Bangladesh University of Engineering and Technology

Report Writing	
Course No	ME 310
Course Title	Thermo Fluid System Design
Project Title	Shell and Tube Heat Exchanger with Helical Baffle

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Submitted by

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1. Abstract

In the present study, three-dimensional numerical simulation of shell-and-tube heat exchangers (STHXs) with continuous helical baffle (STHXsHB) is carried out and is performed based on the simulation results. The STHXs contain 25 tubes inside a 500 mm long and 200 mm diameter shell and mass flow rate of shell-side fluid is varied from .3 kg/s to .4 kg/s. At first, physical and mathematical models are developed and numerically simulated using finite element method (FEM). For the validation of the computational model, shell-side average Nusselt number (Nu) is calculated from the simulation results and compared with the available experimental results. The comparative study shows that STHXsHB has 72-127% higher heat transfer coefficient per unit pressure drop compared to the conventional STHXsSB for the same shell-side mass flow rate. Moreover, STHXsHB has 59-63% lower shell-side pressure drop than STHXsSB.

2. Introduction

Heat exchangers are very important heat and mass exchange equipment in industries like petroleum refining, chemical engineering, electric power generation, food processing, etc. and they consume a good portion of industrial electric power for pumping the fluids. Among different types of heat exchangers, STHXs are commonly used in industries. More than 35–40% of heat exchangers are of the shell-and-tube type because of their robust construction, easy maintenance, and possible upgradation[1]. For a constant flow rate through the device, power consumption is directly proportional to the pressure drop in the device. So, energy efficiency of these devices primarily depends on the pressure drop of the fluids. Pressure drop can be thought as force required per unit area necessary to push the fluid past the surfaces and obstacles in its way as it flows through the device. Baffles are an important part of STHXs which direct the shell-side fluid flow in specified direction, ensure heat transfer effectiveness and support tube bundles. Segmental baffles are commonly used in STHXs which provide the fluid flow in tortuous, zigzag manner across tube bundles and increase heat transfer with a large pressure drop penalty[2]. Therefore, proper designing of the baffle configuration can ensure better thermal performance, increased structural stability and reduced pressure drop and thus, improved energy efficiency of STHXs. Several structures were proposed such as deflector baffle, orifice baffle, disk-and-donut configuration, spacing optimized baffles, rod baffle, etc. However, those configurations could hardly overcome the shortcomings of STHXsSB. Further improvement is going on by designing a new type of baffle called 'helical baffle', which was first introduced by Lutcha and Nemcansky[3], where they investigated the flow field patterns produced by such helical geometry with different helix angles. Several experiments and investigations were conducted on mostly discontinuous helical baffles which conclude lower shell-side pressure drop and higher heat transfer performance. Nearly all helical baffles used in STHXs are discontinuous approximate helicoids due to difficulty in manufacturing. Generally, discontinuous helical baffles are formed by joining four elliptical sector-shaped plates end to end[4]. Each baffle occupies one-quarter of the cross-section of the STHXs and is set at specific angle with respect to the axis of the STHXs. The interspace between the two sector plates, called triangle region, leads to great fluid leakage. Therefore, the flow in the shell-side is not exactly the perfect helical flow. This results in lower thermal performance on the shell-side of the heat exchanger. Moreover, the fluid flow in the shell-side of STHXs becomes more complicated due to the flow leakage in the "triangle region" if discontinuous helical baffles are used. Block plates can be used to block the flow leakage in this region. However, this will increase difficulty in manufacturing and enhance flow resistance. As a result, it is clear that the use of continuous helical baffles in STHXs can reduce the complexity of design and improve the performance of STHXs.

3. Problem Statement

In this project, a shell and tube heat exchangers' heating efficiency has been increased for a lower pressure drop than the conventional segmental baffles. In the shell side, hot fluid was entered and cold fluid passed through the tubes. This is a CFD approach.

4. Objectives

- To design an efficient shell and tube heat exchanger with helical baffle.
- To reduce the pressure drop for both shell side and tube side.
- To reduce vibration by introducing continuous shell side fluid.

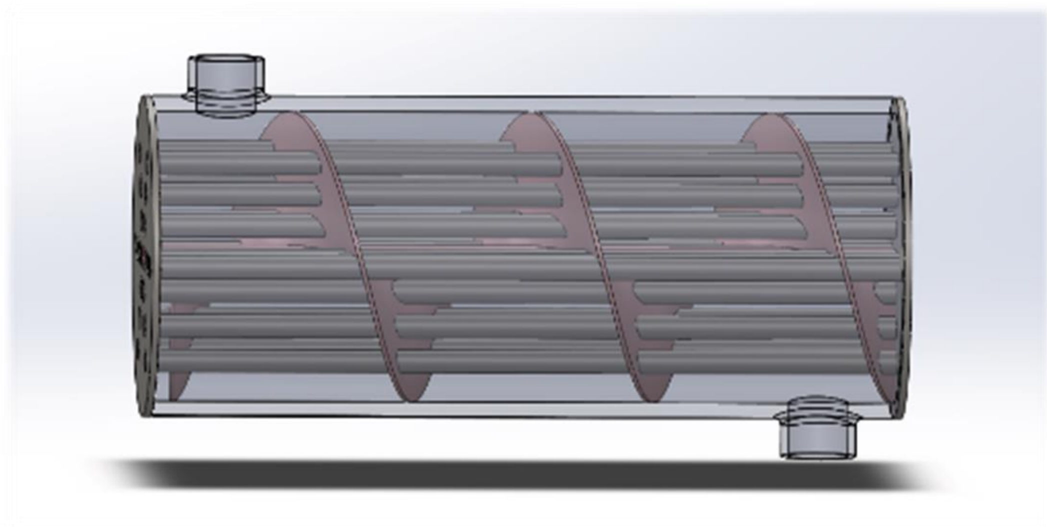
5. Design Strategy (Modeling and Simulation)

