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Heat Exchanger Design for Cerium Oxide Nanoparticles Production

**Anisa Klarasita[1] , Asep Bayu Dani Nandiyanto[2]*, Risti Ragadhita[3]
and Teguh Kurniawan[4]**

[1,2,3]Departemen Kimia, Fakultas Pendidikan Matematika dan Ilmu Pengetahuan Alam. Universitas Pendidikan Indonesia, Bandung, Indonesia.
[4]Departemen Teknik Kimia, Fakultas Teknik. Universitas Sultan Ageng Tirtayasa, Serang, Indonesia

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Abstract

The aim of this study is to design a heat exchanger using water as a cooling fluid and Triethylene glycol as a heating fluid to obtain a product which is cerium oxide nanoparticles. The type of shell and tube on the HE is designed in a simple way but still refers to the existing design rules. The type of shell used is one pass shell and tube with laminar flow. Specifications of HE design among others; 25.4 mm of tube diameter, 21.2 mm of inner tube diameter, 2.1 mm of thickness, 5.4864 m of tube length, 27.78 mm in pitch tube, outer shell diameter is 228.6 mm, and inner shell diameter of the amount 203 mm. The results were obtained in the form of a Q value in 188280 W with an effectiveness value of 92.00%. This design can be used as a reference in designing a heat exchanger to be more effective, economical, has high reliability, and can also be useful as a learning method regarding the design process, working mechanism, and HE performance analysis.

Keywords: Design, Effectiveness, Heat Exchanger, Shell, Tube.

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*Corresponding author.
E-mail address: nandiyanto@upi.edu

1. Introduction

'Design' is defined as the process of determining all the construction dimensions of the exchanger that must be heated to a predetermined limit, both on the shell side and tube side pressure drop. The process of heat exchange between two fluids that are at different temperatures can be separated by a solid wall, the device used to carry out this exchange is called a heat exchanger [1]. Heat exchanger is a series of devices that allow heat transfer between two or more fluids. In the heat transfer process, to achieve an ideal thermal equilibrium, the two fluids having different temperatures will be separated on both the hot and cold sides through a separating medium [2]. Heat exchangers have proven to be important devices for thermal systems in many industrial fields [3]. With the rapid progress in science and technology, the increase in heat transfer is also making the way forward to modern nanotechnology [2].

Heat transfer can be improved by using a variety of methods. The heat transfer coefficient can be increased by improving the coolant properties for a particular heat transfer method [4]. Coolant in the heat transfer process is referred to as a cold fluid, water can act as a cooling fluid in the heat exchanger. Water is used as a cooling fluid because it has high specific heat properties with low viscosity and very low cost so that it brings advantages to the heat transfer application industry [2]. As for the heating fluid used is Triethylene glycol. Triethylene glycol (TEG) is a colorless viscous liquid with a slight odor, and a high boiling point. This liquid is non-flammable, mildly toxic, and considered harmless. TEG is used as a raw material in various industries and is produced with ethylene oxide as raw material [5]. Based on this, TEG was chosen as the heating fluid. Heating or cooling fluids are important for many industrial sectors, including transportation, energy supply and production. The thermal conductivity of these fluids plays an important role in the development of energy-efficient heat transfer equipment [6]. To obtain nanoparticles products, it is necessary to pay attention to parameters such as the volume flow rate of both heating and cooling fluids and fluid temperature [3]. One of the nanoparticle products that can be produced is cerium oxide nanoparticles.

Cerium belongs to a class of lanthanide metals in the periodic table [7]. In oxide form, cerium has a fluorite structure. The nanoscale form, cerium oxide nanoparticles retain the fluorite structure with oxygen deficiencies [8]. In recent years, nanocrystalline cerium oxide (CeO_2) particles have been broadly examined, due to their potential utilize in many applications [9], such as gas sensors [10], UV absorbents and filters [11], catalysts in fuel cell technology [12], and photocatalytic oxidation of water [13]. Several studies on production of cerium oxide nanoparticle have been done, such as produced CeO_2 particles from cerium chloride [14]. The aim of this study is to design a heat exchanger using water as a cooling fluid and Triethylene glycol as a heating fluid to obtain a product which is cerium oxide nanoparticles.

