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GENERAL RESEARCH

Experimental Study of Shell-Side Heat Transfer Coefficient and Pressure Drop for an Integrally Helical Baffled Heat Exchanger Combined with Different Enhanced Tubes

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Experiments were performed to compare the shell-side heat transfer coefficient and pressure drop of an integrally helical baffle heat exchanger with rib-shaped fin tubes to those of that with low-fin tubes for oil cooling using water as a coolant. The experimental results showed that for the heat exchanger with rib-shaped fin tubes, the shell-side Nusselt and Euler numbers were augmented by 90−130% and 10%, respectively. The increase in heat transfer is significantly greater than that in pressure drop for rib-shaped fin tubes. Correlations have been suggested for both the shell-side Nusselt and Euler numbers for the two heat exchangers with different tube types and give very good agreement with experimental results. It is a promising route to use rib-shaped fin tubes instead of low-fin tubes for improving the performance of an integrally helical baffle heat exchanger.

1. Introduction

Heat exchangers are used in several industrial processes, such as refinery, ethylene, gas treatment, gas liquefaction, and polymer and power generation. The conventional shell-and-tube heat exchanger with segmental baffles is very versatile and hence is widely used, but it has some disadvantages over other types of heat exchanger. In particular, the shell-side flow path in a conventional shell-and-tube heat exchanger is wasteful on pressure drop, limits maximum thermal effectiveness, and encourages dead spots or recirculation zones where fouling can occur. The inherent deficiency of the conventional baffle arrangement is due largely to the fact that the flow pattern deviates considerably from the plug flow pattern and consequently causes a significant amount of back mixing. The shell-and-tube heat exchanger has evolved over the years, and a better understanding has been achieved, particularly on the hydrodynamics of shell-side flows. A good baffle design, while attempting to direct the flow in a plug flow manner where there is good radial mixing but no axial mixing, also has to fulfill the main function of providing adequate tube support.[1]

Helical baffle as one of the novel shell-side baffle geometries was developed in the Czech Republic. The Helixchanger technology was acquired in 1994 by ABB Lummus Heat Transfer from the original group of inventors through the VuCHE Company in the Czech Republic. The hydrodynamic and heat transfer studies of the shell side in helically baffled heat exchanger have been carried out by different researchers. Lutuha and Nemcansky[2] found that helical baffle geometry could force the shell-side flow field to approach a plug flow condition, which increased the average temperature driving force. The flow patterns induced by the baffles also caused the shell-side heat transfer to increase markedly. Kral et al.[3] conducted the hydrodynamic studies of the shell side on a helically baffled heat exchanger model made of Perspex using stimulus-response techniques, the results showed that a helically baffled heat exchanger provided an ideal shell-side geometry resulting in a uniform flow path with low degree of back mixing and nearly negligible dead volume. Performance of heat exchangers with helical baffles was discussed using the results of tests conducted on unit with various baffle geometries by Kral et al., also. Stehlik et al.[4] made a study of correction factors for shell-and-tube heat exchangers with segmental baffles as compared to that with helical baffles. Chunangad et al.[5] presented a case study on the industry application of a helically baffled heat exchanger combined with integral low finned tubes, the results showed that, at the same heat duty, both the equivalent bare tube surface and shell-side pressure drop were reduced by one-half of that required in the original segment baffled bare tube design for platform gas cooling with seawater. In all the above papers, the baffle geometry is quadrant shaped segment baffle plates arranged at an angle to the tube axis in a sequential pattern, forming a helical flow path through the shell side of the heat exchanger. This kind of baffle arrangement is a nonintegrally helical baffle geometry configuration where the interspace between the two segment baffles is the so-called triangle region; the large triangular free-flow area between the baffles can cause a bypass stream with a significant longitudinal flow component. Therefore, the flow pattern in the shell side is not exactly the perfect plug flow. This will result in a decrease in heat transfer on the shell side of the heat exchanger. Up to now, very few research works[6−8] on an integrally helical baffled heat exchanger have been reported in open literature, in particular, on heat transfer enhancement.

In the current work, comparison on heat transfer and pressure drop characteristics of rib-shaped fin tubes with low-fin tubes in the shell side of an integrally helical baffle heat exchanger are presented. The aim is to investigate the effect of fin geometry on heat transfer and pressure drop performance of the heat exchanger.