5.1 Design of Physical model

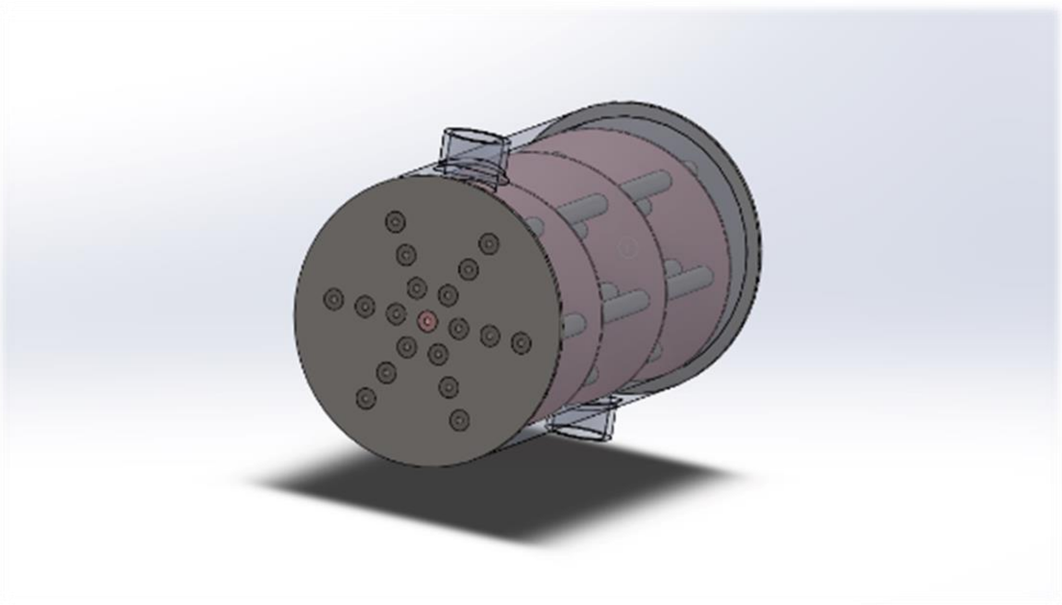
The 3-dimensional model of the Shell and Tube Heat Exchanger is shown in figure 1. Detailed geometrical parameters are listed in table 1. Two different fluids pass in a combination of counter-flow the heat exchanger. The first fluid is cold water which flows through the tubes, while hot water is the second fluid which circulates within the shell of the heat exchangers but outside of the tubes. Shell-side fluid flow pattern is varied in the heat exchangers by the application of helical baffle to get a smooth and continuous flow. The shell was of wrought iron or structural steel and tubes and baffles were made of copper. At first, baffle with multiple helical turns was designed. But as this geometry was creating complexity in the simulation software, a simplified model was designed with lower number of tubes and single turn helix shown in figure 2.

To simplify the mathematical model the following assumptions are considered here:

- Both fluid flows are turbulent (assumed based on hand calculation).
- The shell of the heat exchangers is well-insulated on outside, and hence, the heat loss to the environment is totally neglected.
- The buoyancy force is neglected.
- There is no internal heat generation and negligible viscous dissipation.



(a) Front view



(b) Isometric view

Figure 1. STHX with Helical baffle.

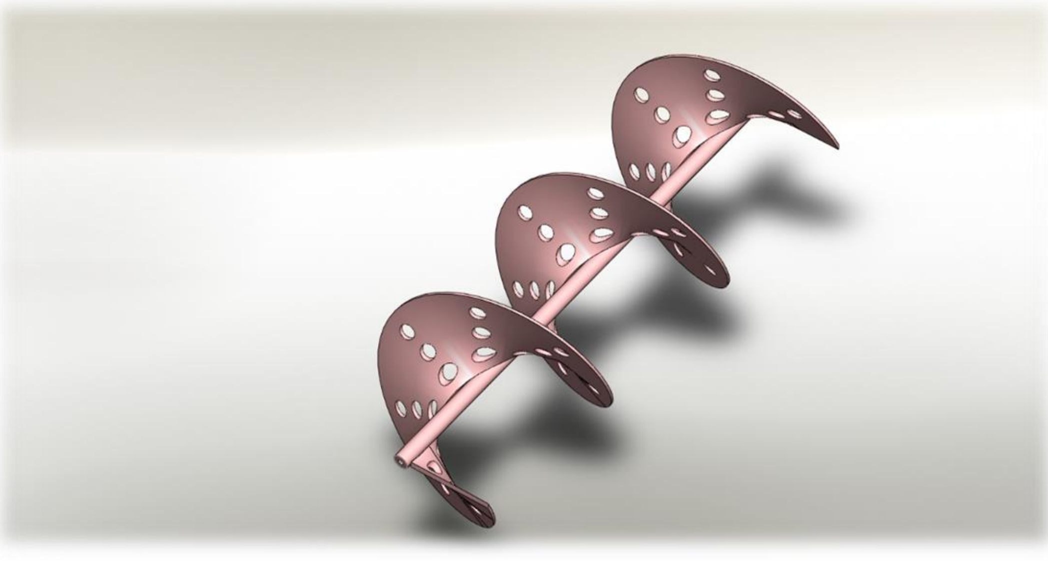
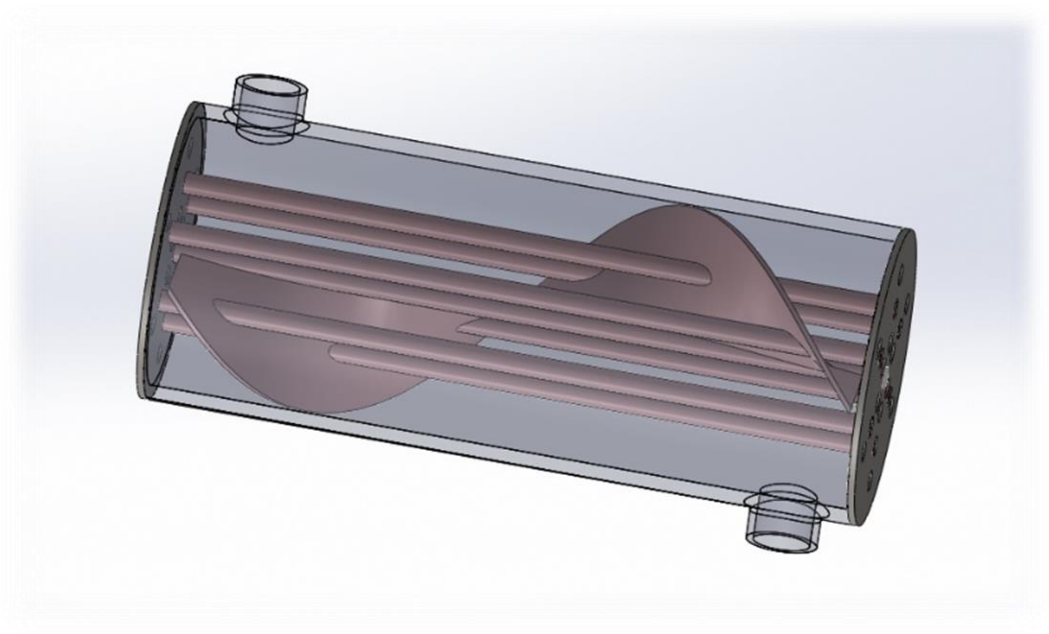
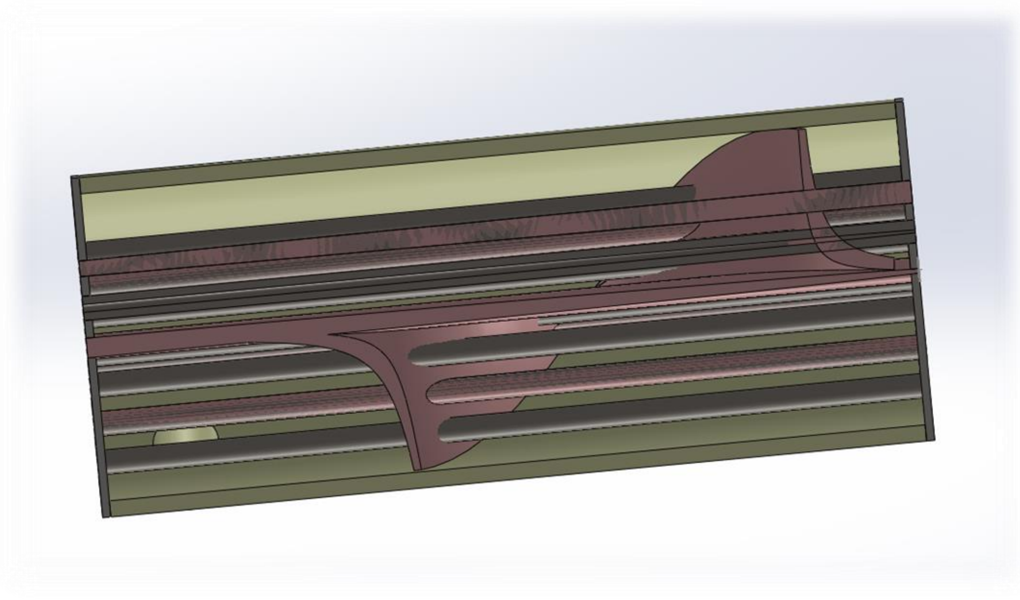


