CHEMICAL ENGINEERING TRANSACTIONS

VOL. 52, 2016

A publication of AIDIC

The Italian Association of Chemical Engineering
Online at www.aidic.it/cet

Guest Editors: Petar Sabev Varbanov, Peng-Yen Liew, Jun-Yow Yong, Jiří Jaromír Klemeš, Hon Loong Lam

ISBN 978-88-95608-42-6; **ISSN** 2283-9216

DOI: 10.3303/CET1652194

Impact of Hybrid Heat Transfer Enhancement Techniques in Shell and Tube Heat Exchanger Design

Ehsan Shekarian[a], Mohammad R. Jafari Nasr[b], Amir H. Tarighaleslami[c]*, Timothy G. Walmsley[c], Martin J. Atkins[c], Nadia Sahebjamee[d], Mohammad Alaghehbandan[e]

[a]Department of Chemical Engineering, College of Engineering, Adadshahr Branch, Islamic Azad University, Azadshahr, Iran
[b]Research Institute of Pertoleum Industry (RIPI), Tehran, Iran
[c]Energy Research Centre, School of Engineering, University of Waikato, Private Bag 3105, Hamilton 3240, New Zealand
[d]Department of Chemical Engineering, College of Engineering, Quchan Branch, Islamic Azad University, Quchan, Iran
[e]Applied Research Centre for Agricultural Inputs, Karaj, Iran
aht5@students.waikato.ac

Despite the advantages of shell and tube heat exchangers, one of their major problems is low thermal efficiency. This problem can be improved by using heat transfer enhancement techniques such as adding nanoparticles to the hot or cold fluids, and/or using tube inserts as turbulators on tube side as well as changing baffles to a helical or twisted profile on the shell side. Although all of these techniques increase the thermal efficiency; however, engineers still need a quantitative approach to assess the impact of these technologies on the shell and tube heat exchangers. This study attempts to provide a combination of such techniques to increase the impact of these improvements quantitatively. For this purpose, at first stage the thermal and hydraulic characteristics of pure fluid, Al_2O_3/water nanofluid in a plain tube equipped with and without twisted tape turbulator is evaluated based on a developed rapid design algorithm. Therefore, the impact of using enhanced techniques either in form of individual or in hybrid format and the increase of nanoparticle concentration in base fluid have been studied. The results show that using turbulators individually and in hybrid format with nanofluid can be effected on design parameters of a typical heat exchanger by reducing the required heat transfer area up to 10 %.

1. Introduction

Shell and Tube Heat Exchangers (STHE) are widely used in various industries such as oil, gas, petrochemical and power plants. Despite the increase in uptake of other types of Heat Exchangers (HE), which have better performance, STHEs remain the most widely used type of heat exchanger.

Iterative trial and error procedures are commonly applied in HE design procedures to achieve desirable heat loads subjected to the allowable fluid pressure drops. As a result, to ensure compliance with the allowable pressure drop and achieve a desired duty, HEs are designed larger than actually required. Polley et al. (1991) introduced a HE design algorithm, which could calculate simultaneously desirable heat load with consideration of maximum allowable pressure drops. The results are higher velocities on both the shell and tube sides, higher heat transfer coefficients and the minimum heat transfer area requirement, i.e. minimum capital cost. The method is called the Rapid Design Algorithm (RDA). Polley et al. (1991) introduced simple equations based on the Kern method (1950) for each stream, which correlated pressure drop, heat transfer coefficient of the fluid, and the heat exchanger area together. Combining pressure drop and basic heat exchange relationships led designers to avoid lengthy trial and error procedure. However, using Kern relations increases shell side errors due to inaccurate relationships. Later, Bell-Delaware equations were used to improve flow pattern in shell side and consequently achieved closer results of thermal-hydraulic analysis to experimental