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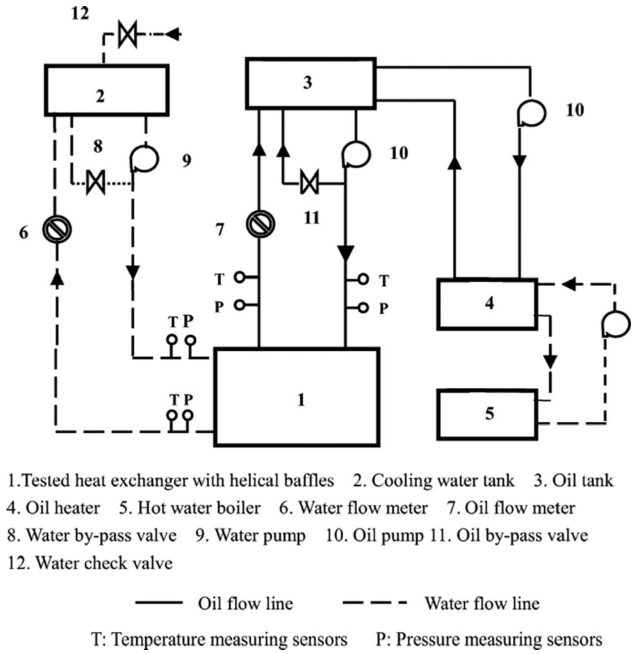

Figure 1. Schematic diagram of the experimental setup.

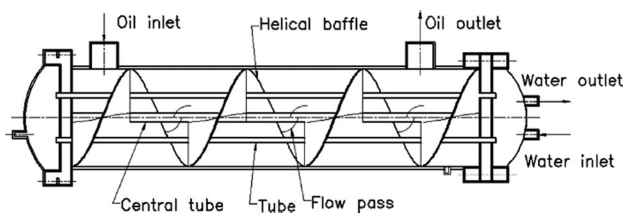

Figure 2. Schematic view of heat exchanger with integrally helical baffles.

2. Experimental Procedure

Figure 1 is the schematic of the experimental system which is used to simultaneously obtain heat transfer and pressure drop data. Two heat exchangers with integrally helical baffles that have the same configuration parameters except for different kinds of heat transfer tubes are used as the test sections. The schematic view of heat exchanger with integrally helical baffles that has one shell pass and two tube pass is showed in Figure 2. Dimensions are 675 mm in effective tube length and 110 mm in inner diameter. Total number of tubes is 20. The baffle spacing and thickness are 72 and 0.6 mm, respectively. The outer diameter of central tube is 42 mm. The heat exchangers are covered with insulating materials.

Oil is heated in an oil heater by steam generated in a water boiler and flows into an oil tank. Heated oil is pumped into the heat exchanger from an oil tank and flows in the shell side, the inlet temperature of oil is held at 65 ± 0.2 °C. Cooling water flows inside tubes. The inlet temperature of water is kept at 30 ± 0.2 °C. Calibrated T-typed thermocouples with a precision of ±0.1 °C are used to measure the temperatures of the water and oil at the inlet/outlet of the test section. The pressure drop of oil flowing through the test sections is measured by a pressure manometer with a pressure sensor with precision of ±0.25% at a range 0−100 kPa. The water and oil flow rates of the section

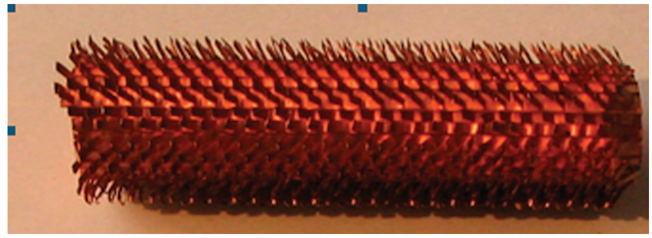

Figure 3. Photography of rib-shaped fin tube.

are measured by volumetric flow meters with an accuracy of ±1% and ±0.5%, respectively. The experiment was initiated to scan each data point while the system reached a steady-state. The procedure was repeated a few times for different flow rates of shell side ranging from 20 to 310 L/min, while the flow rate of the tube side was maintained constant at 60 L/min. After reaching stable condition, the temperature and pressure drop data are recorded by a computer-controlled Agilent 34970A data acquisition system for 15 min, maintaining a span of 5 s between two successive readings. All measurements are arithmetically averaged.

