BANGLADESH UNIVERSITY OF ENGINEERING AND TECHNOLOGY

ME 310

Heat Transfer Equipment Design Sessional

Project Report

Plate and Frame Heat Exchanger

Group: A11

Submitted by

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Acknowledgement

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Abstract

In this present work, a curve plate heat exchanger has been designed such that it can meet some specifications. The objective of this project is to maximize the heat transfer with minimum cost. For a given number of plates, how much heat transfer is possible, or the temperature drop has been found out in this project. The main goal of this project is to produce a thermally and hydraulically optimized heat exchanger. A curve plate has a more effective area than a straight plate. This is the main idea to maximize heat. Moreover, using corrugation makes fluids turbulent and maximizes heat transfer. There are several types of corrugation used depending on the problem. Herringbone corrugation has been used in this project. Flow optimization for different cases has been carried out for high heat transfer with keeping pressure drop as little as possible. Mathematical modeling and simulation have been done in Excel and HTRI Xchanger Suite v6.00, respectively. Mainly a number of transfer units (NTU) method has been used to model the project. After doing a mathematical simulation, the next step was to complete a full mechanical design of the entire heat exchanger for a realistic application. A 3D model of the heat exchanger has been done in SolidWorks 2017 software. To get the idea of how much heat is transferred, the model has been simulated in a simulation software. Ansys (student version 21) has been used for this purpose.

List of Abbreviation and Symbol

μ Dynamic viscosity of fluid, lbm/ft-hr

ν Kinematic viscosity of fluid, ft^2/hr

ρ Density of fluid, lbm/ft^3

k_f Thermal conductivity of fluid, BTU/(hr-ft.˚R)

k Thermal conductivity of plate, BTU/(hr-ft.˚R)

h_h Heat transfer coefficient of hot fluid, BTU/(hr-ft^2.˚R)

h_c Heat transfer coefficient of cold fluid, BTU/(hr-ft^2.˚R)

U Overall heat transfer coefficient, BTU/(hr-ft^2.˚R)

$C_{p,h}$ Specific heat capacity of hot fluid, BTU/(lbm.˚R)

$C_{p,c}$ Specific heat capacity of cold fluid, BTU/(lbm.˚R)

C_R Heat capacity ratio

Q Rate of heat transfer, BTU/s

α Thermal diffusivity, ft^2/hr

Pr Prandtl number

Re Reynolds number

Nu Nusselt number

m_h Mass flow rate of hot fluid, lbm/s

m_c Mass flow rate of cold fluid, lbm/s

$T_{h,I}$ Inlet temperature of hot fluid, ˚F

$T_{c,I}$ Inlet temperature of cold fluid, ˚F

$T_{h,0}$ Outlet temperature of hot fluid, ˚F

$T_{c,o}$ Outlet temperature of cold fluid, ˚F

V_h Hot fluid velocity, ft^2/s

V_c Cold fluid velocity, ft^2/s

L Plate length, ft

L_c Plate arc length, ft

b Plate width, ft

h Plate convex height, ft

s Plate spacing, ft

t Plate thickness, ft

A_o Flat plate surface, ft^2

A_p Curve plate surface, ft^2

A_e Effective plate surface, ft^2

A Flow area, ft^2

D_h Hydraulic diameter, ft

N Number of plate

d_p Port diameter, ft

A_{port} Port area, ft^2

Φ Enlargement factor

V_p Port velocity, ft^2/s

F Friction factor

ΔP Pressure drop, psi

1. Introduction

The heat exchanger is a device used to exchange heat between two or more fluids mainly cold and hot fluids. Here there is at least two media to exchange heat. Refrigeration, heating, and air conditioning systems, power plants, chemical processing systems, food processing systems, vehicle radiators, and waste heat recovery units are just a few of the technical applications where heat exchangers are employed. Heat exchangers used in power plants include air preheaters, economizers, evaporators, superheaters, condensers, and cooling towers. There are many types and designs of heat exchangers such as double pipe heat exchanger, shell and tube heat exchanger, plate and frame heat exchanger. In this project plate and frame heat exchanger is chosen as this type of heat exchanger offers the highest efficiency mechanism for heat transfer today in the industry. This type of heat exchanger is made of very thin plates where the plates are being corrugated as a result when fluid passes between plates this design creates high turbulence and high wall shear stress both of which lead to a high heat transfer coefficient and a high fouling resistance. Generally, two fluids pass in opposite directions for high heat transfer, but sometimes the parallel flow is also allowed. As the requirement of heat transfer, the number of plates can be increased or decreased. For this benefit, this type of heat exchanger becomes the most popular in the industry.

1.1 Project Purpose:

In this project, a curve plate heat exchanger is tried to design. The most common two types are flat plate and spiral plate heat exchangers. A plate heat exchanger is tried to design such that which can incorporate the benefit of both spiral and flat plate heat exchangers. A curve plate heat exchanger is designed expected to have such features.

1.2 Problem Statement:

A curve plate heat exchanger is used to exchange heat in a water-to-water flow system. Hot water enters the exchanger at 75°F with a mass flow rate of 2.16 lbm/s. Coldwater also at 2.18 lbm/s enters the exchanger at 45°F. The exchanger contains 12 plates. The plate spacing is 0.0167 ft., the plate material is copper, the plate thickness is 0.0033 ft., the plate width is 1.5 ft., the plate height is 3 ft.

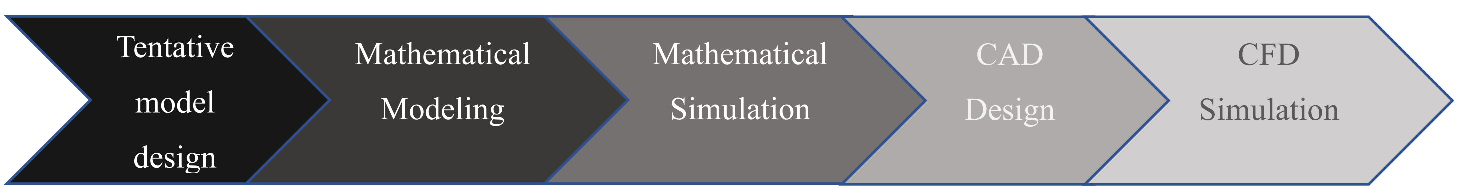
2. Objective:

The objectives of this project are to design a curve plate heat exchanger to enhance the heat transfer by increasing heat transfer area and the optimization fluid flow pass keeping pressure drop as little as possible. Flow optimization pass is the combination of parallel and counterflow. When the fluid temperature is required to drop more keeping the same number of plates, it is needed to pass the fluid again. That’s why it is needed to find out a combination of passes through the plates for maximum heat transfer. In shorts objectives are

1. Heat transfer enhancement
2. Lower Pressure drop
3. Fluid flow pass optimization

3. Design strategy:

This is the project timeline of what have done through this project. At first, a tentative model was designed to meet the objectives. Next, the mathematical modeling was done. For that excel software was used. Then mathematical simulation is conducted in HTRI for the design. After that, a cad model was designed in Solid works software. Finally, the design model was simulated in Ansys 5.6 simulation software.