Figure 2. Helical baffle.



(a) Simplified STHX with helical baffle.



(b) Sectional view.

Figure 3. Simplified model.

5.2 Design factors:

Name of the parts	STHX(Helical Baffle)
Baffle	Pitch: 500 mm Helix angle: 32.5 Material: Copper
Shell	ID:400 mm OD:360 mm Material: Steel
Tube	ID: 15.7 mm OD: 19.05 mm Effective length: Tube layout: Triangular (30°) Tube pitch:25.4 mm Quantity:25 Material: Copper

5.3 Computational domain, mesh and boundary conditions

The heat exchanger is modelled using helical baffle. The meshes are generated using ANSYS Fluent software (student version). The element quality and skewness are checked to keep them within the limits. The pressure drop is monitored to ensure whether the STHX is safe or not and it was found that less than 70 kPa pressure drop in shell side and tube side. Helical Baffle was considered here as a solid boundary. The flow in shell and tube heat exchanger is observed to be highly turbulent. Owing to the merits of k-ε turbulence model, it is preferred for turbulence modelling in shell and tube heat exchanger. The solid walls are set with momentum boundary condition of no slip. The inlet to the shell is set as mass flow inlet. Mass flow rates are varied from 0.3 kg/s to 0.4 kg/s. The shell outlet is said to be a pressure outlet with pressure so that the inlet pressure is equal to the pressure drop. All the governing equations are solved with a second order of accuracy. Simple algorithm was used to solve the pressure linked equations. Meshing element was 363,000. Meshing element size was small because of the limitation of student version of Ansys.

6. Calculation and Results

6.1 Calculations

The calculation was done manually first as hand calculation and in HTRI software. The calculation summary is given below:

6.1.1 HTRI calculation:

Output Summary Page 1

Released to the following company:

Heat Equipment Designer Mehedi Hasan

Xist 7.3.2 2/23/2022 11:26 SN: 46639- **SI Units**

Rating - Horizontal Countercurrent Flow TEMA AES Shell With Single helical Baffles

See Data Check Messages.

See Runtime Messages.

Process Conditions		**Hot Shellside**		**Cold Tubeside**	
Fluid name		Water			water
Flow rate	(kg/s)	0.3000			1.6500
Inlet/Outlet Y	(Wt. frac vap.)	0.0000	0.0000	0.0000	0.0000
Inlet/Outlet T	(Deg C)	90.00	77.00	15.00	17.37
Inlet P/Avg	(kPa)	0.000	0.000	0.000	0.000
dP/Allow.	(kPa)	6.23	70.000	1.958	70.000
Fouling	(m2-K/W)	0.000090			0.000090

Exchanger Performance

Shell h	(W/m2-K)	879.59	Actual U	(W/m2-K)	583.48
Tube h	(W/m2-K)	4421.8	Required U	(W/m2-K)	529.23
Hot regime	(--)	Sens. Liquid	Duty	(MegaWatts)	0.0164
Cold regime	(--)	Sens. Liquid	Eff. area	(m2)	0.461
EMTD	(Deg C)	67.2	Overdesign	(%)	10.25

Shell Geometry			**Baffle Geometry**		
TEMA type	(--)	AES	Baffle type	Single Helical	
Shell ID	(mm)	200.00	Helix angle	(deg)	32.50
Series	(--)	1	Baffle crossing fraction		0.0694
Parallel	(--)	1	Central spacing (mm)		25.000
Orientation	(deg)	0.00	Helical baffle sets	(--)	1.0000

Tube Geometry			**Nozzles**		
Tube type	(--)	Plain	Shell inlet	(mm)	26.645
Tube OD	(mm)	12.700	Shell outlet	(mm)	26.645
Length	(m)	0.500	Inlet height	(mm)	24.909
Pitch ratio	(--)	1.2500	Outlet height	(mm)	24.909
Layout	(deg)	30	Tube inlet	(mm)	52.553
Tubecount	(--)	25	Tube outlet	(mm)	52.553
Tube Pass	(--)	1			

Thermal Resistance, %	
Shell	66.33
Tube	17.83
Fouling	12.35
Metal	3.49

Velocities, m/s	**Min**	**Max**
Tubeside	0.95	0.95
Crossflow	0.32	0.32
Window	--	--

