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Research Article

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Theoretical and analytical analysis of performance parameters of shell and tube heat exchanger

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Abstract In the modern industrial era heat transfer management has been given prime importance to achieve an efficient system. A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, at different temperatures and in thermal contact. The shell and tube type heat exchanger are the most prominently used heat transfer equipment. The design of shell and tube heat exchangers requires a balanced approach between the thermal design and pressure drop. Whereas, most of the time the shell and tube heat exchangers are designed by compromising between heat transfer and pressure drop depending on the applications. Hence, it is very important to optimize the heat exchanger design. The objective of this project is the selection of a heat exchanger for improved thermal and hydraulic performance parameters and then the theoretical analysis of the performance parameters of the shell and tube heat exchanger. This study reports the theoretical results of the effect of the Reynolds number of the hot fluid on the thermo-hydraulic performance of the heat exchanger.

Keywords: Shell and tube heat exchanger, Reynolds number, Nusselt number.

Nomenclature

u_m	mean velocity of fluid [m/s]	m_h	mass flow rate of hot fluid [kg/s]
C_{max}	maximum heat capacity of fluid [W/K]	m_c	mass flow rate of cold fluid [kg/s]
C_{min}	minimum heat capacity of fluid [W/K]	h	heat transfer coefficient[W/m^2-K]
C_p	heat capacity [J/kg-K]	A	heat transfer area [m^2]
N_t	number of Tubes	G	channel mass velocity [kg/m^2-s]
d_i	inner diameter of tube [mm]	NTU	number of transfer units
d_o	outer diameter of tube [mm]	Nu	Nusselt Number
D_e	equivalent diameter [mm]	Re	Reynolds Number
L	length of tubes [mm]	T	temperature [K or °C]
Pe	Peclet number	U_c	overall heat transfer coefficient for clean surface [W/m^2-K]
Δp	pressure drop [Pa]	f	fanning friction factor
N_p	number of passes	k	thermal conductivity [W/m-K]
$\dot{m}$	mass flow rate of fluid [kg/s]	Pr	Prandtl number
Q	heat transfer rate [W]	Δp_t	Tube side pressure drop

h_{id}	heat transfer coefficient for pure cross flow in an ideal tube bank	N_{cw}	the number of tube rows crossed in the baffle window
ε	effectiveness	N_c	the total number of tube rows crossed in the exchanger
j_i	Colburn j factor for an ideal tube bank	N_{cw}	the number of tube rows crossed in the baffle window
N_b	Number of baffles	C_c	Heat capacity of cold fluid
Δp_s	Shell side pressure drop	T_{h_o}	Outlet temperature of the hot fluid
C_h	Heat capacity of hot fluid	T_{c_o}	Outlet temperature of the cold fluid
C^*	Heat capacity ratio		

1. Introduction

Heat exchangers are one of the most used equipment in the process industries. Heat exchangers are used to transfer heat between two process streams. One can realize their usage that any process that involves cooling, heating, condensation, boiling, or evaporation will require a heat exchanger for this purpose. Process fluids, usually are heated or cooled before the process or undergo a phase change. Different heat exchangers are named according to their application. For example, heat exchangers being used to condense are known as condensers, similarly, heat exchangers for boiling purposes are called boilers. Performance and efficiency of heat exchangers are measured through the amount of heat transfer using the least area of heat transfer and pressure drop. A better presentation of its efficiency is done by calculating the overall heat transfer coefficient. Pressure drop and area required for a certain amount of heat transfer provide insight into the capital cost and power requirements (Running cost) of a heat exchanger.

Usually, there is a lot of literature and theories to design a heat exchanger according to the requirements. Heat exchangers are of two types:

Where both media between which heat is exchanged are in direct contact with each other is Direct contact heat exchanger

Where both media are separated by a wall through which heat is transferred so that they never mix, Indirect contact heat exchanger.