3.1 Design:

3.1.1 Tentative model:

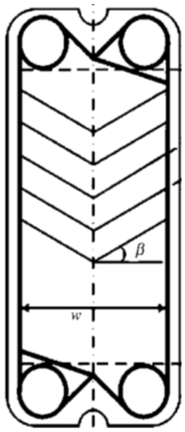


Figure 01: Common chevron flat plate

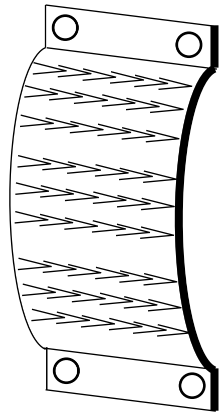


Figure 02: Our tentative designed curved plate

We have designed a model which is different from the commonly used plate heat exchanger to meet our objectives

3.1.2 Basic parameters:

As the requirement of heat transfer number of plates can be increased or decreased, it is not required to sizing the plate, moreover, rating the heat exchanger. That’s why a basic dimension was assumed as usual. A plate of 3ft by 1.5ft has been used with a 4-inch curvature height. A 1 inch circular pipe diameter has been used to carry the fluid. Herringbone corrugated of 30˚ angle has been used in the plates that create 0.2-inch flow space.

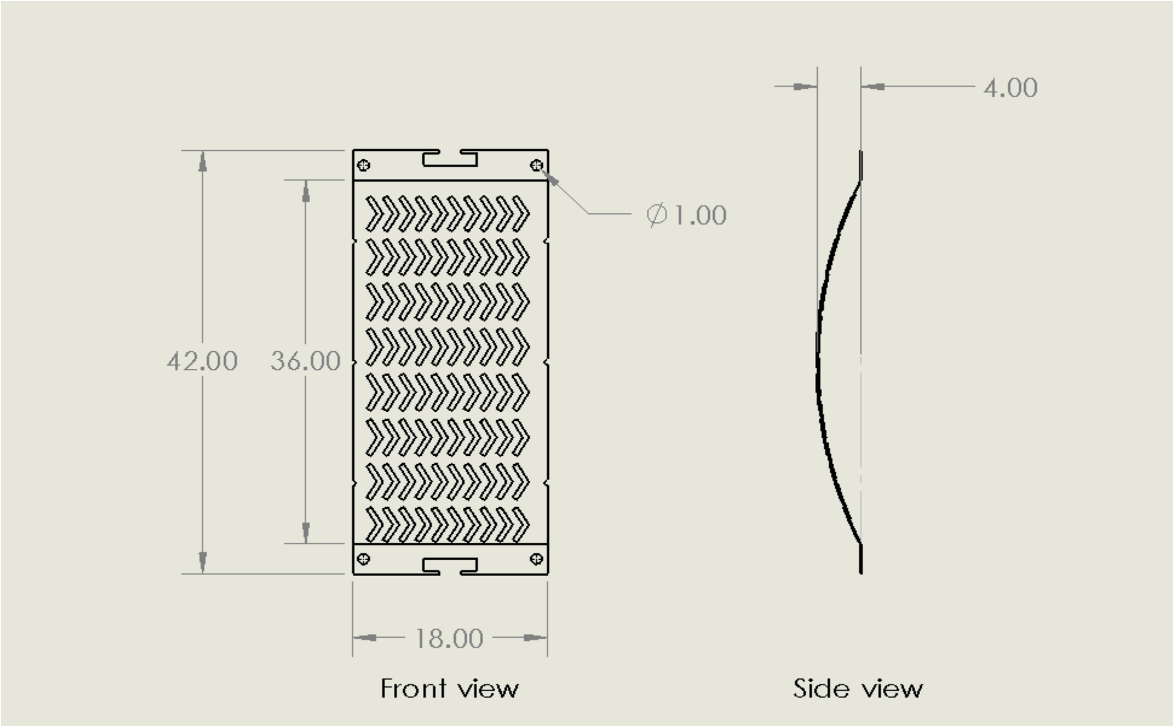


Figure 03: Design of curved plate

3.1.3 Design Parameters:

Height of plate, H=36 inch

Width of the plate, W=18 inch

Number of plates, n =12

Inlet and outlet port diameter, d=1 inch

3.1.4 Plate Specifications:

Maximum distance of curved plate from the axis of the plate, h=4 inch

Plate thickness, t= 0.04 inch

Plate material: Copper

Thermal conductivity, k=398 W/mK

3.1.5 Gasket Specifications:

Gasket material: Synthetic rubber (NBR)

Gasket thickness: 0.4 inch

Operating temperature: -4º F to 320º F

3.1.6 Fixed cover specifications:

Material: 1023 Carbon steel

Thickness: 1 inch

Thermal Expansion coefficient: 1.2×10-5/K

Thermal conductivity: 51W/mK

3.1.7 Flange Specifications:

Flange used: ASME B16.5 1 Class 150

Flange Outer Diameter = 4.25 inch

Number of bolts = 8

Bolt diameter = 0.5 inch

3.1.8 Corrugation:

Design corrugation type has a higher enlargement factor due to greater developed length. Enlargement factor is a very important factor for plate and frame heat exchangers. Enlargement factor is the ratio of plate effective heat transfer area to design area, or it may be said as developed length to protracted length. Generally, the enlargement factor becomes 1.15 to 1.25 and 1.33 for herringbone corrugation. But in this project, it becomes 1.39 which is superior. The Pitch of corrugation is the center distance between two corrugations. This type of corrugation increases turbulence by creating greater disturbance to the flow and for curve plate the effective heat transfer area increases. As a result, heat transfer is enhanced.

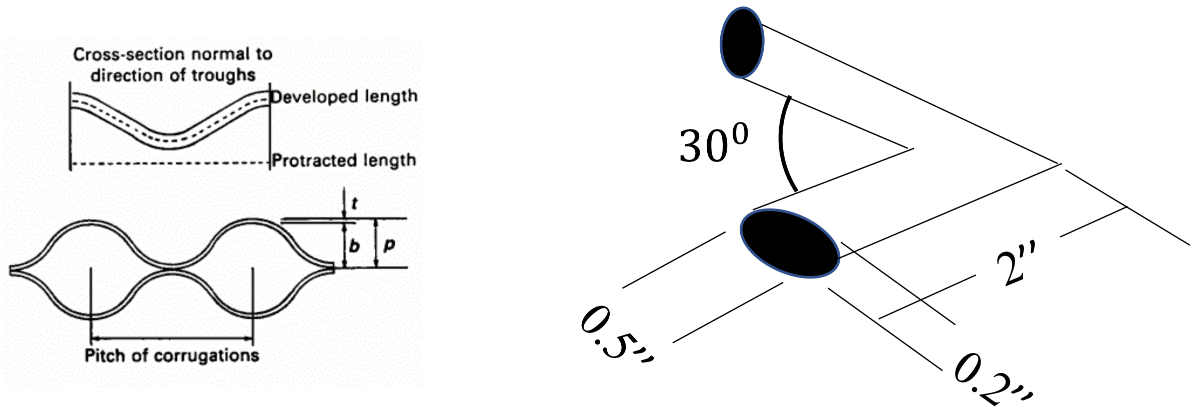


Figure 04: Herringbone corrugation

3.2 Design Drawings:

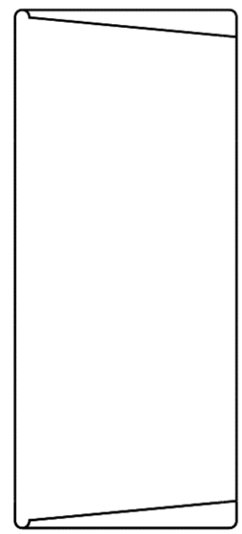


Figure 05: Isometric view of plate

Figure 06: Front view of the gasket

It is suggested to make the plates out of copper having the highest rate of thermal conductivity. But, in case of funding limitations, cheaper materials like stainless steel or aluminum might be used. The notches in the plate are to accommodate for the steel support in the assembly.