Shell Construction Information

TEMA shell type	AES		Shell ID	(mm)	200.00
Shells Series	1 Parallel	1	Total area	(m2)	0.499
Passes Shell	1 Tube	1	Eff. area	(m2/shell)	0.461
Shell orientation angle (deg)	0.00				
Impingement present	No				
Pairs seal strips	2		Passlane seal rods (mm) 0.000 No. 0		
Shell expansion joint	No		Rear head support plate	No	
Weight estimation Wet/Dry/Bundle		288.50 /	262.04 /	73.16 (kg/shell)	

Baffle Information

Type		Single Helical	Helix angle	(deg)	32.50
Helical baffle sets		1.0000	Baffle crossing fraction		0.0694
Central spacing	(mm)	25.000	Helical lead	(mm)	360.38
Inlet spacing	(mm)	261.96			
Outlet spacing	(mm)	174.94			
Baffle thickness	(mm)	5.000			
Use deresonating baffles		No			

Tube Information

Tube type		Plain	Tubecount per shell		25
Overall length	(m)	0.500	Pct tubes removed (both)		4
Effective length	(m)	0.462	Outside diameter	(mm)	12.700
Total tubesheet	(mm)	38.100	Wall thickness	(mm)	1.651
Area ratio	(out/in)	1.3514	Pitch (mm) 15.875	Ratio	1.2500
Tube metal		Copper/nickel 70/30	Tube pattern (deg)		30

6.1.2 Economic calculation

NAIMA 3EPlus V4.1

Company Name
Address
City, State Zip
Phone Number

Item ID = 1
Item Description = Shell Insulation
System Application = Pipe - Horizontal
Dimensional Standard = ASTM C 585 Rigid
Calculation Type = Heat Loss Per Hour Report
Process Temperature = 90
Ambient Temperature = 23.9
Wind Speed = 0.32

Nominal Pipe Size = 200
Bare Metal = Steel
Bare Surface Emittance = 0.8
Insulation Layer 1 = 450F MF BLANKET, Type II, C553-11

Outer Jacket Material = Stainless Steel, new, cleaned
Outer Surface Emittance = 0.13

Variable Insulation Thickness	Surface Temp (°C)	Heat Loss (W/m)	Efficiency (%)
Bare	89.9	572.20	
15.0	52.6	130.60	77.18
25.0	41.1	78.95	86.20
40.0	36.6	60.06	89.50
50.0	34.0	49.27	91.39
65.0	31.9	40.86	92.86
80.0	30.8	36.30	93.66
90.0	29.9	32.86	94.26
100.0	29.2	30.18	94.73
115.0	28.7	28.02	95.10
125.0	28.2	26.24	95.41
140.0	27.9	24.75	95.67
150.0	27.6	23.48	95.90
165.0	27.3	22.39	96.09
175.0	27.1	21.43	96.25
190.0	26.9	20.59	96.40
200.0	26.7	19.84	96.53
215.0	26.5	19.17	96.65
225.0	26.4	18.57	96.75
240.0	26.2	18.02	96.85
250.0	26.1	17.52	96.94

Table 1: Variation of surface temperature with additional insulation

We estimated the wall temperature of the shell almost 90 °C. But 90 °C is the far beyond the safety limit of wall temperature. So, we had to add a significant level of insulation. Wall temperature generally decreases with additional level of insulation. Table 1 depicts the variation of wall temperature. Insulation of 25 mm sets wall temperature 41.1 °C with 86.2% efficiency. While 50 mm thickness sets wall temperature to 34 °C with 91.4 % efficiency. Pollutant reduction is another criterion in insulation selection. According to figure 2, 50 mm thickness is good enough from environmental point of view.

Variable Insulation Thickness	CO2 (kg/m/yr)	CO2 MT (MT/m/yr)	NOx (kg/m/yr)
Bare	3136.00	3.14	7.00
15.0	715.47	0.72	1.60
25.0	432.61	0.43	0.97
40.0	329.13	0.33	0.73
50.0	269.97	0.27	0.60
65.0	223.92	0.22	0.50
80.0	198.92	0.20	0.44
90.0	180.09	0.18	0.40
100.0	165.36	0.17	0.37
115.0	153.54	0.15	0.34
125.0	143.80	0.14	0.32
140.0	135.62	0.14	0.30
150.0	128.68	0.13	0.29
165.0	122.67	0.12	0.27
175.0	117.43	0.12	0.26
190.0	112.82	0.11	0.25
200.0	108.72	0.11	0.24
215.0	105.07	0.11	0.23
225.0	101.74	0.10	0.23
240.0	98.77	0.10	0.22
250.0	96.01	0.10	0.21

Table 2: Pollutant reduction by insulation.

Variable Insulation Thickness	Cost ($/m/yr)	Heat Loss (kWh/m/yr)	Savings ($/m/yr)
Bare	476.06	4761	
15.0	108.63	1086	367.43
25.0	65.68	657	410.38
40.0	49.97	500	426.09
50.0	40.99	410	435.07
65.0	34.00	340	442.06
80.0	30.20	302	445.86
90.0	27.34	273	448.72
100.0	25.11	251	450.95
115.0	23.31	233	452.75
125.0	21.83	218	454.23
140.0	20.59	206	455.47
150.0	19.54	195	456.52
165.0	18.63	186	457.43
175.0	17.83	178	458.23
190.0	17.13	171	458.93
200.0	16.51	165	459.55
215.0	15.95	160	460.11
225.0	15.45	155	460.61
240.0	14.99	150	461.07
250.0	14.58	146	461.48

Table 1: Cost of energy.

Insulation Thickness (mm)	Insulation Cost ($/m)	Fuel Cost ($/m/yr)	Fuel Savings ($/m/yr)	Payback Period (yrs)	Heat Loss (kWh/m/yr)
Bare		476.49			4761
38	76.04	50.02	426.47	0.2	500
51	89.81	41.03	435.46	0.2	410
64	102.46	34.03	442.46	0.2	340
76	115.49	30.23	446.26	0.3	302
102	139.46	25.13	451.36	0.3	251
Double Layer					
76	136.00	30.23	446.26	0.3	302
102	171.40	25.13	451.36	0.4	251
127	205.85	21.85	454.64	0.5	218
152	241.25	19.55	456.94	0.5	195
Triple Layer					
152	276.69	19.55	456.94	0.6	195
178	325.86	17.85	458.64	0.7	178
203	377.21	16.52	459.97	0.8	165
229	429.12	15.46	461.03	0.9	154
254	475.36	14.59	461.90	1.0	146

Table 2: Estimation simple payback period.

