R = \frac{T_{hi} - T_{ho}}{T_{co} - T_{ci}}$ $S = \frac{T_{co} - T_{ci}}{T_{hi} - T_{ci}}$ $F = \frac{\sqrt{R^2+1}\ln\left[\frac{1-S}{1-RS}\right]}{(R-1)\ln\left[\frac{2-S(R+1-\sqrt{R^2+1}}{2-S(R+1+\sqrt{R^2+1}}\right]}$	(3) (4) (5)
	Heat transfer field area (A)	$A = N \times l \times a"$ Where: A = Heat transfer area (m^2) N = Number of tubes l = Length of tube (m) $a"$ = Outside surface of tube (m^2)	(6)
	Overall Heat Transfer Coefficient (U_d)	$U_d = \frac{Q}{A \times LMTD}$ Where: U_d = Overall heat transfer coefficient (W/m^2·K) Q = Energy transferred (W) A = Area of the heat transfer (m^2) $LMTD$ = Logarithmic mean temperature difference (K)	(7)
Tube	Surface area of total Heat transfer in tube (a_t)	$a_t = \frac{N_t \times a't}{n}$ Where: a_t = Total heat transfer surface area in tube (m^2) N_t = Number of tubes $a't$ = Flow area in tube (m^2) n = Number of passes	(8)
	Mass flow rate of fluid in tube (G_t)	$G_t = \frac{m_c}{a_t}$ Where: G_t = Mass flow of fluid in tube (kg/m^2·s) m_c = Mass flow rate of cold fluid (kg/s) a_t = Total heat transfer surface area in tube (m^2)	(9)
	Reynold number (Re,t)	$Re_t = \frac{di_t \times Gt}{\mu}$ Where: Re_t = Reynolds number in tube di_t = Inner tube diameter (m) Gt = Mass flow of shell in tube (kg/m^2·s) μ = Dynamic viscosity of cold fluid at mean temperature (N·s/m^2)	(10)
	Prandtl number (Pr,t)	$Pr_t = \left(\frac{C_p \times \mu}{K}\right)^{\frac{1}{2}}$ Where: Pr_t = Prandtl number in tube C_p = Specific heat of fluid in tube (J/kg·K) μ = Dynamic viscosity of fluid at mean temperature (N·s/m^2) K = Thermal conductivity of tube material (W/m·K)	(11)
	Nusselt number (Nu,t)	$Nu_t = 0.023 \times {Re_t}^{0.6} \times {Pr_t}^{0.33}$ Where: Nu_t = Nusselt number in tube Re_t = Reynolds number in tube Pr_t = Prandtl number in tube	(12)

Section	Parameter	Equation	Eq. no.
	Tube heat transfer coefficient (*hio*)	$$hi = jH \times \frac{k}{D} \times \left(\frac{C_p \times \mu}{k}\right)^{0.33} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$ $$hio = hi \times \frac{D_i}{D_o}$$ Where: hio = Tube heat transfer coefficient (W/m²·K) D_i = Tube inside diameter (m) D_o = Tube outside diameter (m) jH = Tube heat transfer factor k = Thermal conductivity of cold fluid at mean temperature (W/m·K) D = Tube inside diameter (m) C_p = Specific heat of fluid in tube (J/kg·K) μ = Dynamic viscosity of cold fluid at mean temperature (N·s/m²)	(13) (14)
Shell	Surface area of total Heat transfer in shell (a_s)	$$a_s = \frac{d_s \times C \times B}{P_T}$$ Where: a_s = Total heat transfer surface area in shell (m²) d_s = Shell inner diameter (m) C = Clearance (m) B = Baffle spacing (m) P_T = Tube pitch (m)	(15)
	Mass flow rate of fluid in shell (G_s)	$$G_s = \frac{m_h}{a_s}$$ Where: G_s = Mass flow of fluid in shell (kg/m²·s) m_h = Mass flow rate of hot fluid (kg/s) a_s = Total heat transfer surface area in shell (m²)	(16)
	Reynold number (*Re,s*)	$$Re_s = \frac{d_e \times Gs}{\mu}$$ Where: Re_s = Reynolds number in shell d_e = Equivalent diameter (m) Gs = Mass flow of fluid in shell (kg/m²·s) μ = Dynamic viscosity of hot fluid at mean temperature (N·s/m²)	(17)
	Prandtl number (*Pr,s*)	$$Pr_s = \left(\frac{C_p \times \mu}{K}\right)^{\frac{1}{2}}$$ Where: Pr_s = Prandtl number in shell C_p = Specific heat of fluid in shell (J/kg·K) μ = Dynamic viscosity of fluid at mean temperature (N·s/m²) K = Thermal conductivity of tube material (W/m·K)	(18)
	Nusselt number (*Nu,s*)	$$Nu_s = 0.023 \times {Re_s}^{0.6} \times {\mathrm{Pr}_s}^{0.33}$$ Where: Nu_s = Nusselt number in shell Re_s = Reynolds number in shell Pr_s = Prandtl number in shell	(19)
	Shell heat transfer coefficient (*ho*)	$$ho = jH \times \frac{k}{D_e} \times \left(\frac{C_p \times \mu}{k}\right)^{0.33} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$ Where: ho = Shell heat transfer coefficient (W/m²·K) jH = Shell heat transfer factor	(20)