The gaskets guide the flow of fluids between the plates as well as eliminate any chance of leakage. these also ensure that the hot and cold fluids do not mix. we suggest making the gaskets out of synthetic rubber which provides the desired properties at a comparatively low cost.

Figure 07: Isometric view of curved plate heat exchanger

here we see the final assembly of the heat exchanger. the fluid enters and exits through the flanges The plates are stacked in an alternating manner to cause the counter-current flow. Multiple plates are clamped together and sealed at the edges. The design allows for the two fluids to flow in alternate directions and not be mixed. However, heat can be transferred from one medium to the other through the plates.

4. Mathematical Modeling

4.1 Effective heat transfer area

Arc radius R = $\frac{(L/2)^2+h^2}{2h}$ = $\frac{(\frac{3}{2})^2+0.333^2}{2\times0.333}$ = 3.5417 ft

Arc angle $\Theta = 2 \times sin^{-1}(\frac{L}{2\times R})$ = $2 \times \sin^{-1}\left(\frac{3}{2\times3.5417}\right) = 0.8747$

Characteristics length $L_c = R \times \Theta$ = $3.5417 \times 0.8747 = 3.0978\ ft$

Effective heat transfer area $A_e = \emptyset \times L_c \times$ b = $0.8747 \times 3.0978 \times 1.5 = 6.4732\ ft^2$

4.2 Inlet condition:

$T_{h,i} = 75°F$

$T_{c,i} = 45°F$

$m_h = 2.16\ \frac{lbm}{s}$

$m_c = 2.18\ \frac{lbm}{s}$

4.3 Thermal analysis

Re,h = $\frac{\rho\times V\times D_h}{\mu}$ = $\frac{62.3263 \times 0.23 \times0.033}{7.0228\times10^{-4}} = 681.54$

Re,c = $\frac{\rho\times V\times D_h}{\mu}$ = $\frac{62.3887 \times 0.20 \times0.033}{8.1105\times10^{-4}} = 511.97$

$Nu,h = 0.374 \times Re^{0.668} \times Pr^{1/3}$ = $0.374 \times 681.54^{0.668} \times 7.34^{1/3} = 56.78$

$Nu,c = 0.374 \times Re^{0.668} \times Pr^{1/3} = 0.374 \times 511.97^{0.668} \times 9.7^{1/3} = 51.47$

Heat transfer coefficient $h,h = \frac{Nu,h \times K_f}{D_h} = \frac{56.78 \times 0.3439}{0.0333} = 585.82\ \frac{BTU}{hr.ft^2R}$

Heat transfer coefficient $h, c = \frac{Nu, c \times K_f}{D_h} = \frac{51.47 \ \times 0.3382}{0.0333} = 522.24 \frac{BTU}{hr.ft^2R}$

Overall heat transfer coefficient U = $(\frac{1}{h_c} + \frac{1}{h_c} + R_f + \frac{t}{k})^{-1} = (\frac{1}{585.82} + \frac{1}{522.24} + \frac{0.0033}{222})^{-1}$ = $274.962\frac{BTU}{hr.ft^2R}$

Heat capacity C,h = m × $C_p = 2.16 \ \times 0.9998 = 2.16 \frac{BTU}{s.R}$

Heat capacity C,c = m × $C_p = 2.18 \ \times 1.0012 = 2.18 \frac{BTU}{s.R}$

Heat capacity Ratio C_R=$\frac{C_{min}}{C_{max}} = \frac{2.16}{2.18} = 0.99$

NTU=$\frac{UA_cN}{C_{min}} = \frac{274.962 \times 6.4732 \times 12}{2.16} = 2.75$

Effective NTU=(1-0.0166×NTU)× NTU = (1-0.0166 × 2.75) $\times 2.75 = 2.62$

Effectiveness, e=$\frac{1\text{-exp (-NTU(1-}C_R))}{1\text{-}C_R\text{exp(-NTU(1-}C_R))} = \frac{1-\exp(-2.62\times(1-0.99))}{1-0.99\times\exp(-2.62\times(1-0.99))} = 0.70$

Heat transfer, Q=e × C_{min} ΔT_{max} = 0.70 × $2.16 \ \times (75-45) = 47.079 \frac{BTU}{s}$

Outlet Temperature, $T_{h,o}$= $T_{h,i}$ +-$\frac{Q}{C_h} = 75 + \frac{47.079}{2.16} = 55.88°F$

Outlet Temperature, $T_{c,o}$= $T_{c,i}$ +-$\frac{Q}{C_c} = 45 + \frac{47.079}{2.18} = 63.92°F$

Pressure drop (hot side) ΔP=$\frac{fL\rho v^2}{D_h \times 2 \times g} + \frac{1.3 \times \rho \times v_p^2}{2g}$=0.52 psi

Pressure drop (cold side) ΔP=$\frac{fL\rho v^2}{D_h \times 2 \times g} + \frac{1.3 \times \rho \times v_p^2}{2g}$=0.49 psi

5. Mathematical simulation:

Mathematical simulation is done using HTRI software. Report generated by HTRI is given below for single-pass flow through curved plate:

HTRI

Output Summary Page 1

Released to the following HTRI Member Company:

Bangladesh University of Engineering and

Maruf Tahmid

Xphe Ver. 6.00 22-02-22 22:55 SN: Vals100+ **US Units**

Rating - Single Pass Countercurrent Flow

See Data Check Messages Report for Warning Messages.

See Runtime Message Report for Warning Messages.

Process Conditions		**Hotside**		**Coldside**	
Fluid name		Hot water		cold water	
Flow rate	(1000-lb/hr)		7.776		7.848
Temperature, Inlet/Outlet	(Deg F)	75.00	62.30	45.00	57.55
Weight fraction vapor, Inlet/Outlet	(--)	0.00	0.00	0.00	0.00
Temperature, Average/Skin	(Deg F)	68.65	62.14	51.27	60.33
Pressure, Inlet/Average	(psia)	130.727	130.340	43.704	43.310
Pressure drop, Total/Allow	(psi)	0.774	65.364	0.788	21.852
Nominal channel velocity	(ft/sec)		0.23		0.23
Fouling resistance	(ft2-hr-F/Btu)		0.00000		0.00000
Equivalent shear stress	(psi)		7.041e-5		7.477e-5
Maldistribution parameter	(--)		7.16		6.53

Exchanger Performance

Hot film coefficient	(Btu/ft2-hr-F)	264.43	Actual U	(Btu/ft2-hr-F)	121.973
Cold film coefficient	(Btu/ft2-hr-F)	227.18	Required U	(Btu/ft2-hr-F)	122.272
Hot regime		Sens. Liquid	Duty	(MM Btu/hr)	0.099
Cold regime		Sens. Liquid	Area	(ft2)	51.109
EMTD	(Deg F)	15.8	Overdesign	(%)	-0.24

Figure 08: HTRI simulation report for single-pass flow through the curved plate

Report generated by it is given below for single-pass flow through flat plate:

Output Summary Page 1

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Xphe Ver. 6.00 26-02-22 15:05 SN: Vals100+ **US Units**

Rating - Single Pass Countercurrent Flow

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See Runtime Message Report for Warning Messages.