Insulation Thickness	Insulation Cost	Annualized Cost	Payback Period	Heat Loss	Surface Temp
mm	$/m	$/m	Years	J/hr/m	°C
38	76.04	59.81	0.18	216414	37
51	89.81	52.62	0.21	177517	34
64	102.46	47.28	0.24	147242	32
76	115.49	45.14	0.27	130789	31
102	139.46	43.18	0.32	108734	29
Double Layer					
76	136.00	47.80	0.32	130789	31
102	171.40	47.31	0.39	108734	29
127	205.85	48.49	0.47	94551	28
152	241.25	50.79	0.55	84609	28
Triple Layer					
152	276.69	55.38	0.63	84609	28
178	325.86	60.04	0.74	77219	27
203	377.21	65.35	0.86	71491	27
229	429.12	71.03	0.98	66905	26
254	475.36	76.15	1.08	63141	26

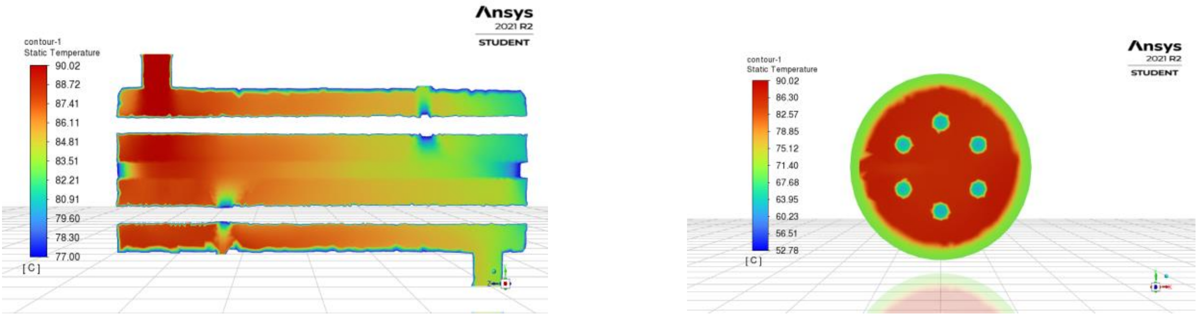
Table 3: Insulation cost estimated by FEA method

Considering all the criterion like safe operation, less pollution to environment, cost of energy, simple payback period and overall cost estimation of insulation thickness, we selected shell outer diameter to be 250 mm with 50 mm thickness.

6.2 Results

6.2.1 Simulation

In the present study, three-dimensional numerical simulations are conducted for using FEM. The inlet parameters of shell and tube sides of the heat exchangers are set to be identical for comparison of their performance. Cold water flows in the tube-side and hot water flows in the shell side of the heat exchangers and thus, heat is transferred from hot water to cold water. The visualization of the shell-side temperature distribution can be observed from the volume plots as shown in Figure 3. Left side is the hot inlet at around 90℃ and shell fluid outlet is around 77℃.



(a) **(b)**

Figure 4. Temperature variation in the shell fluid. (a) YZ plane, (b) XY plane.

Figure 4 shows the pressure distribution through pipes. The cold fluid enters the pipe from right side of the figure. The red portion shows high pressure end. This figure illustrates the pressure drop of about 2 kPa.

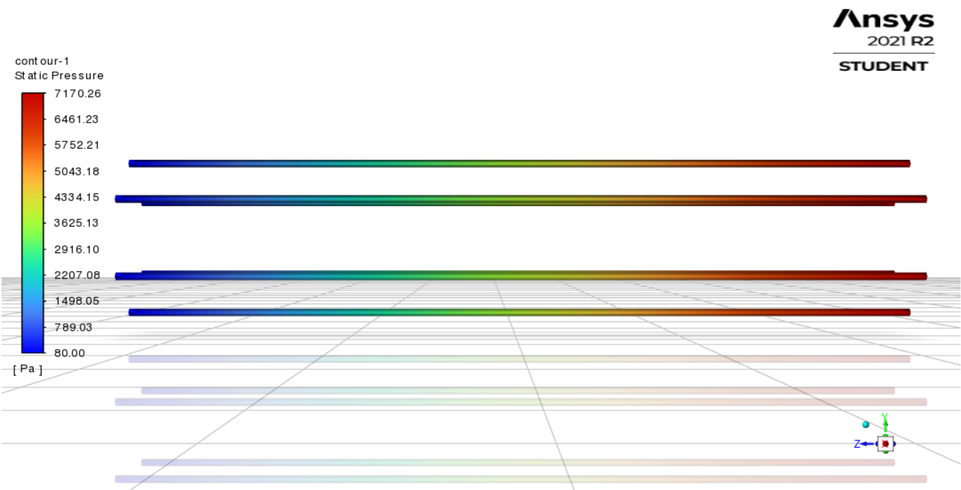


Figure 05. Pressure distribution along the pipes.

Figure 05 shows the pressure drop in the shell side. As the figure describes the entering portion has higher pressure area and the exit portion is in a lower pressure area. The pressure drop is around 6 kPa.