A typical heat exchanger, usually for higher pressure applications up to 552 bars, is the shell and tube heat exchanger. Shell and tube type heat exchanger is an indirect contact type heat exchanger. It consists of a series of tubes, through which one of the fluids runs. The shell is the container for the shell fluid. Generally, it is cylindrical with a circular cross-section, although shells of different shapes are used in specific applications. For this particular study shell is considered, which is generally a one-pass shell. A shell is the most commonly used due to its low cost and simplicity and has the highest log-mean temperature-difference (LMTD) correction factor. Although the tubes may have single or multiple passes, there is one pass on the shell side, while the other fluid flows within the shell over the tubes to be heated or cooled. The tube side and shell side fluids are separated by a tube sheet. 3 Baffles are used to support the tubes for structural rigidity, preventing tube vibration and sagging, and to divert the flow across the bundle to obtain a higher heat transfer coefficient. Baffle spacing (B) is the center line distance between two adjacent baffles, Baffle is provided with a cut (Bc) which is expressed as the percentage of the segment height to shell inside diameter. Baffle cut can vary between 15% and 45% of the shell inside diameter. In general, conventional shell and tube heat exchangers result in high shell-side pressure drop and formation of recirculation zones near the baffles. Most of the researches nowadays are carried out on helical baffles, which give better performance than single segmental baffles but they involve high manufacturing cost, installation cost, and maintenance cost. The effectiveness and cost are two important parameters in heat exchanger design.

2. Literature Review

Reference/Title	Parameters Investigated	Conclusions
Kallannavar et al. [1]	Heat transfer rate vs tube layout was studied predominantly	The experiment was carried out for different tube layouts keeping other parameters the same. It was found that the best tube layout was 30° as the best heat transfer rate was obtained in that layout. It was also found that increase in mass flow rate the heat transfer rate decreases. It is observed that the heat transfer in counter-flow conditions is better compared to parallel flow conditions.
Kücük et al. [2]	Hs vs Re for diff. Correlation, Colburn factor vs Re, ΔP vs ms, Nu vs Re,	Mini channels in STHEs increase the heat transfer area per unit volume and enhance the convective heat transfer by minimizing the hydraulic diameter. The tube-side total pressure drop can only be controlled by changing the tube length. The experimental results were in agreement with Kern's design, within the Re numbers ranging from 250 to 2500.
Bichkar et al. [3]	Pressure drop vs mass flow rate	Increasing the number of baffles beyond a certain number has serious effects on pressure drop. So changing the types of baffles without hampering the other dimensions suggested that single segmental baffles show the maximum pressure drop which is reduced when helical baffles are used.
Mahendran et al. [4]	Pressure drop vs baffle geometry	In the simulation, it was found that the heat exchanger has more efficient cooling when compared to a normal heat exchanger. Hence flower baffles were much more efficient than normal baffles.
Mellal et al. [5]	Nu vs Re, F vs Re, For diff. configuration	A change in baffle spacing leads to a change in the heat exchanger performance, where a decrease in the baffles spacing results in an increase in the Nu and f. The baffle orientation is also an important parameter in designing an efficient shell and tube heat exchanger.
Bhatt et al. [6]	Baffle spacing Vs Overall heat transfer coefficient	A lot of factors affect the performance of the heat exchanger and the optimization obtained by the formulas depicts the cumulative effect. by changing the value of one variable by keeping the rest variable constant we can obtain different results.
Leong et al. [7]	Nu vs Re, U vs flow rate, Q vs flow rate,	The heat transfer rate is improved with copper nanoparticle volume fractions. The thermal performance of the heat recovery exchanger is increased with flue gas and coolant mass flow rate.
He et al. [8]	ΔPs vs Gs, ΔP vs Re,	Double tube pass can increase the recovered heat quality in the shell side without decreasing the heat transfer rate. Flow analysis in the shell side shows that velocity distribution in helical baffles is more uniform and homogenous as compared to segmental and flower baffles.