Section	Parameter	Equation	Eq. no.
		k = Thermal conductivity of hot fluid at mean temperature (W/m·K) D_e = Equivalent diameter (m) C_p = Specific heat of the fluid in tube (J/kg·K) μ = Dynamic viscosity of hot fluid at mean temperature (N·s/m²)	
	Equivalent diameter (D_e)	$D_e = \dfrac{4 \times \left(\frac{1}{2} P_T \times 0.86 P_T - \dfrac{\frac{1}{2}\pi \times {D_o}^2}{4}\right)}{\frac{1}{2}\pi \times D_o}$ Where: D_e = Equivalent diameter (m) P_T = Tube pitch (m) D_o = Tube outside diameter (m)	(21)
	Shell diameter (D_s)	$D_s = 0.63 \left(\dfrac{\sqrt{\frac{CL}{CTP}} \times A \times (PR)^2 \times D_o}{l}\right)^{1/2}$ Where: D_s = Shell diameter (m) A = Area of the heat transfer area (m²) P, R = Correction factor D_o = Tube diameter (m) l = Tube length (m) For CTP value (one tube pass = 0,93; two tube passes = 0,90; and three tube passes = 0,85) and CL value (90° and 45° = 1,00; and 30° and 60° = 0,87)	(22)
Heat rate	Hot Fluid Rate (C_h)	$C_h = m_h \times C_{ph}$ Where: C_h = Hot fluid rate (W/K) m_h = Mass flow rate of hot fluid (kg/s) C_{ph} = Specific heat capacity of hot fluid (J/kg·K)	(23)
	Cold Fluid Rate (C_c)	$C_c = m_c \times C_{pc}$ Where: C_c = Cold fluid rate (W/K) m_c = Mass flow rate of cold fluid (kg/s) C_{pc} = Specific heat capacity of cold fluid (J/kg·K)	(24)
Effectiveness	Maximum heat transfer (Q_{max})	$Q_{max} = C_{min}(T_{h,i} - T_{c,i})$ Where: Q_{max} = Maximum heat transfer (W) C_{min} = Minimum heat capacity rate (W/K) $T_{h,i}$ = Inlet temperature of the hot fluid (°C) $T_{c,i}$ = Inlet temperature of the cold fluid (°C)	(25)
	Clean Overall Heat Transfer Coefficient (U_c)	$U_c = \dfrac{hio \times ho}{hio + ho}$ Where: U_c = Clean overall heat transfer coefficient (W/m²·K) $hio = hio$ = Tube heat transfer coefficient (W/m²·K) ho = Shell heat transfer coefficient (W/m²·K)	(26)
	Heat Exchanger Effectiveness (ε)	$\varepsilon = \dfrac{Q_{act}}{Q_{max}} \times 100\ \%$ Where: Q_{act} = Actual energy transferred (W) Q_{max} = Maximum heat transfer (W)	(27)

Section	Parameter	Equation	Eq. no.
	Number of Transfer Unit (*NTU*)	$NTU = \frac{U_d \times A}{C_{min}}$ Where: NTU = Number of Transfer Unit U_d = Overall heat transfer coefficient (W/m²·K) A = Heat transfer area (m²) C_{min} = Minimum heat capacity rate (W/K)	(28)
	Fouling factor (R_f)	$R_f = \frac{U_c - U_d}{U_c \times U_d}$ Where: R_f = Fouling factor (m²·K/W) U_c = Clean overall heat transfer coefficient (W/m²·K) U_d = Overall heat transfer coefficient (W/m²·K)	(29)

3. Results and Discussion

To design a heat exchanger, we must first understand the materials that will be used to construct the heat exchanger. Knowing the material and the amount of cold material that needs to be heated with a stream of water as a heating medium allows us to measure dimensions. The type of flow and the amount of water that needed to be heated were then determined. Additionally, designing the testing mechanism and the pipe layout design are completed.

Steel pipe is considered to be the material used in this design for the shell and tube type heat exchanger apparatus. N-butanol was used as the hot fluid in the shell side of the heat exchanger and toluene as the cold fluid in the tube side. The physical characteristics of hot and cold fluids are shown in Table 2.