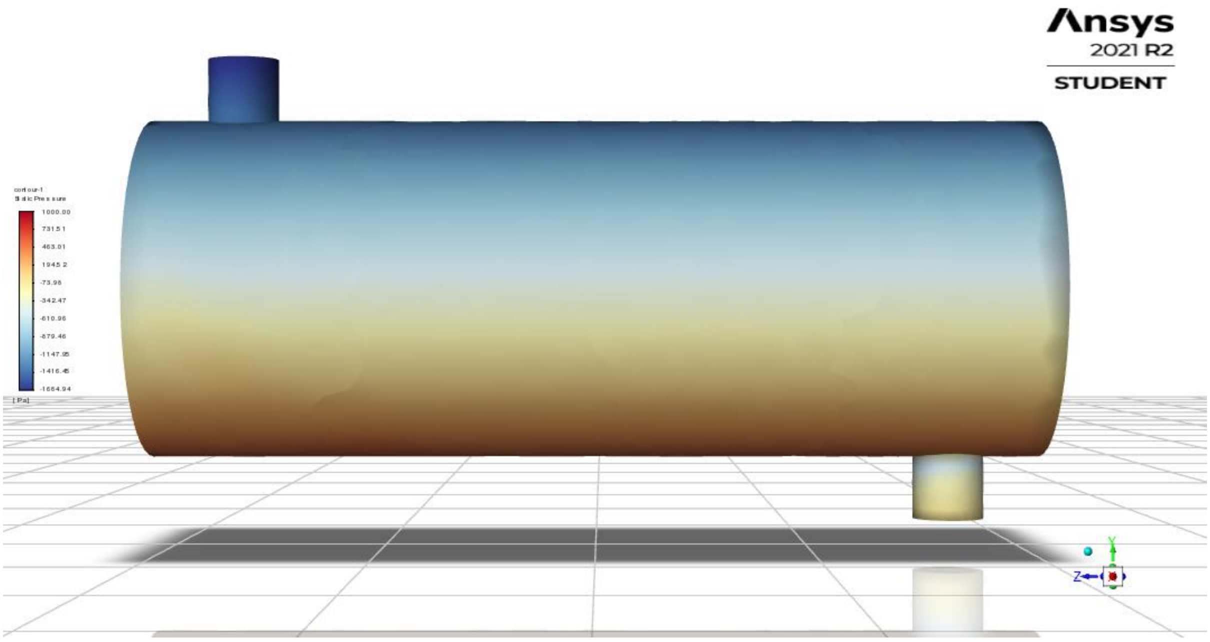
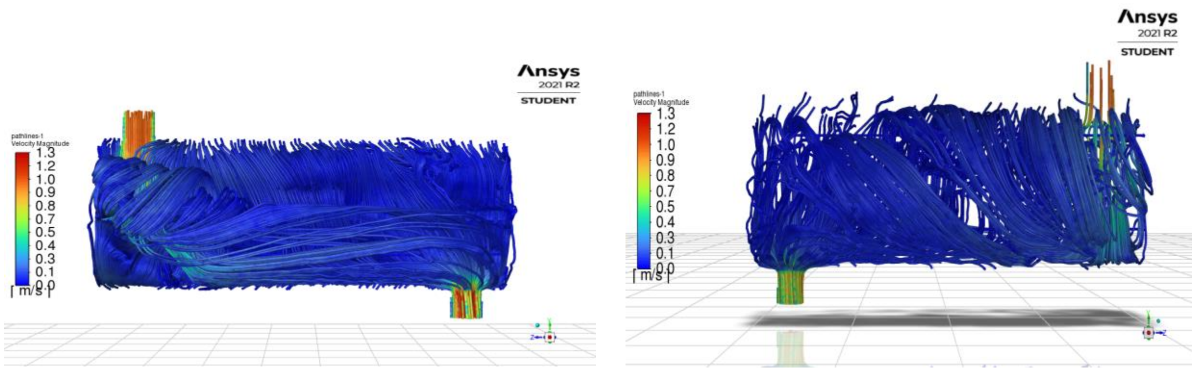


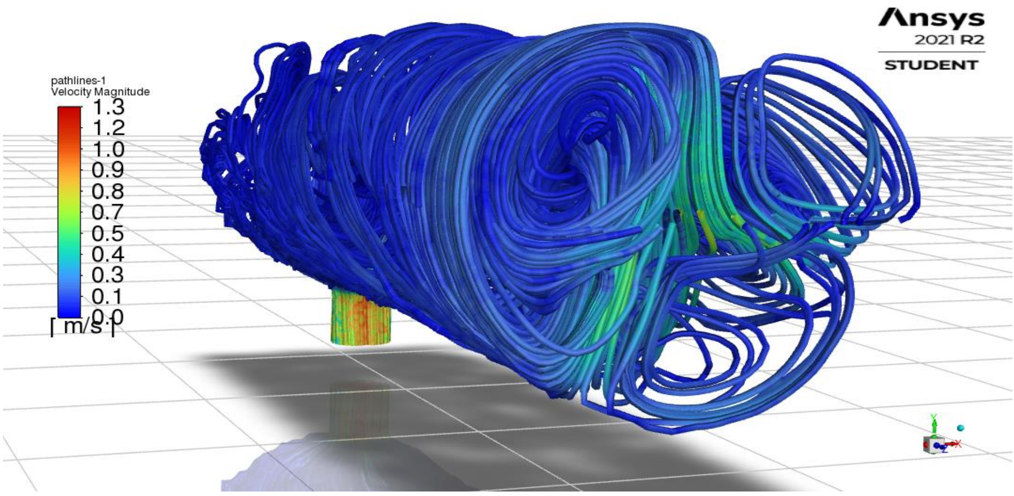
Figure 06. Shell side pressure distribution.

Figure 06 describes the flow pathlines across the shell and the helical baffle. Figure 06(a) and (b) shows a front view of the shell fluid flow. The revolution of the fluid along the baffle wall is clear from the figures below. From the simulations it is obvious that the flows are very smooth and continuous to avoid all kinds of vibration. In segmental baffle STHX this is an important issue for shell wall life.

High vibration lowers the shell life. Helical baffle is introduced to get rid of this problem. Another phenomenon is the dead zone created in the segmental baffles has been taken care of in the case of helical baffle.
Figure 6 (c) gives a better and clearer look of the flow simulation. The flow vortices around the baffle and the pipes can be observed here. So, STHX with helical baffles increases heat transfer between the fluids keeping the pressure drop in a minimal range than segmental baffles.



(a) (b)



(c)

Figure 07. Velocity pathlines for the shell fluid.

(a)Front view, (b) Back view, (c)Angle view

Figure 07 shows the shell wall temperature distribution. Shell wall temperature is around 70°C. So, insulation is necessary for the shell.

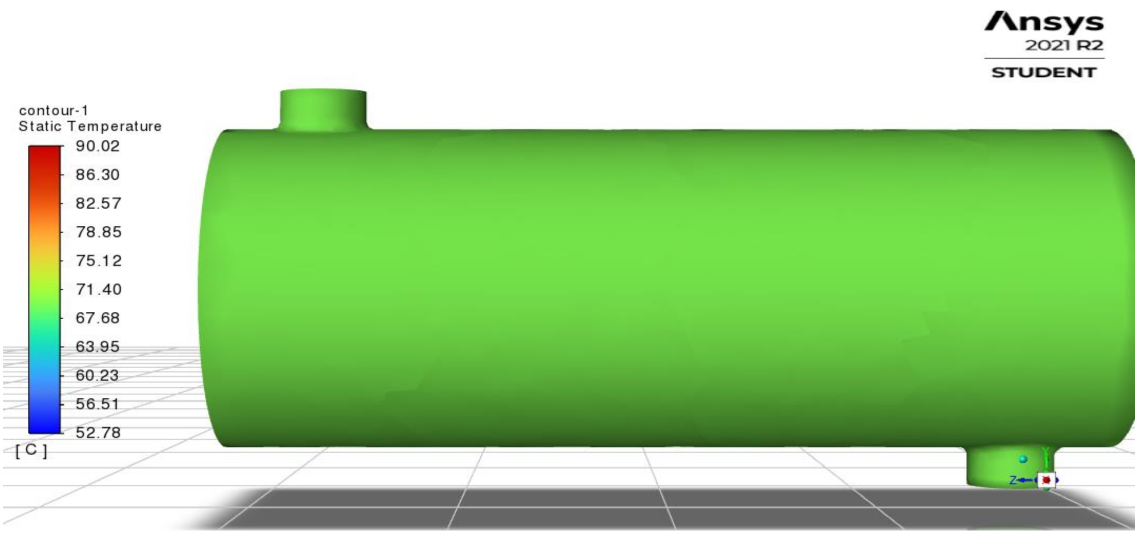


Figure 08. Shell wall temperature distribution.