The parameters of tested tubes are shown in Table 1. Photography of the rib-shaped fin tube is shown in Figure 3. The rib-shaped fin tube is formed from a bare tube by a cold-cutting extrusion process with no metal being removed. The outer diameter over fins is larger than that of the staring bare tube size. The low-fin tube is formed from bare tube by a cold-rolling extrusion process, and the tube retains almost the same outer diameter over fins as the starting bare tube size. Both the rib-shaped fin and low-fin tubes are manufactured from bare copper tube.

3. Data Reduction and Uncertainty

To compare the characteristics of heat transfer of heat exchangers combined with different enhanced tubes, the shell-side heat transfer coefficients α_o must be determined.

The energy balance between the oil and cooling water sides was found to be within 5.0% for all runs. That is

$$\left|\frac{Q_{\mathrm{oil}} - Q_{\mathrm{w}}}{Q_{\mathrm{ave}}}\right| \le 5.0\% \tag{1}$$

where

$$Q_{\mathrm{ave}} = \frac{Q_{\mathrm{oil}} + Q_{\mathrm{w}}}{2} \tag{2}$$

$$Q_{\mathrm{oil}} = V_{\mathrm{oil}}\rho_{\mathrm{oil}}C_{\mathrm{p,oil}}(T_{\mathrm{oil,in}} - T_{\mathrm{oil,out}}) \tag{3}$$

$$Q_{\mathrm{w}} = V_{\mathrm{w}}\rho_{\mathrm{w}}C_{\mathrm{p,w}}(T_{\mathrm{w,out}} - T_{\mathrm{w,in}}) \tag{4}$$

The thermodynamic and transport properties of oil and water are calculated according to the averages of the test section inlet and outlet temperatures. The overall heat transfer coefficient of heat exchanger is computed using the mean temperature difference (MTD) method:

Table 1. The Geometrical Parameters of Tested Tubes

contents	starting bare tube mm	d_{of} mm	d_{i} mm	d_{r} mm	h_{f} mm	p mm	s mm	t mm
rib-shaped fin tube	Φ 12.0 × 1.0	16.2	10	11.6	2.3	1.2	1.2	0.2
low-fin tube	Φ 12.0 × 1.3	11.9	9.4	9.7	1.1	0.7	-	0.6

$$U = \frac{Q_{ave}}{A_o \mathrm{MTD}} \tag{5}$$

$$\mathrm{MTD} = F \times \mathrm{LMTD} \tag{6}$$

$$\mathrm{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \tag{7}$$

where A_o is heat transfer area based on outside nominal area of tubes, F is correction factor based on one shell pass and two tube passes heat exchanger design,[9] and LMTD is the log mean temperature difference.

$$A_o = N\pi d_o l \tag{8}$$

$$\Delta T_1 = T_{oil,in} - T_{w,out}, \Delta T_2 = T_{oil,out} - T_{w,in} \tag{9}$$

The overall heat transfer coefficient of heat exchanger based on the outside nominal surface area of test tubes is

$$\frac{1}{U} = \frac{1}{\alpha_o} + \frac{A_o}{A_i}R_{wall} + \frac{A_o}{A_i \times \alpha_i} \tag{10}$$

where A_i is inside surface area of test tubes, R_{wall} is the thermal resistance of the tube wall.

$$A_i = N\pi d_i l \tag{11}$$

$$R_{wall} = \frac{d_i \ln(d_o/d_i)}{2\lambda_{wall}} \tag{12}$$

The tube side heat transfer coefficients α_i were obtained by Wilson plot technique.[10]

$$\alpha_i = 0.029\frac{\lambda_w}{d_i}Re_i^{0.8}Pr^{0.4} \qquad \text{for low-fin tubes} \tag{13}$$

$$\alpha_i = 0.024\frac{\lambda_w}{d_i}Re_i^{0.8}Pr^{0.4} \qquad \text{for rib-shaped fin tubes} \tag{14}$$

It is found that the constants in correlation 13 and 14 are slightly larger than the well-known constant 0.023 of the Dittus−Boelter correlation recommended to calculate heat transfer coefficient in smooth tube for turbulent flow. The reason is that microribs are formed in the inner surface of test tubes when the fin tubes are manufactured by the extrusion process, and heat transfer coefficients of the tube side are improved.