- It is observed that in most of the cases, there was a limit to fabricating the actual heat exchanger and performing experiments on it and many researchers tend to justify the results with computational analysis only.
- There is a need for actual experimentation for analysis of performance parameters of STHX and earlier studies are done only on modeling and simulation.
- It is observed that in most of the cases, there was a limit to fabricating the actual heat exchanger and performing experiments on it and many researchers tend to justify the results with computational analysis only.
- Hence there is a need for experimental studies for the maximum heat recovery from the waste heat source.

3. Materials and Methods

Following is the stepwise procedure for theoretical analysis (calculate performance parameters like effectiveness, heat load, and pressure drop) of STHX.

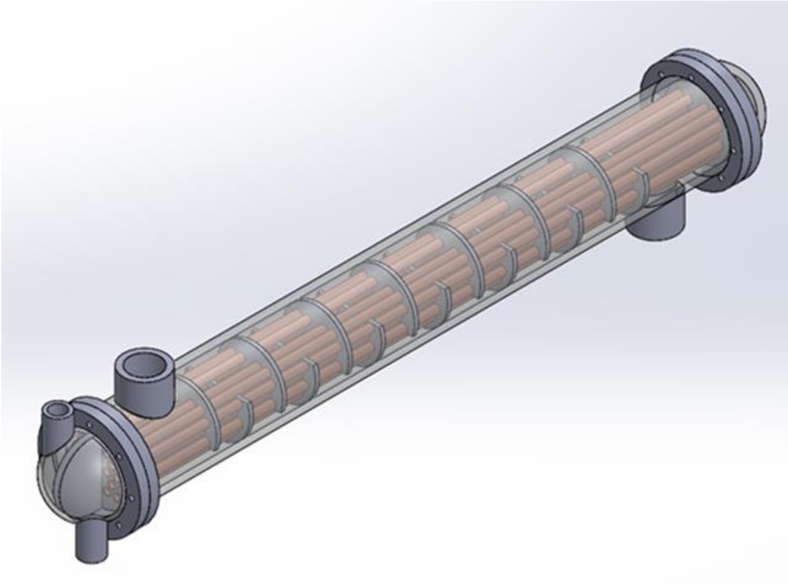

Figure 1: 3-D model of the shell and tube heat exchanger geometry.

Mass flow rate of hot and cold stream, m_h, m_c

Tube Side Calculation

Mean velocity of flow, $u_m = \frac{\dot{m}_h}{\rho_h \times A_{tp}} = \frac{\dot{m}_h}{\rho_h \times (\frac{\pi}{4} \times {d_i}^2 \times N_t)}$

Reynold Number, $Re = \frac{\rho \times u_m \times D_h}{\mu_h}$

Prandtl's Number, $Pr = \frac{\mu_h \times {C_p}_h}{k_h}$

$\frac{Pe \times d_i}{L} = \frac{Re \times \mathrm{Pr} \times d_i}{L}$

Friction factor, $f(turbulent) = (1.58 lnRe - 3.28)^{-2}$

Nusselt's Number, $Nu_t = \frac{(f/2)Re_b Pr_b}{1.07 + 12.7\left(\frac{f}{2}\right)^{\frac{1}{2}}({Pr_b}^{2/3} - 1)}$

Heat Transfer Coefficient on the inner side, $h_i = \frac{Nu_t \times k_h}{d_i}$

Tube-Side Pressure drop, $\Delta p_t = 4f \frac{LN_p}{d_i} \rho \frac{{u_m}^2}{2}$

Q_h =Hot fluid volume flow rate

Shell Side Calculation

$h_o(shell\ side\ heat\ transfer\ coefficient) = h_{id} J_c J_l J_b J_s J_r$

The values of coefficients a_1, a_2, a_3,and a_4 were taken from Sadik, and the average of values for 45^0 and 90^0 were taken as our layout angle is 75.26^0.