Please cite this article as: Shekarian E., Jafari Nasr M. R., Tarighaleslami A. H., Walmsley T. G., Atkins M. J., Sahebjamee N., Alaghehbandan M., 2016, Impact of hybrid heat transfer enhancement techniques in shell and tube heat exchanger design, Chemical Engineering Transactions, 52, 1159-1164 DOI:10.3303/CET1652194

results (Serna and Jiménez, 2005). In this model, the amounts of leakage near the baffles and by pass flow were considered.
Various techniques have been applied to increase heat transfer rates in HEs and decrease heat and energy losses in process industries. These methods are known as Heat Transfer Enhancement (HTE). HTE can be applied in both shell side and tube side of STHEs. Helical baffles are more common in shell side (Wang et al., 2010). Jafari Nasr and Shafeghat (2008) presented a combination of HTE using helical baffles and RDA. Applying different types of turbulators and/or nanofluids are examples of HTE techniques in tube side. HTE techniques can be applied in both grassroots and retrofit designs to improve heat transfer characteristics of HEs. However, due to high revamp costs in large industries such as oil, gas, petrochemical and power plants, applying HTE techniques are more reliable and profitable in economical point of view.
Using nanofluids as service fluid in HEs is another HTE technique, which recently has been investigated for large industrial plants (Tarighaleslami et al., 2015). Nanofluids are prepared by distributing a nanoparticle through a base fluid, which helps increase its thermal conductivity (Shekarian et al., 2014). Bubbico et al. (2015) showed that different type of nanofluid may provide improved heat transfer efficiency. Elias et al. (2014) investigated the effect of different shape of nanoparticles on overall heat transfer coefficient of a STHE using different baffle angles and nanofluid. Sundar and Sharma (2010) studied the effect of increasing in heat load by using twisted tape inserts in laminar and turbulent flows. They used twisted tape inserts in presence of Al_2O_3/water nanofluid and proposed relationships for friction factor and Nusselt for simultaneous use of twisted tape insert and Al_2O_3/water nanofluid.
The aim of this paper is to develop the required inputs to RDA for three tube-side HTE scenarios for STHEs using the empirical correlations of Sundar and Sharma (2010). The three considered scenarios are: (1) Al_2O_3/water nanofluid, (2) twisted tape turbulators, and (3) a hybrid combination of both techniques.

2. Methodology

RDA method is one of the heat exchanger design methods that eliminates any trial and error calculations due to geometrical changes, leading to faster achievement of optimal design results. In RDA method, calculation of the heat exchanger area is based on the maximum usage of the given allowable pressure drop on both the shell- and tube-sides and solving the set of equations (1) simultaneously (Serna and Jiménez, 2004).

$$
\begin{aligned}
&A = \frac{Q}{F_T \Delta T_{LM}}\left(\frac{1}{h_s} + \frac{1}{h_t} + \sum R\right) \\
&\qquad \Delta P_t = K_T . A . h_t^n \\
&\qquad \Delta P_s = K_S . A . h_s^m
\end{aligned}
\tag{1}
$$

ΔP_s and ΔP_t are the maximum allowable pressure drop for shell and tube sides, K_T, K_S, m and n are dependent to the geometric parameters of the heat exchanger and physical properties of the fluid. K_T and n values are obtained based on semi-empirical equations that have been reported by reliable sources for the Nusselt number, friction factor and some mathematical simplifications. m and K_S are also available in terms of the shell side and based on the Bell-Delaware method.

2.1 The heat exchanger design algorithm and tube and shell sides equations

In each of the scenarios, the same shell-side correlations are applied. The Bell-Delaware equations that are needed as inputs to the RDA (Shenoy, 1995) are:.

$$
\begin{aligned}
&\Delta P_s = (K_{S1} . A + K_{S2}) . h_s^2 \\
&K_{S1} = \frac{\left[(1+0.3N_{tcw}).\rho_s.R_l + 2.f_i.\rho_s.N_{tcc}.\left(\frac{\mu_s}{\mu_{sw}}\right)^{-0.14} R_l.R_b\right]}{(\pi.D_t.N_t.L_{bc}.NS).\left[j_{\mathrm{h}}\frac{k_s.\rho_s}{\mu_s}\left(\frac{Cps.\mu_s}{k_s}\right)^{\frac{1}{3}}\left(\frac{\mu_s}{\mu_{sw}}\right)^{0.14} J_c.J_l.J_b.J_r.J_s\right]^2} \\
&K_{S2} = \frac{\left[2f_i\rho_s N_{tcc}\left(\frac{\mu_s}{\mu_{sw}}\right)^{-0.14}\left(1+\frac{N_{tcw}}{N_{tcc}}\right)R_s R_b - (1+0.3N_{tcw})\rho_s R_l - 4f_i\rho_s N_{tcc}\left(\frac{\mu_s}{\mu_{sw}}\right)^{-0.14} R_l R_b\right]}{\left[j_{\mathrm{h}}\frac{k_s.\rho_s}{\mu_s}\left(\frac{Cps.\mu_s}{k_s}\right)^{\frac{1}{3}}\left(\frac{\mu_s}{\mu_{sw}}\right)^{0.14} J_c.J_l.J_b.J_r.J_s\right]^2}
\end{aligned}
\tag{2}
$$