The shell-side Nusselt number can be obtained depending on the shell-side heat transfer coefficient from the following equation:

$$Nu = \frac{\alpha_o d_r}{\lambda_{oil}} \tag{15}$$

d_r is the outer diameter of the fin tube at the fins root, λ_{oil} is the oil thermal conductivity.

The shell-side Reynolds number and Euler number are defined as follows:

$$Re = \frac{\rho u_{max} d_r}{\mu} \tag{16}$$

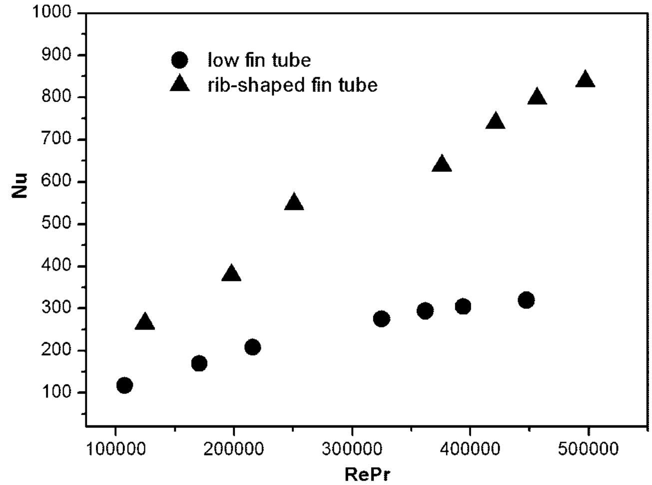

Figure 4. Variation of *Nu* with *Re* •*Pr*.

$$Eu = \frac{\Delta P}{\rho u_{max}} \tag{17}$$

where u_{max} is the maximum velocity of oil through the tube bundle. To find it, the minimum cross section area A_{min} must be evaluated. This is given as

$$A_{min} = \frac{1}{2}(D_i - d_c)S\left(1 - \frac{d_r}{P_T}\right) \tag{18}$$

where D_i is the shell inside diameter, d_c is the central tube outside diameter, S is the baffle spacing, and P_T is tube pitch.

An uncertainty analysis of the experimental results has been carried out. Experimental uncertainties in the shell-side Nusselt and Euler number were estimated by the procedure described Kline and McClitock.[11] The highest uncertainties of *Nu* and *Eu* are ±8.1% and 10.2%, respectively.

4. Results and Discussion

Experiments were performed under the condition that the shell-side pressure drops were lower than 100 kPa. Figure 4 shows the variation of the shell-side Nusselt number (*Nu*) with product of Reynolds and Prandtl number (*Re*•*Pr*) for helically baffled heat exchangers with low fin and rib-shaped fin tubes. It can be observed from Figure 4 that *Nu* increases with the increasing of *Re*•*Pr* for both low fin and rib-shaped fin tubes, and *Nu* increases more rapid with *Re*•*Pr* for rib-shaped fin tubes than low-fin tubes. Based on *Re*•*Pr* range, the shell-side Nusselt number of the helically baffled heat exchanger with rib-shaped fin tubes is 1.9−2.3 times as large as that of the helically baffled heat exchanger with low-fin tubes. The heat transfer rate is enhanced because the outside surface area of rib-shaped fin tubes is larger than that of low-fin tubes, and rib-shaped fin tubes have repeatedly interrupted flow passages along the circumferential direction of tube outside surface that create successive entrance regions resulting in the periodic boundary layer disruptions and promoting vortex shedding. Furthermore, this three-dimensional finned geometry of the rib-shaped finned tube is able to induce highly three-dimensional vorticity and good cross-flow mixing.