Thus for 10< Re< 100 a_1=0.699, a_2=-0.6345, a_3= 1.5585, a_4=0.435

$a = \frac{a_3}{1+0.14(Re_s)^{a_4}}$

$j_i = a_1 \left(\frac{1.33}{P_T/d_o}\right)^a (Re_s)^{a_2}$

$h_{id} = j_i C_{ps} \left(\frac{\dot{m}_s}{A_s}\right) \left(\frac{k_s}{C_{ps}\mu_s}\right)^{2/3}$

For J_l, J_b and J_s average values for an ideal shell and tube heat exchanger were taken from Sadik Pg398 J_l=0.75, J_b=0.9 and J_s=0.93

For J_r,

$J_r = (J_r)_r + \left(\frac{20-Re_s}{80}\right)[(J_r)_r - 1]$

$where, (J_r)_r = \left(\frac{10}{N_c}\right)^{0.18},$

$N_c = (N_{cc} + N_{cw}) * (N_b + 1),$

$J_c = 0.55 + 0.72F_c$ From E U Schlunder Design Data Book

$F_c = 1 - 2F_w$

$F_w = \frac{\theta_{ctl}}{360} - \frac{\sin\theta_{ctl}}{2\pi}$

$\theta_{ctl} = 2\cos^{-1}\left\{\frac{D_s}{D_{ctl}}\left[1 - 2\frac{B_c}{100}\right]\right\}$

$B_c = Bafffle\ Cut = 25\%$

For Pressure drop,

$\Delta p_s = [(N_b - 1)\Delta p_{bi} R_b + N_b \Delta p_{wi}] R_l + 2\Delta p_{bi}\left(1 + \frac{N_{cw}}{N_c}\right) R_b R_s$

Average values of coefficients R_l and R_b are taken**.** Hence, **R_l=0.45 and R_b=0.65**

$\Delta p_{bi} = 4f_i \frac{{G_s}^2}{2\rho_s}$

$f_i = b_1 \left(\frac{1.33}{P_T/d_o}\right)^b (Re_s)^{b_2}$

$\Delta p_{wi} = \frac{{\dot{m}_s}^2(2+0.6N_{cw})}{2\rho_s A_s A_w}$ if Re >=100

$\Delta p_{wi} = 26\frac{\mu_s \dot{m}_s}{\rho_s\sqrt{A_s A_w}}\left(\frac{N_{cw}}{P_t - d_0} + \frac{B}{{D_w}^2}\right) + \frac{{\dot{m}_s}^2}{\rho_s A_s A_w}$ if Re <=100

$N_{cw} = \frac{0.8L_c}{P_p}$

$S_{wg} = \frac{\pi}{4}(D_s)^2\left(\frac{\theta_{ds}}{360} - \frac{\sin\theta_{ds}}{2\pi}\right)$

$S_{wt} = N_{tt}F_w\left(\frac{\pi}{4}{D_t}^2\right)$

$S_w = S_{wg} - S_{wt}$

$\boldsymbol{S_w = A_w}$

$D_w = \frac{4S_w}{\pi D_t N_{tw} + \pi D_s \theta_{ds}/360}$

$(\Delta p_s = [(N_b - 1)\Delta p_{bi}R_b + N_b\Delta p_{wi}]R_l + 2\Delta p_{bi}\left(1 + \frac{N_{cw}}{N_c}\right)R_b R_s$

<u>Performance Characteristics Calculations</u>

Overall Heat Transfer Coefficient for clean surface, $U_c = \frac{1}{\frac{d_o}{d_i \times h_i} + \frac{d_o ln\left(\frac{d_o}{d_i}\right)}{2k} + \frac{1}{h_o}}$

Heat Capacity of hot fluid, $C_h = m_h \times C_h$

Heat Capacity of cold fluid, $C_c = m_c \times C_c$

Number of Transfer Units, $NTU = \frac{UA}{C_{min}}$

Heat Capacity Ratio, $C^* = \frac{C_{min}}{C_{max}}$

Effectiveness, $\varepsilon = \frac{2}{1 + C^* + (1 + {C^*}^2)^{1/2}\frac{1 + exp\left[-NTU(1 + {C^*}^2)^{1/2}\right]}{1 - exp\left[-NTU(1 + {C^*}^2)^{1/2}\right]}}$