A temperature contour in section plane is shown in Figure 8, this figure gives an overall idea of the temperature distribution.

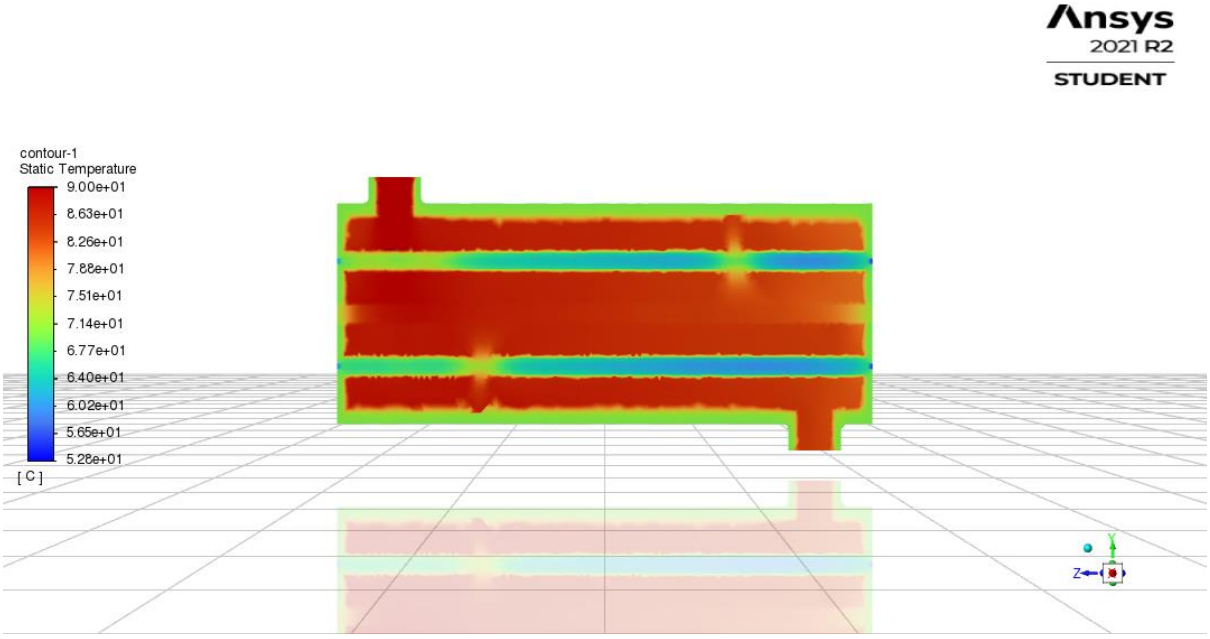
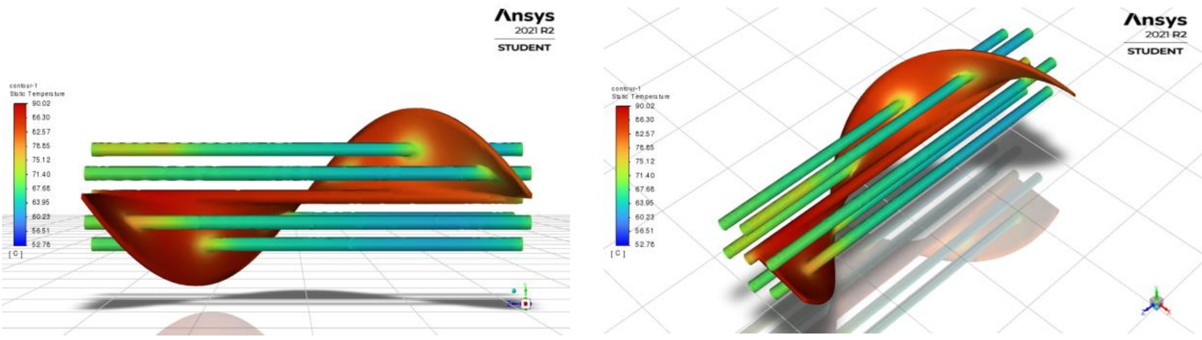


Figure 09. Temperature distribution on a section plane.

Temperature distribution on tube wall is visualized in Figure 10. The pipe walls are at moderate temperature throughout the section hot to cold from left to right. The baffle is at the same temperature as equal as the shell fluid around 90°C.



(a) (b)

Figure 10. Temperature variation on tube wall and baffle.

6.2.2 HTRI Results

Final Results — Page 5

Released to the following company:
Heat Equipment Designer
Mehedi Hasan

Xist 7.3.2 2/23/2022 11:26 SN: 46639- — **SI Units**

Rating - Horizontal Countercurrent Flow TEMA AES Shell With Single helical Baffles

Process Data		Hot Shellside		Cold Tubeside	
Fluid name			Water		water
Fluid condition			Sens. Liquid		Sens. Liquid
Total flow rate	(kg/s)		0.3000		1.6500
Weight fraction vapor, In/Out	(--)	0.0000	0.0000	0.0000	0.0000
Temperature, In/Out	(Deg C)	90.00	77.00	15.00	17.37
Skin temperature, Min/Max	(Deg C)	36.28	41.34	26.67	29.94
Wall temperature, Min/Max	(Deg C)	33.09	37.56	30.98	35.05
Pressure, In/Average	(kPa)	0.000	0.000	0.000	0.000
Pressure drop, Total/Allowed	(kPa)	6.23	70.000	1.958	70.000
Velocity, Mid/Max allow	(m/s)	0.32		0.95	
Mole fraction inert	(--)				
Average film coef.	(W/m2-K)		879.59		4421.8
Heat transfer safety factor	(--)		1.0000		1.0000
Fouling resistance	(m2-K/W)		0.000090		0.000090

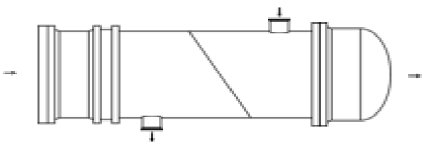
Overall Performance Data

Overall coef., Reqd/Clean/Actual	(W/m2-K)	529.23	665.67	583.48
Heat duty, Calculated/Specified	(MegaWatts)	0.0164 /		
Effective overall temperature difference	(Deg C)	67.2		
EMTD = (MTD) * (DELTA) * (F/G/H)	(Deg C)	67.23 *	1.0000	* 1.0000

See Runtime Messages Report for warnings.