The variation of the shell-side Euler number (*Eu*) with *Re*•*Pr* for helically baffled heat exchangers with low fin and rib-shaped fin tubes is shown in Figure 5. The same trend in the variation of Euler number is also observed for different values of *Re*•*Pr*. The shell-side Euler number of helically baffled heat exchanger

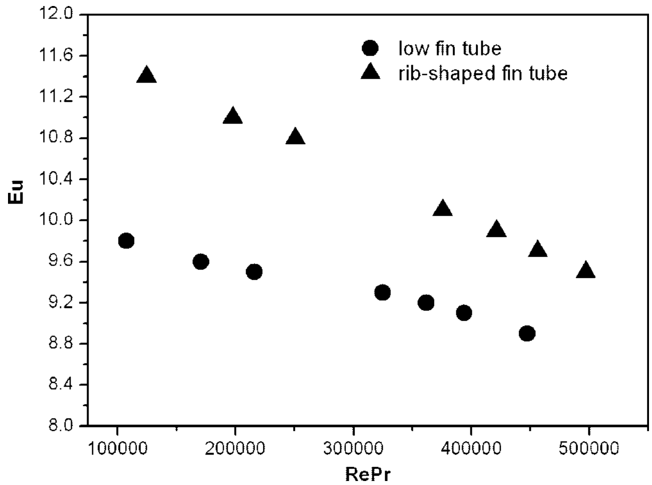

Figure 5. Variation of Eu with $Re \cdot Pr$.

Table 2. Constants Suggested for Nu and Eu in Equations 19 and 20

low-fin tube		rib-shaped fin tube		
a	m	a	m	
0.03	0.715	0.017	0.823	$Nu = a(Re \cdot Pr)^m$
19.3	−0.058	54.6	−0.132	$Eu = a(Re \cdot Pr)^m$

Table 3. Standard Deviations of Equations 19 and 20

	Δ_{av} %	
correlation	low-fin tube	rib-shaped fin tube
eq 19	3.6	5.7
eq 20	1.1	1.4

with rib-shaped fin tubes is about 10% higher than that of helically baffled heat exchanger with low-fin tubes.

Compared the experimental data in Figure 4 with Figure 5, it can be found that the increase in heat transfer is significantly greater than that of the increase in pressure drop at constant $Re \cdot Pr$ for rib-shaped fin tubes.

The following correlations are suggested for Nu and Eu (as described in literature[12]), on the shell side of shell-and-tube heat exchanger with one shell pass and two tube passes in the form of

$$Nu = a(Re \cdot Pr)^m \tag{19}$$

$$Eu = a(Re \cdot Pr)^m \tag{20}$$

On the basis of experimental results, suggested values of a and m are given in Table 2. These correlations are suggested for the range of $Re \cdot Pr$ for different tubes as follows.

For low-fin tubes:

$$10000 < Re \cdot Pr < 45000$$

For rib-shaped fin tubes:

$$12000 < Re \cdot Pr < 50000$$

Equations 19 and 20 give very good agreements with experimental results for rib-shaped fin and low-fin tubes; the standard deviations (Δ_{av}) are shown in Table 3.

5. Conclusions

This paper presents the experimental work carried out to compare the shell-side Nusselt and Euler numbers of a helically baffled heat exchanger with rib-shaped fin tubes to those with low-fin tubes for oil cooling using water as a coolant. Both the values of Nu and Eu increase with the increasing $Re \cdot Pr$ for a helically baffled heat exchanger combined with rib-shaped fin tubes and low-fin tubes, and Nuincreases more rapidly with an $Re \cdot Pr$ increase for rib-shaped fin tubes than low-fin tubes. Based on the constant $Re \cdot Pr$, the values of Nu and Eu of a helically baffled heat exchanger with rib-shaped fin tubes are about 1.9−2.3 and 1.1 times as large as that of a helically baffled heat exchanger with low-fin tubes, respectively. The increase in heat transfer is significantly greater than that of the increase in pressure drop for rib-shaped fin tubes. A simple correlation was presented for the determination of the shell-side Nusselt and Euler number of a helically baffled heat exchanger with rib-shaped fin tubes and low-fin tubes. For the rib-shaped fin tubes, $Nu = 0.017(Re \cdot Pr)^{0.823}$, $Eu = 54.6(Re \cdot Pr)^{-0.132}$; for low-fin tubes, $Nu = 0.03(Re \cdot Pr)^{0.715}$, $Eu = 19.3(Re \cdot Pr)^{-0.058}$. These correlations give very good agreement with experimental results for rib-shaped fin and low-fin tubes.