Outlet Temperature of Hot Fluid, $T_{h_o} = T_{h_i} - \frac{(\varepsilon \times C_{min}) \times (T_{h_i} - T_{c_i})}{C_{max}}$

Outlet Temperature of Cold Fluid, $T_{c_o} = T_{c_i} + (\left(\frac{C_h}{C_c}\right) \times \left(T_{h_i} - T_{h_o}\right))$

Heat Recovery, $Q = C_h \times (T_{h_i} - T_{h_o})$

Table 1: Input parameters

Parameters	Values
Hot fluid	ISO-VG45
Cold fluid	Water
Hot fluid inlet temperature (°C)	100
Cold fluid inlet temperature (°C)	30

4. Results and Discussion

After using the appropriate correlations and fixed input parameters we obtain the geometry of the shell and tube heat exchanger by theoretical analysis as follows:

Table 2: Result for geometry of STHX by theoretical analysis.

Parameters	Values
Tube inner diameter (mm)	5.23
Tube outer diameter (mm)	6.35
No. of tubes	20
Length of tubes (m)	0.5
No. of passes (Tube side)	2
Number of baffles	14
Baffle spacing (mm)	27.46
Tube Layout	30°- Triangular
Shell Diameter (mm)	55

Table 3: Results for performance parameters (Th_i=100°C).

Re_h	U(W/m^{2}K)	ε	Nu	h_i	h_o	ΔP_t(Pa)	ΔP_s(Pa)	Heat Load(W)	Re_c
3401.16	981.1265	0.209	46.945	5528.632	1253.0	7015.299	18532.320	10927.07	6358.7575
4761.63	1258.737	0.195	64.732	7623.373	1578.2	13155.85	35987.350	14269.07	8902.2604
6122.10	1512.217	0.185	81.522	9600.756	1875.3	21101.10	59095.939	17364.11	11445.763
7482.56	1748.871	0.177	97.624	11497.03	2152.3	30819.41	87828.061	20281.53	13989.266
8843.03	1972.770	0.170	113.20	13332.24	2414.0	42287.34	122160.60	23061.46	16532.769
10203.5	1929.013	0.148	128.37	15118.93	2290.9	55486.47	143896.80	23139.82	19076.272
11563.9	2089.942	0.142	143.21	16865.64	2469.7	70401.79	183103.23	25230.81	21619.775

Figure 1: Effect of Re of hot fluid on the inner heat transfer coefficient for different Inlet temperatures (Th_i).

The variation of inner heat transfer coefficient with Re of the hot fluid is depicted in the above graph for different Th_i. The trend remains the same throughout as the h_i increases with the increase in the Reynold number. As Th_i increases the slope of the graph becomes steeper indicating a slow rate of increase of heat transfer with the increase of Re.

Figure 2: Effect of Re of hot fluid on the outer heat transfer coefficient for different inlet temperatures (Th_i).

The variation of the outer heat transfer coefficient with Re of the hot fluid is depicted in the above graph for different Thi. The trend remains the same throughout as the h_o increases with the increase in the Reynold number. In that case, for a very high Reynold number the h_o attains maximum value and then further decreases.