Scenario 1: No HTE - water flows in plain wall tube
In this case, water is used as the fluid on the tube side. Equations for Nusselt number and friction factor equations (Shenoy, 1995) are:

$$Nu = 0.023Re^{0.8}Pr^{1/3}$$
$$f = 0.184Re^{-0.2} \tag{3}$$

The relationship between pressure drop and heat transfer coefficient, therefore, is:

$$\Delta P_t = K_T.A.h^{3.5}$$
$$K_T = k_1 \frac{\rho_t D_i^2}{4M_t D_t}\left(\frac{1}{k_2}\right)^{3.5}$$
$$k_1 = 0.092\left(\frac{\rho_t}{D_i}\right)\left(\frac{\rho_t D_i}{\mu_t}\right)^{-0.2} \tag{4}$$
$$k_2 = 0.023Pr^{1/3}\left(\frac{k_t}{D_i}\right)\left(\frac{\rho_t D_i}{\mu_t}\right)^{0.8}$$

Scenario 2: Nanofluid HTE - Al_2O_3/water nanofluid in plain wall tubes
In this case, the object is to evaluate the effect of nanofluid on heat exchanger design parameters. Applied Nusselt number and friction factor (Pak and Cho, 1998) are:

$$Nu = 0.021Re^{0.8}Pr^{0.5}$$
$$f = 0.316Re^{-0.2} \tag{5}$$

The following equations express the relationship between pressure drop and heat transfer coefficient:

$$\Delta P_t = K_T.A.h^{3.5}$$
$$K_T = k_1 \frac{\rho_t D_i^2}{4M_t D_t}\left(\frac{1}{k_2}\right)^{3.5}$$
$$k_1 = 0.632\left(\frac{\rho_t}{D_i}\right)\left(\frac{\rho_t D_i}{\mu_t}\right)^{-0.2} \tag{6}$$
$$k_2 = 0.021Pr^{0.5}\left(\frac{k_t}{D_i}\right)\left(\frac{\rho_t D_i}{\mu_t}\right)^{0.8}$$

Scenario 3: Turbulent flow of water in tubes containing twisted tapes
In this case, the effect of turbulator on HE design parameters has been studied. For this scenario, suggested equations by Sundar and Sharma (2010), which are obtained from a regression of their experimental results, are used.

$$Nu = 0.02649Re^{0.8204}Pr^{0.4}(0.001 + \frac{H}{D})^{0.06281}$$
$$f = 2.068Re^{-0.4330}(1 + \frac{H}{D})^{0.004815} \tag{7}$$

This corresponds to the following relationships as RDA inputs:

$$\Delta P_t = K_T.A.h^{3.1290}$$
$$K_T = k_1 \frac{\rho_t D_i^2}{4M_t D_t}\left(\frac{1}{k_2}\right)^{3.1290}$$
$$k_1 = 4.136(1 + \frac{H}{D})^{0.004815}\left(\frac{\rho_t}{D_i}\right)\left(\frac{\rho_t D_i}{\mu_t}\right)^{-0.4330} \tag{8}$$
$$k_2 = 0.02649(0.001 + \frac{H}{D})^{0.06281}Pr^{0.4}\left(\frac{k_t}{D_i}\right)\left(\frac{\rho_t D_i}{\mu_t}\right)^{0.8204}$$