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Nomenclature

a = constant
A = surface area (m^2)
C_p = constant pressure specific heat (J/(kg·K))
d_i = inside diameter of tube (m)
D_i = inside diameter of shell (m)
d_i = inside diameter of tube (m)
$d_{o,f}$ = outside diameter over fins (m)
d_r = fin root diameter (m)
Eu = Euler number
F = correction factor
h_f = fin height (m)
l = effective length of enhanced tube (m)
LMTD = log mean temperature difference (K)
m = constant
MTD = mean temperature difference (K)
N = total number of tubes
p = fin pitch along tube axis (m)
P = pressure (Pa)
P_T = tube pitch (m)
Pr = Prandtl number
Q = heat transfer rate (W)
R = thermal resistance of the tube wall (m^2·K/W)
Re = Reynolds number
s = fin space across circumference (m)
S = baffle spacing (m)
T = temperature (K)
t = fin thickness (m)
u = oil velocity (m/s)
U = overall heat transfer coefficient (W/(m^2·K))
V = volumetric flow rate (m^3/s)
Z_i = measured value of parameter Z in the ith experimental run
Z_i^c = correlated value of parameter Z in the ith experimental run

Greek Symbols

ρ = density (kg/m^3)
α = heat transfer coefficient (W/(m^2·K))

λ = thermal conductivity (W/(m·K))
Δ = difference
μ = dynamic viscosity (Pa·s)
Δ_{av} = standard deviation, $\Delta_{av} = ((\sum_1^n ((Z_i - Z_i^c)/Z_i)^2)/n)^{1/2}$

Subscripts

ave = average
c = central tube
i = inside
in = inlet
max = maximum
min = minimum
o = outside
out = outlet
w = water

Literature Cited

(1) Stehlik, P.; Wadekar, V. V. Different Strategies to Improve Industrial Heat Exchange. *Heat Transfer Eng.* **2002**, *23*, 36.

(2) Lutcha, J.; Nemcansky, J. Performance Improvement of Tubular Heat Exchangers by Helical Baffles. *Trans. IchemE* **1990**, *68*, 263.

(3) Kral, D.; Stehlik, P.; Van Der Ploeg, H. J.; Master, B. I. Helical Baffles in Shell-and-Tube Heat Exchangers, Part I: Experimental Verification. *Heat Transfer Eng.* **1996**, *17*, 93.

(4) Stehlik, P.; Nemcansky, J.; Karl, D.; et al. Comparison of Correction Factors for Shell-and-Tube Heat Exchangers with Segmental or Helical Baffles. *Heat Transfer Eng.* **1994**, *15*, 55.

(5) Chunangad, K.; Oonnell Jr., J.; Master, B. I. *Helifin Heat Exchanger*. Proceedings of the International Conference on Compact Heat Exchangers for the Process Industries; Begell House: New York, 1997; pp 281−289.

(6) Peng, B.; Wang, Q. W.; Zhang, C. An Experimental Study of Shell-and-Tube Heat Exchangers with Continuous Helical Baffles. *J. Heat Transfer* **2007**, *129*, 1425.

(7) Son, Y. S.; Shin, J. Y. Performance of a Shell-and-Tube Heat Exchanger with Spiral Baffle Plates. *KSME Int. J.* **2001**, *15*, 1555.

(8) Zhang, Z. G.; Fang, X. M. Comparison of Heat Transfer and Pressure Drop for Helically Baffled Heat Exchanger Combined with Three-Dimensional and Two-Dimensional Finned Tubes. *Heat Transfer Eng.* **2006**, *27*, 17.

(9) Incropera, F. P.; DeWitt, D. P. *Fundamentals of Heat and Mass Transfer*; John Wiley & Sons: New York, 1990.

(10) Panchal, C. B.; France, D. M. Heat Transfer and Pressure Drop in Large Pitch Spirally Indented Tubes. *Int. J. Heat Mass Transfer* **1993**, *36*, 565.

(11) Kline, S. J.; McClintock, F. A. Describing Uncertainties in Single-Sample Experiments. *Mech. Eng.* **1953**, *1*, 3.

(12) Hosseini, R.; Hosseini-Ghaffar, A.; Soltani, M. Experimental Determination of Shell Side Heat Transfer Coefficient and Pressure Drop for an Oil Cooler Shell-and-Tube Heat Exchanger with Three Different Tube Bundles. *Appl. Therm. Eng.* **2007**, *27*, 1001.

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