Process Conditions		**Hotside**		**Coldside**	
Fluid name		Hot water		cold water	
Flow rate	(1000-lb/hr)		7.776		7.848
Temperature, Inlet/Outlet	(Deg F)	75.00	62.52	45.00	57.33
Weight fraction vapor, Inlet/Outlet	(--)	0.00	0.00	0.00	0.00
Temperature, Average/Skin	(Deg F)	68.76	62.14	51.17	60.31
Pressure, Inlet/Average	(psia)	130.727	130.340	43.704	43.310
Pressure drop, Total/Allow	(psi)	0.774	65.364	0.788	21.852
Nominal channel velocity	(ft/sec)		0.23		0.23
Fouling resistance	(ft2-hr-F/Btu)		0.00000		0.00000
Equivalent shear stress	(psi)		7.039e-5		7.480e-5
Maldistribution parameter	(--)		7.16		6.52

Exchanger Performance

Hot film coefficient	(Btu/ft2-hr-F)	264.49	Actual U	(Btu/ft2-hr-F)	121.966
Cold film coefficient	(Btu/ft2-hr-F)	227.11	Required U	(Btu/ft2-hr-F)	122.326
Hot regime		Sens. Liquid	Duty	(MM Btu/hr)	0.097
Cold regime		Sens. Liquid	Area	(ft2)	49.500
EMTD	(Deg F)	16.0	Overdesign	(%)	-0.29

Figure 09: HTRI simulation report for single-pass flow through the flat plate

So, it is observed that heat duty in the curved plate (0.099 MM BTU/hr) is greater than the heat duty obtained in the flat plate heat exchanger (0.097 MM BTU/hr) for the same flow condition.

5.1 Effect of the height of curvature:

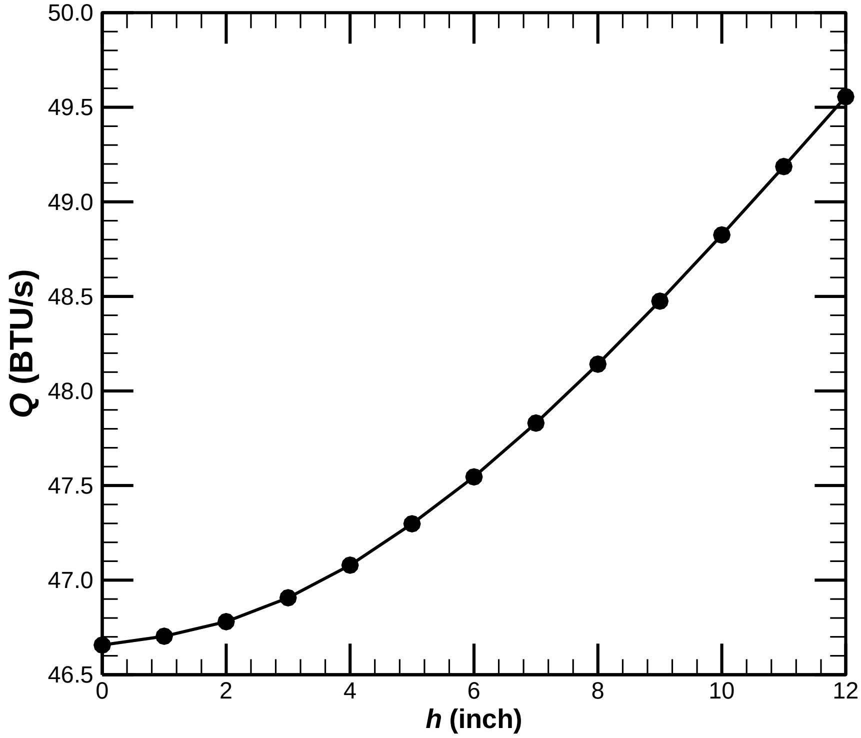


Figure 10: Variation of heat transfer with curvature height h

In this figure, it is illustrated that if the curvature of a plate is increased heat transfer rate Q must be increased. In the case of flat plate value of h obviously, zero which results in the heat transfer rate of around 46.6 BTU/s. Then with an increment of every inch of curvature heat transfer rate is significantly increased especially at a higher value of h. This curvature increases the heat transfer not only by increasing the heat transfer area but also increases the turbulence due to pressure gradient though it can be considered as a disadvantage in terms of kinetic analysis.

6. Flow optimization:

Our design is intended to come out highest heat transfer that's why flow optimization is a crucial part of our design. As we have calculated multiple passes increase the heat transfer rate significantly because the same fluid can again get touch with the oppositely heated flow. As a result, hot fluid can release heat and cold fluid can extract more heat from the hot fluid. That's why increasing pass in both sides results in heat transfer enhancement. Moreover, it is observed that the opposite inlet position of the fluid increases the heat duty of the heat exchanger.

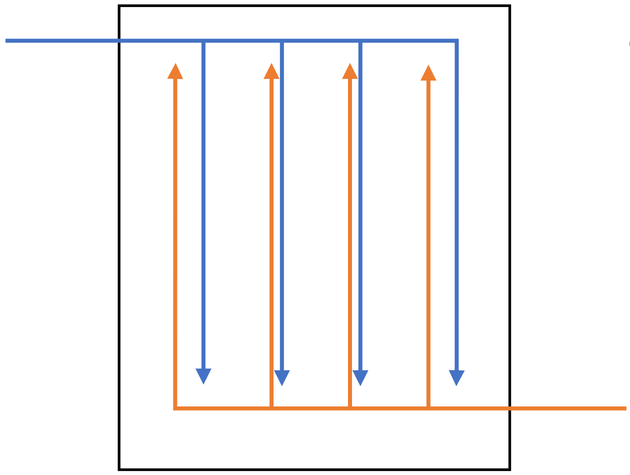


Figure 11: Single-pass flow

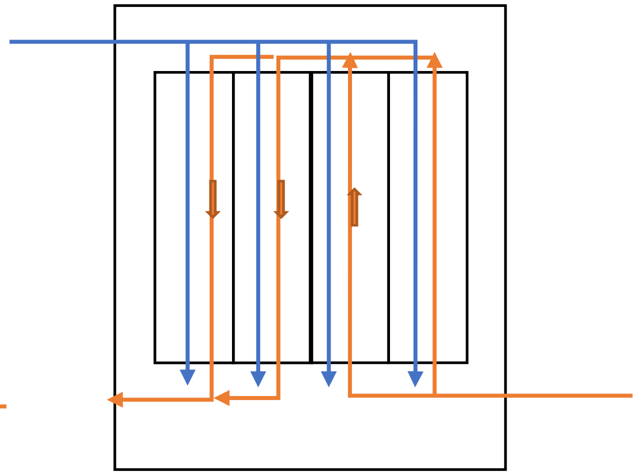


Figure 12: Multi-pass flow

Here the schematic shows how multiphase arrangement is done. In Fig firstly hot fluid pass through the first 2 channels from the opposite side of the cold inlet then the fluid get mixed again and make the second pass through the next 2 channel. This type of flow doesn't only increase heat transfer rate also makes uniform heating or cooling throughout the heat exchanger.