2. Materials and methods

2.1. Manufacturing of Cerium Oxide Nanoparticles

Cerium oxide nanocrystallites were synthesized by co-precipitation process at relatively low temperature, using cerium (III) nitrate as starting material in aqueous solution with a pH in the range of 8-9. The crystallization temperature of cerium oxide powders is estimated to be around 273 K. When calcined at temperatures from 473 to 1273 K, face-centered cubic crystallization is produced [9].

2.2. Mathematical models for designing heat exchangers

In the HE processes, in order for the ideal displacement to occur, it is necessary to first calculate the various parameters that have been set. Data processing on HE is done using Microsoft Excel to make it more efficient. **Table 1** shows the parameters for calculating the heat exchanger to estimate the performance of the heat exchanger.

Table 1. Calculation parameters of heat exchanger [15]

No	Section	Parameter	Equation	Eq
1	Basic parameters	The energy transferred (Q)	$Q_{in} = Q_{out}$ $m_c \times Cp_c \times \Delta T_c = m_h \times Cp_h \times \Delta T_h$ Q = the energy transferred (Wt) m = the mass flow rate of the fluid (Kg/s) Cp = the specific heat ΔT = the fluid temperature difference (℃)	(1)
		Logarithmic Mean Temperature Differenced (LMTD)	$LMTD = \frac{(T_{hi} - T_{Ci}) - (T_{ho} - T_{Co})}{ln \frac{(T_{hi} - T_{Ci})}{(T_{ho} - T_{Co})}}$ T_{hi} = the temperature of the hot fluid inlet T_{ho} = the temperature of the hot fluid outlet T_{Ci} = the temperature of the cold fluid inlet T_{Co} = the temperature of the cold fluid outlet All temperature units are in ℃	(2)
		Correction factor	$R = \frac{(T_{hi} - T_{ho})}{(T_{Co} - T_{Ci})}$ $P = \frac{(T_{Co} - T_{Ci})}{(T_{hi} - T_{Ci})}$	(3) (4) (5)

			$F=\frac{\sqrt{R^2+1}\ln[\frac{1-P}{1-PR}]}{(R-1)\ln(\frac{2-P(R+1-\sqrt{R^2+1}}{2-P(R+1+\sqrt{R^2+1}})}$	
		Heat Transfer Field Area (A)	$A=\frac{Q}{U\times LMTD}$ Q = the energy transferred (W) U = the overall heat transfer coefficient $LMTD$ = the logarithmic mean temperature difference	(6)
		Number of Tubes (N)	$N=\frac{A}{\pi\times D_o\times l}$ N = the number of tubes A = the area of the heat transfer area (m^2) π = the value of 3.14 D_o = the tube diameter (m) l = the tube length (m)	(7)
		Shell Diameter	$D_s=0.63\,(\frac{\sqrt{\frac{CL}{CTP}\times((A\times(PR)^2\times D_o}}{l})^{\frac{1}{2}}$ D_s = the shell diameter (m) A = the area of the heat transfer area (m^2) PR = the correction factor D_o = the tube diameter (m) CTP = the constant (one tube pass = 0.93; two tube pass = 0.90; and three tube pass = 0.85) CL = the constant (CL at 45 and 90° = 1.00; CL at 30° and 60° = 0.87)	(8)
2	Tube	Surface Area of Total Heat Transfer in Tube (a_t)	$a_t=N_t\frac{a'_t}{n}$ 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 of Water in Tube (Gt)	$$Gt = \frac{m_h}{a_t}$$ Gt = the mass flow of water in tube (Kg/m^{2}s) 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)	$$Re_t = \frac{di_t \times Gt}{\mu}$$ 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)	$$Pr = (\frac{C_p \times \mu}{K})^{\frac{1}{2}}$$ Pr = Prandtl Number C_p = 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)	(12)
		Nusselt Number (Nu, t)	$$Nu = 0{,}023 \times {Re_t}^{0{,}6} \times Pr^{0{,}33}$$	(13)
		Inside coefficient (h_i)	$$h_i = \frac{Nu \times K}{d_i, t}$$ h_i = the convection heat transfer coefficient in the tube (W/m°C) K = the thermal conductivity of the material (W/m°C) d_i, t = the inner tube diameter (m^2)	(14)
3	Shell	Shell flow area (A_s)	$$A_s = \frac{d_s \times C \times B}{P_t}$$ $$D_b = d_o(\frac{N_t}{k_1})^{\frac{1}{n_1}}$$	(15) (16)