Scenario 4: Hybrid Al_2O_3/water nanofluid in a tube containing a twisted tape turbulators
In this case, the effect of the combination of nanofluid and turbulators is investigated. Again Sundar and Sharma (2010) equations are used.

$$Nu = 0.03666Re^{0.8204}Pr^{0.4}(0.001 + \phi)^{0.04704}(0.001 + \frac{H}{D})^{0.06281} \tag{9}$$

$$f = 2.068Re^{-0.4330}(1+\phi)^{0.01}(1+\frac{H}{D})^{0.004815}$$

The relation between pressure drop and heat transfer coefficient is:

$$\Delta P_t = K_T.A.h^{3.1290}$$

$$K_T = k_1 \frac{\rho_t D_i^2}{4M_t D_t}\left(\frac{1}{k_2}\right)^{3.1290}$$

$$k_1 = 4.136(1+\phi)^{0.01}(1+\frac{H}{D})^{0.004815}\left(\frac{\rho_t}{D_i}\right)\left(\frac{\rho_t D_i}{\mu_t}\right)^{-0.4330} \quad (10)$$

$$k_2 = 0.03666(0.001+\phi)^{0.04704}(0.001+\frac{H}{D})^{0.06281}Pr^{0.4}\left(\frac{k_t}{D_i}\right)\left(\frac{\rho_t D_i}{\mu_t}\right)^{0.8204}$$

3. Illustrative case study

To study the effect of HTE techniques, a case study is investigated for base case and three HTE scenarios. Allowable pressure drop in the tube and shell sides are 42,000 and 7,000 Pa. The HE is entirely made of carbon steel and consists of one shell pass and 6 tube passes. Shell side flow rate is 14.9 kg/s. Thermo-physical properties of the tube side fluids (service fluid, water and nanoparticle) and shell side fluid are presented in Table 1. Using the equations provided by Pak and Cho (1998), the thermo-physical properties of nanofluid can be calculated.

Table 1: Thermo-physical properties of fluids and nanoparticle, and flow stream data.

Fluid / Nanoparticle Direction	ρ (kg/m^3)	C_p (kg °C)	k (W/m °C)	μ (kg/m s)	Fouling Factor	T_{in} (°C)	T_{out} (°C)
Water / Tube	998	4,180	0.60	0.001	0.00015	15	25
Service Fluid / Shell	777	2,684	0.11	0.00023	0.00015	98	65
Al_2O_3	3970	880					

4. Results and discussion

4.1 Validation

To validate the result of RDA method, the result outcomes of this design has been compared with results of Kern and Bell-Delaware methods. Results show that in both RDA with Kern and RDA with Bell-Delaware methods acceptable ranges are achieved. Moreover, in Kern and Bell-Delaware methods, due to iterative trial and error calculations, it is not always possible to reach to the maximum allowable pressure drop, while it is a disadvantages of these methods compared with RDA.

4.2 Effect of nanofluid

Table 2 shows the effect of nanofluid by increasing of nanofluid concentration compared with water. According to the results, using nanofluid leads to a slight reduction in required heat transfer surface area. The reduction in heat transfer area is an advantage of nanofluids, which in some cases makes a reduction in initial investment cost compared to the cases that pure water is used. According to Table 2 it can be concluded that the use of nanoparticles only up to a certain concentration can cause an improvement; higher concentration has no benefit in the design parameters (most notably is the exchanger surface). The reason of this phenomenon is that the thermo-physical properties of nanofluid are different from the thermos-physical properties of the base fluid. Convective heat transfer coefficient of nanofluids is a function of the thermal conductivity and heat capacity of the base fluid and nanoparticles, flow patterns, Reynolds and Prandtl numbers, the volume fraction of nanoparticles in nanofluid, nanoparticle size and shape.

To increase the heat transfer coefficient in turbulent flow, both the conductive heat transfer coefficient and heat capacity should increase while the viscosity needs to be reduced. However, as it can be seen in Table 2, by increasing the concentration of nanoparticles in the base fluid, thermal conductivity and viscosity increase, and heat capacity decreases. As a result, there is a trade-off between these parameters. Hence it can be concluded that for concentration up to 0.1 % of nanoparticles causes the thermal conductivity to dominate heat capacity and viscosity. This results in the convective heat transfer coefficient increasing. By using more amounts of nanoparticles, the process demonstrates reversed behaviour. Thus a decline in the convective heat transfer coefficient of tubes happens and the surface increases.