In the following figure, the simulation result got from HTRI software is shown which describes that heat duty increases in multiuse flow and opposite inlet. As a result, the temperature difference between inlet and outlet increases significantly.

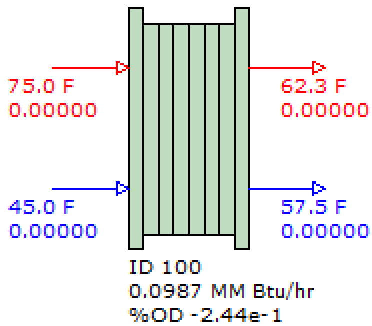
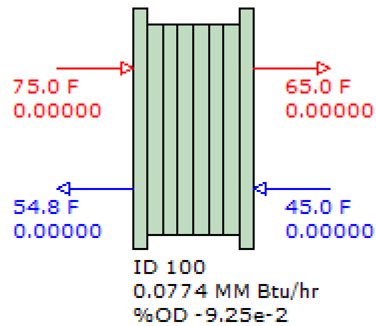
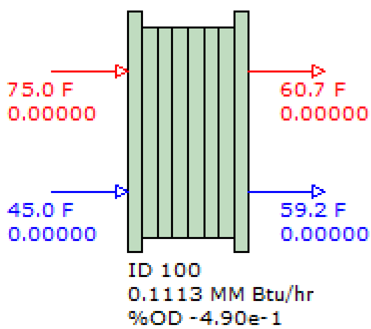
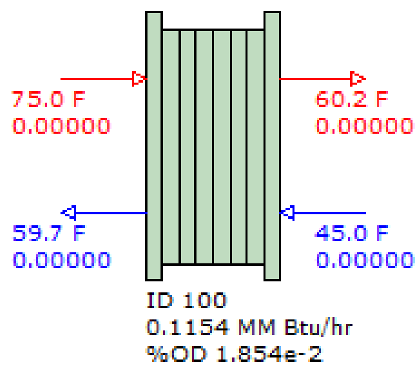
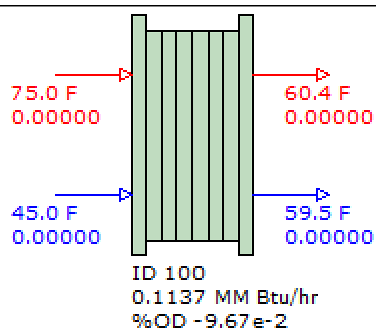
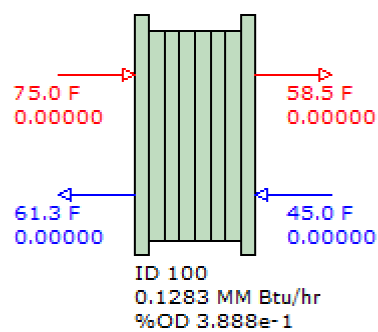
	Same side inlet	Opposite side inlet
Case 1		
Case 1		
Case 1		

Figure 13: Flow schematic for case1: Hot fluid pass1 cold fluid pass 1, case2:Hot fluid pass2 cold fluid pass 2,case3:Hot fluid pass 3 cold fluid pass 2

With an increasing number of a pass, pressure drops must be increased which is not desired. That's why it is needed to find out an appropriate pass arrangement so that pressure drop increment doesn't dominate overheat transfer increment.

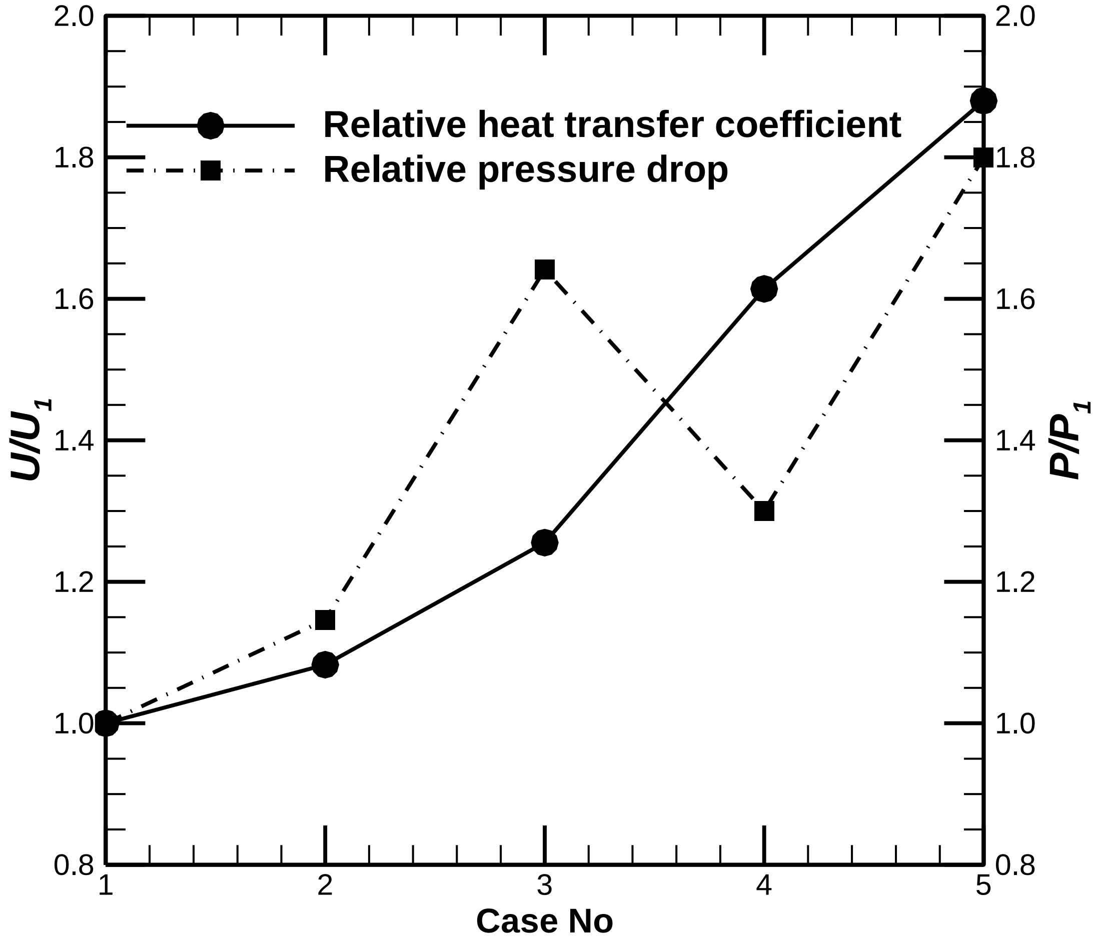


Figure 14: Variation of heat transfer coefficient relative to heat transfer coefficient for single-pass flow and variation of pressure drop relative to pressure drop for single-pass flow.