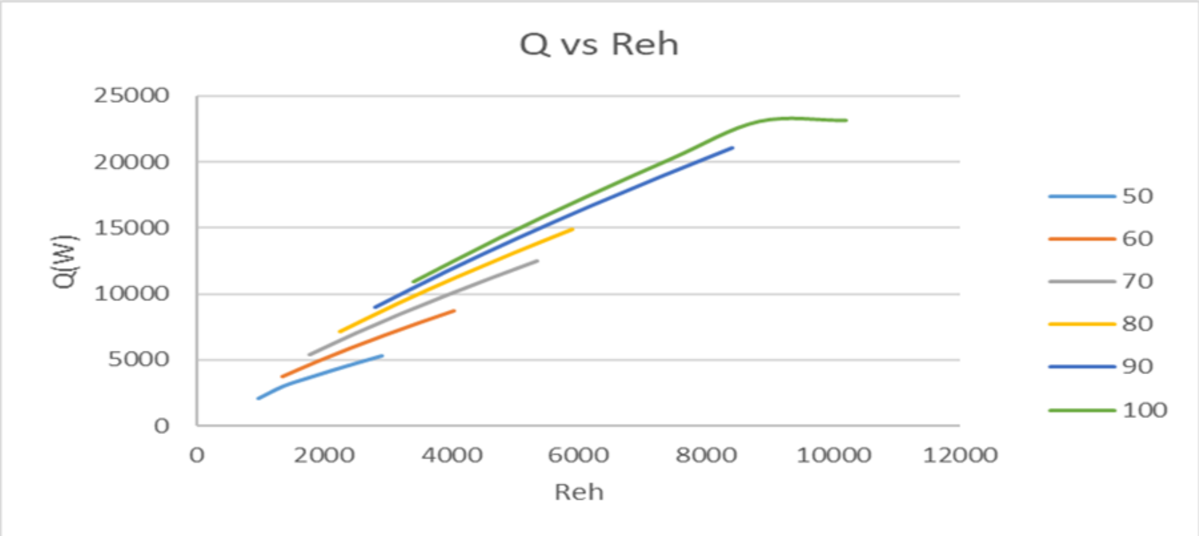

Figure 3: Effect of Re of the hot fluid on the heat recovery rate for different Inlet temperatures (Th_i).

The variation of heat transfer rate(Q) with Re (hot fluid) is depicted in the above graph. It can be observed that the rate increases with an increase in Re. For high inlet temperature i.e., Th_i=100°C the rate of heat transferred becomes constant for higher Re.

Figure 4: Effect of Re of the hot fluid on the outlet temperature of the hot fluid for different inlet temperatures (Th_i)

The variation between the outlet temperature of hot fluid (Th_o) and Re for hot fluid (Re_h) is depicted for different hot inlet temperatures (Th_i). It is observed that for increasing Re the outlet temperature is slightly increasing by a small margin. The smaller the inlet temperature, the smaller the variation of outlet temperature with increasing Reynold number.

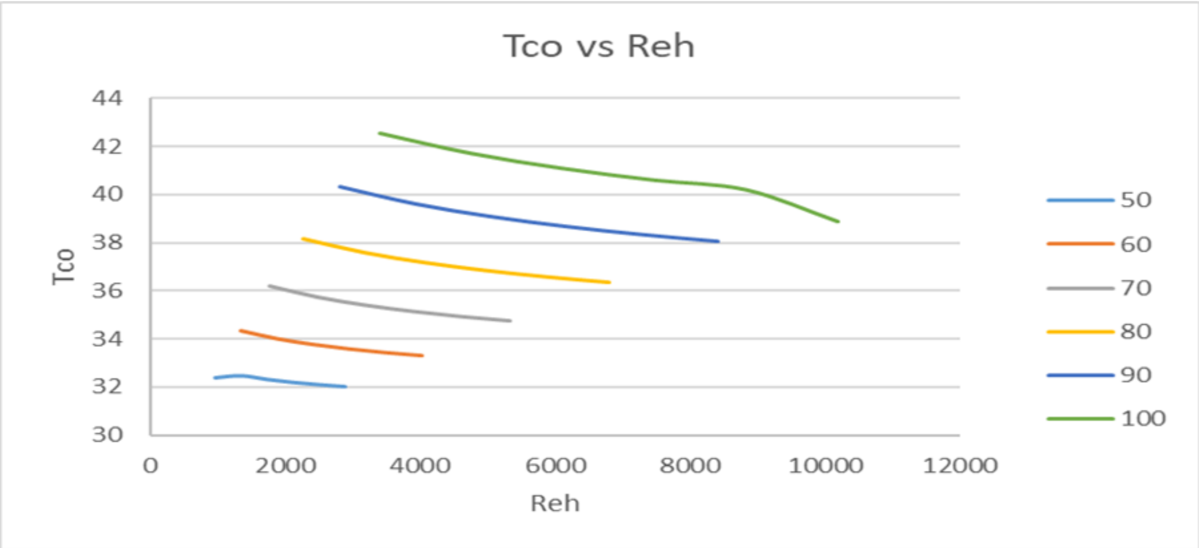

Figure 5: Effect of Re of the hot fluid on the outlet temperature of the cold fluid for different inlet temperatures (Th_i).