Table 2: Effect of nanofluid and its concentration on the HE design parameters.

Parameter	Plain Tube	NF 0.1 %	NF 0.2 %	NF 0.3 %	NF 0.4 %	NF 0.5 %
Area (m^2)	15.695	15.487	15.514	15.541	15.569	15.596
h_t (W/m^2 °C)	10,247	11,016	10,908	10,802	10,699	10,599
M_t (kg/s)	31.572	31.597	31.622	31.647	31.672	31.697
V_t (m/s)	3.427	2.826	2.812	2.798	2.784	2.771
Re_t	52,665	41,889	40,238	38,663	37,168	35,748

4.3 Effect of a twisted tape turbulator

To investigate the influence of twisted tape turbulator on design parameters of HE, results are compared with an HE with the plain tubes (in both cases, the fluid is water). Results are summarised in Table 3, which shows, by using this type of turbulator, heat exchanger surface will decrease about 10 %. The turbulator disrupts the boundary layer. The tube-side fluid velocity slightly increases due to the reduction in cross sectional area because of the presence of the turbulator and constant flow rate. These changes cause convective heat transfer coefficient to increase and required surface to decrease.

Table 3: Effect of the twisted tape turbulator on design parameters of a HE

Parameter	Plain Tube	Pipe with Turbulator
Area (m^2)	15.695	14.190
h_t (W/m^2 °C)	10,247	20,675
M_t (kg/s)	31.572	31.572
V_t (m/s)	3.427	3.709
Re_t	52,665	56,998

4.4 Effect of the hybrid use of nanofluid and turbulator

According to Table 4, the hybrid use of nanofluid and turbulator decreases the required surface. There is also an optimum concentration for nanoparticles of 0.2 %, above which using more nanoparticles leads to increases in area.

Table 4: Effect of the hybrid use of different concentrations of nanofluid and turbulator on HE design parameters

Parameter	Plain Pipe	Insert + NF 0.1 %	Insert + NF 0.2 %	Insert + NF 0.3 %	Insert + NF 0.4 %	Insert + NF 0.5 %
Area (m^2)	15.695	14.166	14.161	14.165	14.173	14.183
h_t (W/m^2 °C)	10,247	21,024	21,088	21,036	20,923	20,775
V_t (m/s)	3.427	3.681	3.652	3.623	3.594	3.566
Re_t	52,665	54,583	52,269	50,070	47,984	46,007

4.5 Comparison of the different scenarios results

In this section the best results of four different scenarios are compared with each other. The best result appertains to the case of the simultaneous use of the twisted tape turbulator and the Al_2O_3/water nanofluid (at an optimal concentration of 0.2 %). However, the reduction in exchanger surface is negligible compared to the case of using only a turbulator. The more economical solution is there to use only turbulator.

5. Conclusions

The effects of Al_2O_3/water nanofluid, twisted tape turbulator and hybrid use of these two methods, which is considered as new method of HTE at the design parameters of STHE by the means of RDA have been successfully evaluated. The RDA is effective in designing for a specified pressure drop even with heat transfer enhancement techniques. In the case study, results illustrate that increases in nanofluid concentration up to an optimum level insignificantly reduces the heat transfer area and associated investment costs. It is likely that the cost of using nanoparticles in the system will out-weigh the benefits. Using a twisted tape turbulator reduces the required surface by up to 10 %.