In this figure, 5 cases have been studied for relative heat transfer increment and relative pressure drop increment (relative to single-pass flow in both hot and cold fluid) case1: Hot fluid pass cold fluid pass 1, case2: Hot fluid pass2 cold fluid pass 1, case3:Hot fluid pass 3 cold fluid pass 1, case4:Hot fluid pass 2 cold fluid pass 2. case5: Hot fluid pass 3 cold fluid pass 2. It is observed that for a cold fluid single pass if the number of hot passes is increased pressure drop dominates overheat transfer increment, that's why we cannot use these cases. But if we use 2 cold passes then we get our desired heat transfer increment rate domination up to several hot fluids passes 3. So, this is our optimized flow condition, hot pass 3 cold passes 2 with opposite inlet condition.

6.1 simulation result of optimized flow:

HTRI

Output Summary Page 1

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Xphe Ver. 6.00 22-02-22 23:10 SN: Vals100+ **US Units**

Rating - Asymmetric Pass

See Data Check Messages Report for Warning Messages.

See Runtime Message Report for Warning Messages.

Process Conditions		**Hotside**		**Coldside**	
Fluid name		Hot water		cold water	
Flow rate	(1000-lb/hr)		7.776		7.848
Temperature, Inlet/Outlet	(Deg F)	75.00	58.50	45.00	61.31
Weight fraction vapor, Inlet/Outlet	(--)	0.00	0.00	0.00	0.00
Temperature, Average/Skin	(Deg F)	66.75	60.38	53.16	60.28
Pressure, Inlet/Average	(psia)	130.727	129.845	43.704	43.191
Pressure drop, Total/Allow	(psi)	1.764	65.364	1.026	21.852
Nominal channel velocity	(ft/sec)		0.69		0.46
Fouling resistance	(ft2-hr-F/Btu)		0.00000		0.00000
Equivalent shear stress	(psi)		5.221e-4		2.827e-4
Maldistribution parameter	(--)		1.24		2.44

Exchanger Performance

Hot film coefficient	(Btu/ft2-hr-F)	539.51	Actual U	(Btu/ft2-hr-F)	218.113
Cold film coefficient	(Btu/ft2-hr-F)	368.16	Required U	(Btu/ft2-hr-F)	217.269
Hot regime		Sens. Liquid	Duty	(MM Btu/hr)	0.128
Cold regime		Sens. Liquid	Area	(ft2)	51.109
EMTD	(Deg F)	11.6	Overdesign	(%)	0.39

Figure 15: Simulation report of the optimized flow of 3 hot passes with 2 cold passes and opposite inlet

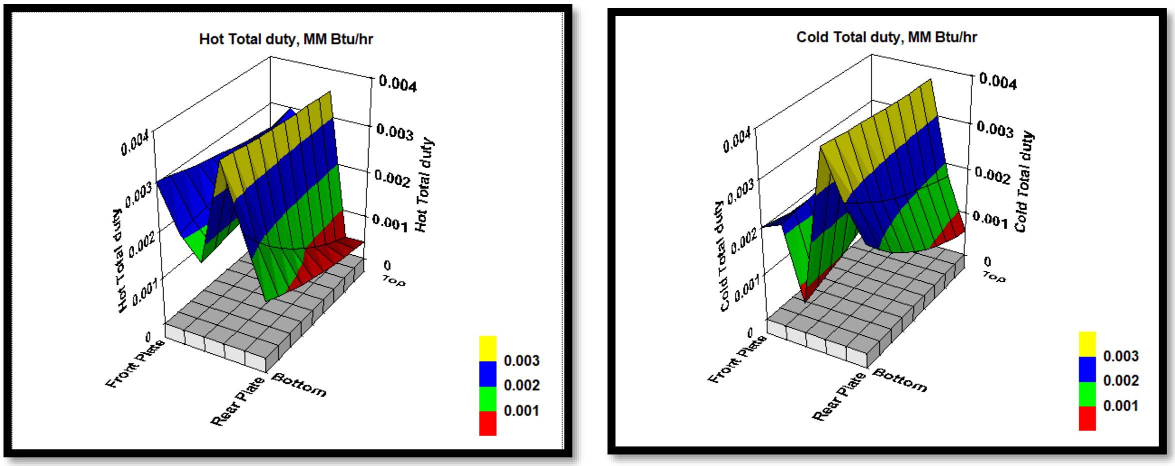

Figure 16: Simulation graph of total duty of both hot side fluid and cold side fluid of optimized flow of 3 hot passes with 2 cold passes and opposite inlet

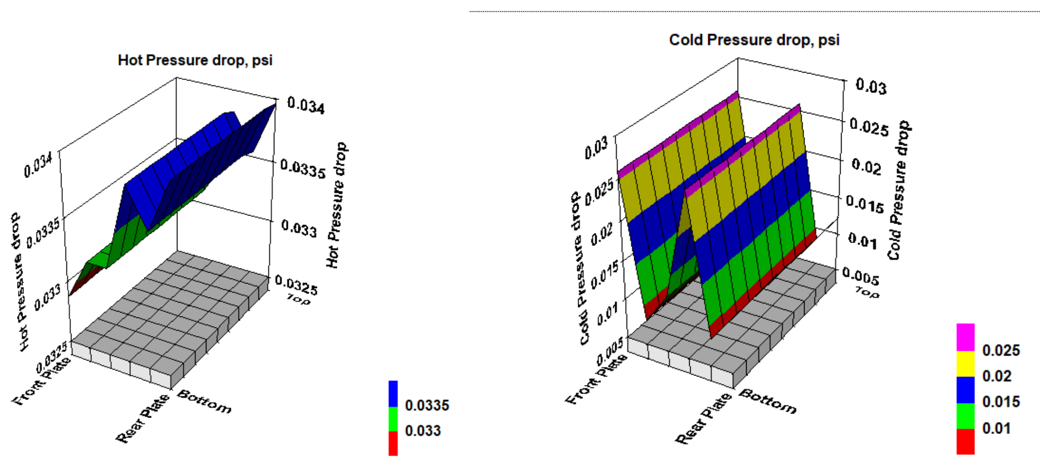

Figure 17: Simulation graph of total pressure drop of both hot side fluid and cold side fluid of optimized flow of 3 hot passes with 2 cold passes and opposite inlet

7. CFD simulation:

Flow-through the curved plate heat exchanger was intended to be simulated using ANSYS. However, due to cell limitations in the student version of the software, simulation was performed in a small prototype of the actual design. An assembly of three plates was used for this purpose. The hot fluid leaks in between the first and second plate & the cold fluid leaks in between the second and third plate. Here is an overview of the inlet and outlet of the hot and cold fluids.