Exchanger Fluid Volumes

Approximate shellside	(L)	15.3
Approximate tubeside	(L)	11.2



6.2.3 Economic consideration Results

Year	Cash In Flows	Cumulative Cash Flows
0	-42000	-42000
1	19000	-23000
2	14500	-8500
3	11000	2500
4	9000	11500
5	7000	18500
6	5000	23500
7	2000	25500
8	5500	31000

Discount Rate	10.0%

Net Present Value	12428.71
IRR	21.0%
Payback Period (Yr)	2.36

From economic consideration, the net present value (NPV) of the Heat Exchanger is 12428 taka and internal rate of return (IRR) is 21% which is greater than the overall discount rate. So, we had positive feedback from the economic consideration of the HX.

6.2.4 Graphical Analysis

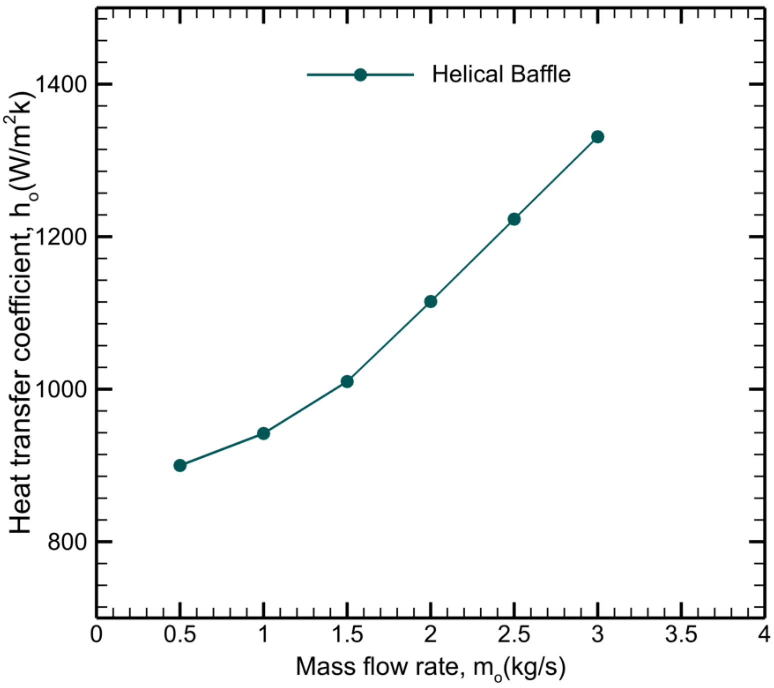


Figure 11: Variation of heat transfer coefficient with mass flow rate in shell side.

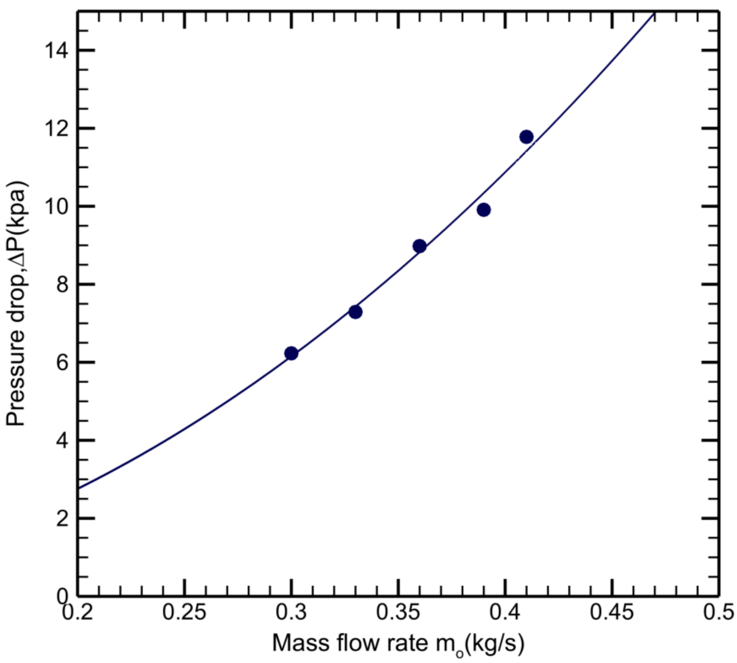


Figure 12: Variation of pressure drop with mass flow rate in shell side domain.

6.2.5 Result summary

	SHELL SIDE - WATER	TUBE SIDE - WATER
Outer Dia, mm	250	14.35
Inner Dia, mm	200	12.7
Length, mm	500	500
N_p	1	1
N_t	25	

7. Conclusion

In this study, we tried to increase the heat transfer efficiency for a shell and tube heat exchanger adding helical shaped baffles. Our main objective was to increase heat transfer with lower pressure drop. From the calculation and simulation, the pressure drop was little for both shell fluid and tube fluid. This incurs lower pumping power to flow the fluid through shells and pipes. The helical baffles used here has made the flow smoother and more continuous. As we have simulated for a simplified baffle and tube geometry, there was some mismatch between the result from simulation and software calculation. The fluid flow pattern in the shell-side of STHXsHB is rotational. STHXsHB can also reduce flow induced vibration and fouling in the shell side. Increasing the number of baffles beyond certain number gives serious effects on pressure drop. Based on the results, conventional STHXs can be replaced by STHXsHB in industrial applications to prolong service life, operational time, save energy and reduce cost of pumping due to reduced pressure drop in shell-side while maintaining high heat transfer rate.

8. Recommendation

This study has showed the performance of helical baffle for different aspects. There was some problems and limitations. If those limitations could be overcome this study could be more accurate. Limitations like meshing restrained in Ansys Student version within a range, complex helical shape was another limitation. Professional simulation could give more accurate and precise simulation.

9. Reference

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The End