Nomenclature

A : Heat Transfer Area (m^2)
C_p : Specific Heat (J/kg °C)
D : Diameter of the Tube (m)
f : Friction Factor
K : RDA Constant
L : Length of the Tube (m)
L_{tp} : Tube Pitch Length (m)
L_{bc} : Baffle Spacing Length (m)
M : Mass Flow Rate (kg/s)
N_b : Number of Baffle
NF : Nanofluid
Q : Heat Transfer Rate (W)
Nu : Nusselt Number
Pr : Prandtl Number
Re : Reynolds Number
T : Temperature (°C)
V : Velocity (m/s)
N_{tcc}: Number of Effective Tube Rows in Cross Flow
N_t : Number of Tube
F : Shell and Tube Correction Coefficient
h : Convective Heat Transfer Coefficient (W/m^2 °C)
H/D : Twist Ratio
k : Thermal Conductivity (W/m °C)
NS : Number of Shell Passes
R_l : Correction Factor for Baffle Leakage
R_b : Correction Factor for Baffle Bypass
h_{ic} : Heat Transfer Coefficient for Ideal Cross Flow
J_c : Segmental Baffle Window Correction Factor
J_l : Baffle Leakage Correction Factor
J_b : Bundle Bypass Correction Factor
J_r : Adverse Temperature Gradient Build up Correction Factor at Low Reynolds Number
J_s : Unequal Baffle Spacing Correction Factor
ΔP : Pressure Drop (Pa)
ΔT : Temperature Difference (°C)
ϕ : Nanoparticles Volume Concentration (%)
μ : Dynamic Viscosity (kg/m^2 s)
ρ : Density (kg/m^3)
ΣR : Resistance Summation

Subscripts

B : Baffle
bf : Base Fluid
I : Inner
o : Outer
LM : Log Mean
nf : Nanofluid
p : Nanoparticle
s : Shell
S : Shell Side
tp : Tube Pitch
t : Tube
T : Tube Side

Reference

Bubbico R., Celata G.P., D'Annibaleb F., Mazzarotta B., Menale C., 2015, Comparison of the Heat Transfer Efficiency of Nanofluids, Chemical Engineering Transactions, 43, 703–708.

Elias M.M., Shahrul I.M., Mahbubul I.M., Saidur R., Rahim N.A., 2014, Effect of different nanoparticle shape on STHE using different baffle angles and operated with nanofluid, Int. J. Heat Mass Transf., 70, 289–297.

Jafari Nasr M.R., Shafeghat A., 2008, Fluid flow analysis and extension of rapid design algorithm for helical baffle heat exchangers, Appl. Therm. Eng., 28, 1324–1332.

Kern D.Q., 1950, Process Heat Transfer. Mc-Graw-Hill, New York, USA.

Pak B.C., Cho Y.I., 1998, Hydrodynamic and Heat Transfer Study of Dispersed Fluids with Submicron Metallic Oxide Particles, Exp. Heat Transf., 11, 151–170.

Polley G.T., Panjeh Shahi M.H., Picon Nunez M., 1991, Rapid design algorithms for shell-and-tube and compact heat exchangers, Chem. Eng. Res. Des., 69, 435–444.

Serna M., Jiménez A., 2005, A Compact Formulation of the Bell–Delaware Method for Heat Exchanger Design and Optimization, Chem. Eng. Res. Des., 83, 539–550.

Serna M., Jiménez A., 2004, An Efficient Method for the Design of STHEs, Heat Transf. Eng., 25, 5–16.

Shekarian E., Tarighaleslami A.H., Khodaverdi F., 2014, Review of Effective Parameters on the Nanofluid Thermal Conductivity, J. Middle East Appl. Sci. Technol., 6, 776–780.Shenoy U.V., 1995, Heat Exchanger Network Synthesis: Processes Optimization by Energy and Resource Analysis, Gulf Publishing Company, Houston, USA.

Sundar L.S., Sharma K.V., 2010, Turbulent heat transfer and friction factor of Al2O3 Nanofluid in circular tube with twisted tape inserts, Int. J. Heat Mass Transf., 53, 1409–1416.

Tarighaleslami A.H., Walmsley T.G., Walmsley M.R.W., Atkins M.J., Neale J.R., 2015, Heat Transfer Enhancement in Heat Recovery Loops Using Nanofluids as the Intermediate Fluid, Chemical Engineering Transactions, 45, 991–996.

Wang Q., Chen G., Zeng M., Chen Q., Peng B., Zhang D., Luo L., 2010, Shell-side heat transfer enhancement for shell-and-tube heat exchangers by helical baffles, Chemical Engineering Transactions, 21, 217–222.