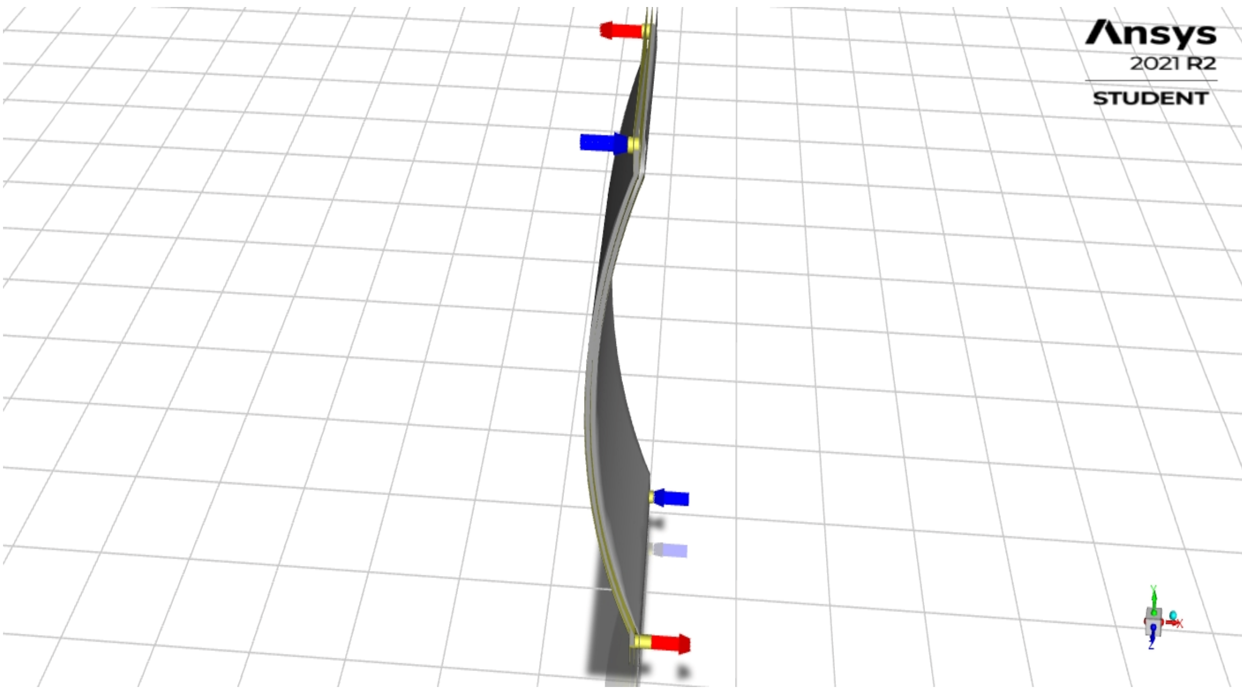

Figure 18: Inlet and outlet direction of the hot and cold fluid in simulation software

The solid material was copper and the fluid was water. K-epsilon method was applied for solving it. Since the simulation model was smaller, the final results of the simulation varied a little from the hand calculations.

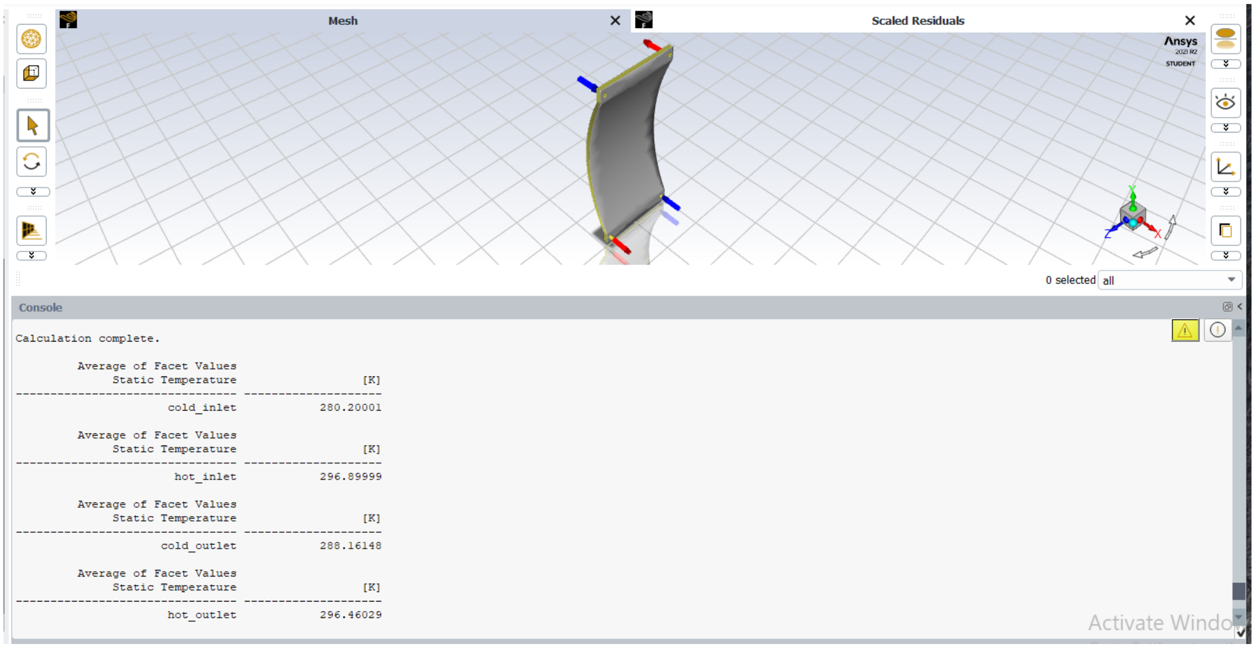

Figure 19: Average temperature of cold outlet and hot outlet in simulation software

The temperature distribution on the plate can be visualized through temperature contours. The hot side of the plate i.e. on the area where the hot fluid leaks, has temperature mostly on the higher side except for the two points; the cold inlet and the cold outlet (On the bottom left and the top left of the plate respectively).

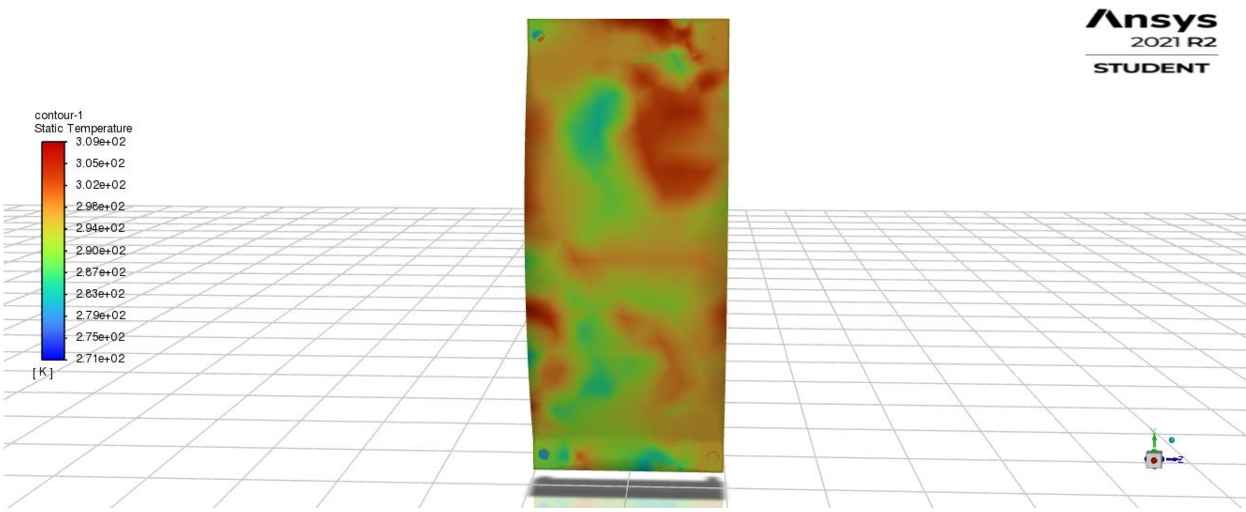

Figure 20: Temperature distribution of hot side plate in simulation software

Similarly, temperature distribution on the cold side of the plate looks like this. The hot inlet (bottom left) and hot outlet (top left) are one of the few outlier zones with a high-temperature gradient. Most of the plate has a low temperature as expected.