Table 2. Physical and thermal properties of the fluid

	Shell Side (Hot Fluid)	Tube Side (Cold fluid)
	n-Butanol	Toluene
Inlet Temperature (T_{in})	100 °C	25 °C
Outlet Temperature (T_{out})	40 °C	90 °C
Fluid Flow Rate	7.05 kg/s	8.33 kg/s
Operating Pressure	1 atm	1 atm
Dynamic Viscosity at T mean (μ)	0.0011 N·s/m²	0.0004 N·s/m²
Thermal Conductivity at T mean (k)	0.1448 W/m·K	0.1258 W/m·K
Density at T mean (ρ)	766.6231 kg/m³	832.9671 kg/m³
Heat Capacity at T mean (c_p)	2268.9454 J/kg·K	1772.5410 J/kg·K

The Tubular Exchanger Manufacturers Association (TEMA) reportedly defined more than 280 distinct types of shell-and-tube heat exchangers in the past. The simplest shell-and-tube heat exchanger has a single pass through the shell and a single pass through the tubes [23]. In this HE designs, one pass shell and two passes tube are used. The tube pitch, PT is the shortest center-to-center distance between adjacent tubes. The dimension of the tube pitch used are 1-in. OD on 1¼-in. triangular pitch.

Based on the calculations, the results are 958198.91 W for the initial heat transfer rate (Q), area of heat transfer (A) is 429.62 m² and Logarithmic Mean Temperature Difference (*LMTD*) is -10.88 °C.

Reynolds numbers greater than 2300 indicate turbulent flow characteristics in the shell and tube. In industrial processes like heating and cooling, turbulent flow has a variety of applications. The majority of industrial heat exchangers use turbulent flow, which has a higher heat convection coefficient than laminar flow and, as a result, a better heat transfer rate [24].

The shell and tube type heat exchanger has been successfully designed with a heat transfer effectiveness of 48.05 percent based on the calculation results. HE effectiveness is the actual heat transfer rate divided by the maximum possible heat transfer rate [25]. If there is a significant temperature difference between the input and output, the resulting effectiveness value—which calculates the amount of heat carried—will be high. Another way to look at it is that the efficiency of the heat exchanger is directly correlated with the temperature difference [26].

The number and spacing of baffles specified in the heat exchanger's specification are among many other elements that influence how effectively it performs. The effectiveness of the heat exchanger will be increased by a small percentage of baffle cut as well as by a close baffle distance [27]. **Table 3** presents both the heat exchanger design specifications and the calculation-based performance results of the heat exchanger based on the assumed specifications.

Table 3. Specification of shell and tube heat exchanger and operating condition for mineral oils and ethylene glycol fluid based on calculation result

Parameter	Value
Initial Heat Transfer Rate (Q)	958198.91 W
Logarithmic Mean Temperature Difference ($LMTD$)	-10.88 °C
Overall Fluid Heat Coefficient (U_d)	323.27 W/m^2·K
Area of Heat Transfer (A)	429.62 m^2
Tube Outside Diameter (D_o)	0.025 m
Tube Inside Diameter (D_i)	0.021 m
Wall Thickness	0.0021 m
Tube Length (l)	7.315 m
Tube Pitch (P_T)	0.032 m
Number of Tube (N_t)	736
Total Heat Transfer Surface Area in Tube (a_t)	0.13 m^2
Mass Flow Rate of Fluid in Tube (G_t)	64.30 kg/m^2·s
Reynold Number in Tube (Re, t)	3203
Prandtl Number in Tube (Pr, t)	7.74
Nusselt Number in Tube (Nu, t)	5.73
Convection Heat Transfer Coefficient in the Tube (hio)	719.52 W/m^2·K
Shell Inside Diameter (d_s)	0.84 m
Clearance (C)	0.0064 m
Baffle Spacing (B)	0.21 m
Equivalent Diameter (D_e)	0.018 m
Total Heat Transfer Surface Area in Shell (a_s)	0.035 m^2
Mass Flow Rate of Water Fluid in Shell (G_s)	200.19 kg/m^2·s
Reynold Number in Shell (Re, t)	3189
Prandtl Number in Shell (Pr, t)	4.19
Nusselt Number in Shell (Nu, t)	4.67
Convection Heat Transfer Coefficient in Shell (ho)	5433.31 W/m^2·K
Clean Overall Heat Transfer Coefficient (U_c)	635.38 W/m^2·K
HE Effectiveness (ε)	48.05 %
Number of Transfer Unit (NTU)	9.4085
Fouling Resistance (R_d)	0.0015195 m^2·K/W
Pressure Drop in Tube	0.00081 atm
Pressure Drop in Shell	0.05628 atm

4. Conclusion

The one Shell and two Tube passes type of heat exchanger has been successfully designed for the production of SnO_2 nanoparticles. The results of the designed HE have a shell diameter of 0.84 m, a tube length of 7.315 m, and a tube inside diameter of 0.021 m. The flow type of HE is turbulent. The heat transfer rate produced is 958198.91 W, with a heat transfer efficiency of 48.05% and a fouling factor of 0.0015195 $m^2 \cdot K/W$.

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