The variation between the outlet temperature of cold fluid (Tc_o) and Reynold number for hot fluid (Re_h) is depicted for different hot inlet temperature (Th_i). It is observed that the Temperature for the cold outlet decreases with increasing Re. The trend stays the same as we increase the Th_i.

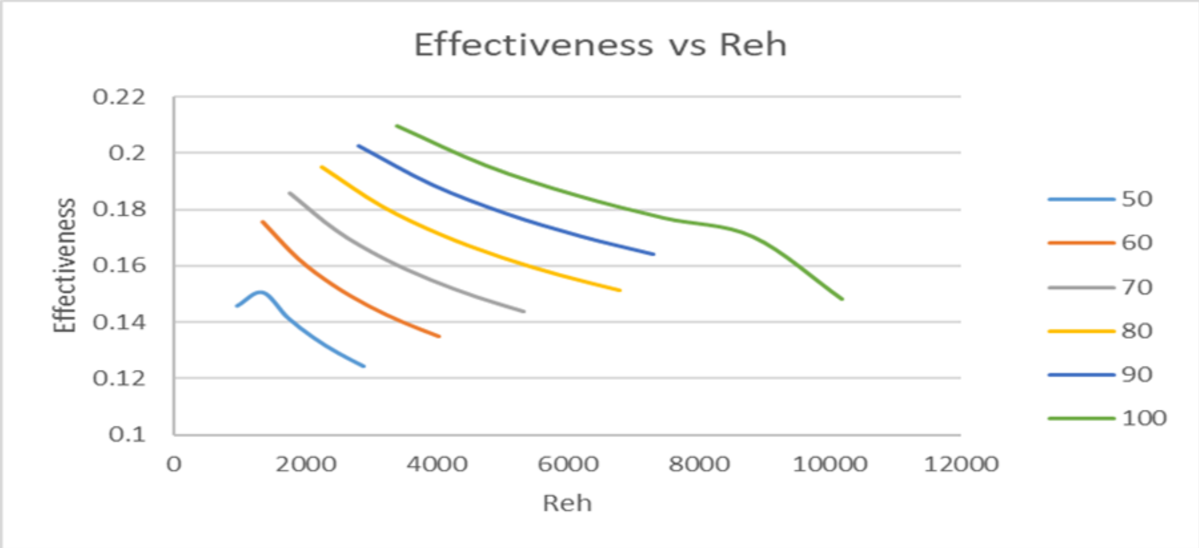

Figure 6: Effect of the Re of the hot fluid on the effectiveness of the heat exchanger for different inlet temperatures (Th_i).

The variation between effectiveness and the Reynold number for the hot fluid is depicted for different hot inlet temperatures (Th_i). It is observed that the trend is decreasing for the effectiveness as we go on increasing the Re. It also shows some abnormal trends for higher Th_i.

5. Conclusion

In the present study, theoretical analysis of Shell and Tube heat exchanger was carried out successfully. The aim was focused on investigating the performance parameters of the STHX such as heat transfer coefficient, and fluid outlet temperatures. The analysis was done for $1300 \leq Re \leq 4100$ on the tube side and $6000 \leq Re \leq 20000$ on the shell side. These characteristics were plotted against Re and these were further compared with the theoretical predictions at different inlet temperatures for hot fluid. The inner heat transfer coefficient, outer heat transfer coefficient, and heat transferred increased with an increase in Reynolds for all different inlet temperatures of the hot fluid. The outlet temperature of the hot fluid increases with an increase in Re whereas an opposite trend is observed for the outlet temperature of the cold fluid. The effectiveness of the heat exchanger decreases with an increase in the Re of the hot fluid

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