			d_s = the shell diameter (m) C = clearance ($P_t - d_o$) B = the a shell bundle P_t = the tube pitch (1.25 × d_o) (m)	
		Mass Flow Rate of Water in Shell (G_s)	$G_s = \frac{m_c}{a_s}$ 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)	$d_e = \frac{4 \times (\frac{P_t}{2} \times 0.87\, P_t - \frac{1}{2}\pi \frac{d_{o,t}}{4})}{\frac{1}{2}\pi d_{o,t}}$ P_t = the tube pitch (1.25 × d_o) (m) π = the value of 3.14 $d_{o,t}$ = the tube outside diameter (m)	(18)
		Reynold number (Re, s)	$Re_s = \frac{di_t \times Gt}{\mu}$ Re_s = the Reynold number di_t = the inner tube diameter (m) Gt = 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 = the Prandtl number C_p = the specific heat capacity kJ/kg°C μ = the dynamic fluid viscosity (Kg/ms) K = the thermal conductivity (W/m°C)	(19)
		Nusselt number (Nu, s)	$Nu_s = 0.023 \times Re_s^{0.6} \times Pr^{0.33}$ Nu_s = the Nusselt number	(20)

			Re_s = the Reynold number Pr = the Prandtl number	
		Convection Heat Transfer Coefficient (ho)	$$ho = \frac{Nu \times K}{d_e}$$ ho = the convection heat transfer coefficient (W/m^2.°C) K = the thermal conductivity (W/m°C) d_e = the equivalent diameter (m)	(21)
4	Shell and Tube	Actual Overall Heat Transfer Coefficient (U_{act})	$$U_{act} = \frac{1}{\frac{1}{h_i} + \frac{\Delta r}{K} + \frac{1}{h_o}}$$ h_i = the inside heat transfer coefficient (W/m^2.°C) ho = the outside heat transfer coefficient (W/m^2.°C) Δr = the wall thickness (m) K = the thermal conductivity (W/m°C)	(22)
5	Heat rate	Hot Fluid Rate (C_h)	$$C_h = m_h . Cp_h$$ C_h = the hot fluid rate (W/°C) m_h = the specific heat capacity (J/kg.°C) Cp_h = the mass flow rate of hot fluid (kg/s)	(23)
		Cold Fluid Rate (C_c)	$$C_c = m_c . Cp_c$$ C_c = the cold fluid rate (W/°C) m_c = the specific heat capacity (J/kg.°C) Cp_c = the mass flow rate of cold fluid (kg/s) $$Q_{maks} = C_{min}(T_{h,i} - T_{c,i})$$ Q_{maks} = the maximum heat transfer (W) C_{min} = the minimum heat capacity rate (W/°C) $T_{h,i}$ = the temperature of the hot fluid inlet (°C)	(24)

			$T_{c,i}$ = the temperature of the cold fluid inlet (°C)	
6	Effectiveness	HE Effectiveness (ε)	$\varepsilon = \frac{Q\ act}{Q\ max} \times 100\%$ $Q\ act$ = the actual energy transferred (W) $Q\ max$ = the maximum heat transfer (W)	(25)
		Number of Transfer Unit *(NTU)*	$NTU = \frac{U \times A}{C_{min}}$ U = the overall heat transfer coefficien (W/m^2.°C) A = the heat transfer area (m^2) C_{min} = the minimum heat capacity rate (W/°C)	(26)
		Fouling factor *(Rf)*	$Rf = \frac{U_a - U_{act}}{U_a \times U_{act}}$ Rf = the fouling factor U_a = the overall heat transfer coefficient (W/m^2.°C) U_{act} = the actual overall heat transfer coefficient (W/m^2.°C)	(27)

3. Results and discussion

3.1. Fluid properties and parameters for heat exchanger design

The principle of HE is to equalize the difference between the inlet temperature of the hot fluid (T_h in) and the inlet temperature of the cold fluid (T_c in), with a visible effect on the exit temperature [15]. Heat transfer plays an important role in almost every industry [16]. In this heat exchanger design, to obtain cerium oxide nanoparticles, water is used as the cooling fluid and the heating fluid used is Triethylene glycol. Assumptions regarding the characteristics of the fluid used in the heat exchanger device are shown in **Table 2.**

Table 2. Fluid properties assumption for working on HE [17, 18]

Side	Shell Side	Tube Side
Fluid type	Hot fluid (*Triethylene glycol*), T_h	Cold fluid (water), T_c
Temperature inlet, T_{in} (℃)	43	90
Temperature outlet, T_{out} (℃)	100	75
Fluid flow rate, m (kg/s)	2	3
Density, ρ (kg/m^3)	1100	997
Viscosity (N.s/m^2)	0.049	0.00089
Thermal conductivity (W/m.K)	0.196	0.595
Heat specific (J/Kg.K)	2177.136	4184
Operating pressure (bar)	1.013	1.013

To obtain the value of the HE design specifications, several assumptions were used by developing shell and tube types on the HE, including the outer tube diameter of 25.4 mm, inner tube diameter 21.2 mm, wall thickness 2.1 mm, tube length 5.4864 m, 27.78 mm pitch tube, tube-side passes 1, tube characteristic angle 30℃, outer shell diameter 228.6 mm, inner shell diameter 203 mm, buffle cut 25%, and baffle spacing 220.4 m. The calculation results of the parameters for designing the HE performance are presented in **Table 3.**

The results show that the value for the Initial Heat Transfer Rate (Q) in the shell and tube HE design is 188280 W with a Reynolds Number in Tube value of less than 2300 which indicates the type of flow used is laminar flow. In industrial processes, there are two types of flow namely laminar flow and turbulent flow which are used for heating and cooling. A connected laminar flow heat exchanger may experience non-uniform flow and a reduction in the effective heat exchanger coefficient at the Reynolds number [19]. The number and spacing of baffles in the HE specification affect the performance of the HE, a close baffle distance will increase the effectiveness of the HE and a small percentage of baffle cut will increase the effectiveness of the HE [20].

The HE effectiveness value obtained is 92.00%, where the HE effectiveness is the actual heat transfer rate divided by the maximum possible heat transfer rate [21]. HE effectiveness measures the amount of heat carried, the resulting value will be high if the temperature difference between input and output is large. So it can also be interpreted that the effectiveness of HE is directly proportional to the magnitude of the temperature difference [22]. In addition, the total performance (efficacy) of HE is determined by the thermal conductivity, viscosity, density, and specific heat of the working fluid [23].

Table 3. Parameters design of HE performance based on calculation

No.	Parameter	Hasil
1	Initial Heat Transfer Rate (Q)	188280 W
2	Logarithmic Mean Temperature Difference (LMTD)	31.46℃
3	Area of Heat Transfer (A)	85.11 m^2
4	Number of Tube (Nt)	194.52
5	Total Heat Transfer Surface Area in Tube (A_t)	0.44 m^2
6	Mass Flow Rate of Water Fluid in Tube (Gt)	87.56 Kg/m^2.s
7	Reynold Number in Tube (Re, t)	2084.19
8	Prandtl Number in Tube (Pr, t)	6.26
9	Nusselt Number in Tube (Nu, t)	4.13
10	Convection Heat Transfer Coefficient in the Tube (h_i)	3891.14 W/m^2.C
11	Bundle Shell (Db)	402.35 mm
12	Total Heat Transfer Surface Area in Shell (A_s)	0.02 m^2
13	Mass Flow Rate of Water Fluid in Shell (Gs)	95.68 Kg/m^2.s
14	Equivalent Diameter (De)	25835.15 m
15	Reynold Number in Shell (Re, t)	50446.67
16	Prandtl Number in Shell (Pr, t)	544.28
17	Nusselt Number in Shell (Nu, t)	1940.71
18	Convection Heat Transfer Coefficient in Shell (h_o)	84266.32
19	Overall Heat Transfer Coefficient Actual (U_{act})	7.82
20	HE Effectiveness (ε)	92.00%
21	Number of Transfer Unit (NTU)	14.07

Conclusion

For the Heat Exchanger design, the shell is one pass shell and tube with laminar flow specifications with tube diameter of 25.4 mm, inner tube diameter 21.2 mm, thickness 2.1 mm, tube length 5.4864 m, outer shell diameter 228.6 mm, and inner shell diameter 203. mm obtained a Q value of 188280 W with an effectiveness of 92.00%. This design can be used for learning methods regarding the design process, working mechanism, and HE performance analysis.

Conflict of Interest

This study was conducted without any commercial or financial relationship with any institution.

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