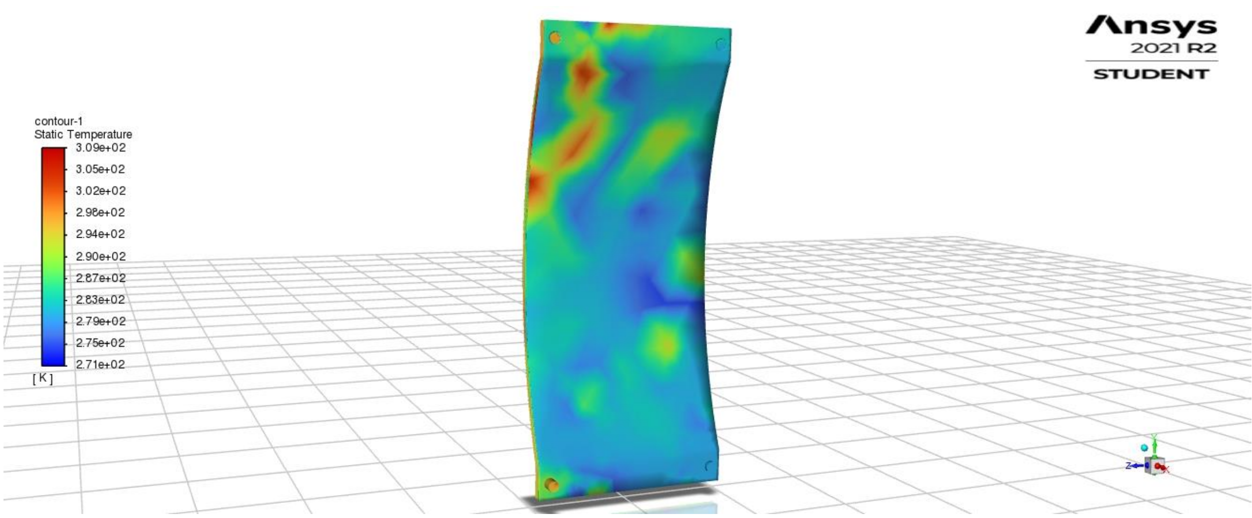

Figure 21: Temperature distribution of cold side plate in simulation software

A side view of the curved plate heat exchanger after the simulation is shown below.

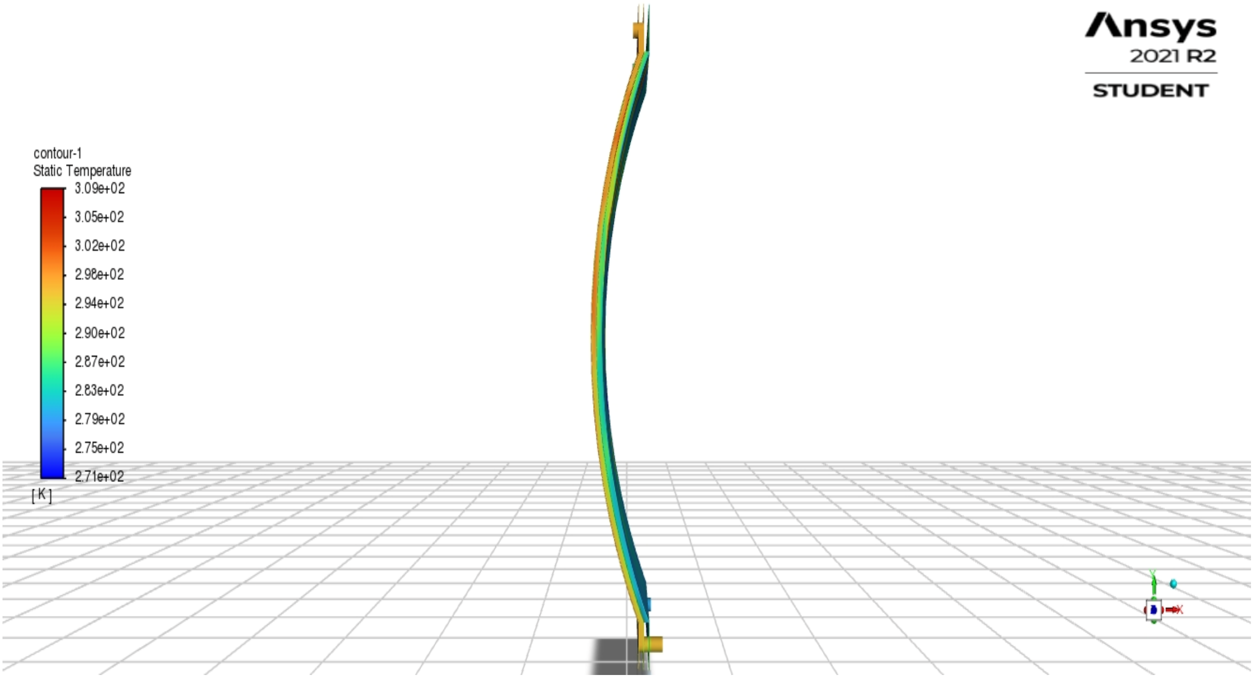

Figure 22: Temperature distribution of side view of the plate in simulation software

8. Insulation economics:

If heat loss to the surroundings is not allowed, a layer of insulation can be added. Here we have a table produced by 3E+ that relates cost savings by preventing heat loss by insulating the heat exchanger. The first column lists the insulation thickness in inch and the fourth column shows the amount of savings that can be made. The table indicates that the optimal insulation thickness is around 2 inch; however, the cost of insulation will ultimately determine it.

Table 01: Insulation thickness and energy calculation

Variable Insulation Thickness	Cost ($/ft/yr)	Heat Loss (kBTU/ft/yr)	Savings ($/ft/yr)
Bare	2.86	441	
0.5	0.81	125	2.05
1.0	0.47	73	2.39
1.5	0.33	51	2.53
2.0	0.26	40	2.60
2.5	0.21	32	2.65
3.0	0.18	27	2.68
3.5	0.15	24	2.71
4.0	0.14	21	2.72
4.5	0.12	19	2.74
5.0	0.11	17	2.75
5.5	0.10	15	2.76
6.0	0.09	14	2.77

9. Advantages of curved plate heat exchanger:

The curved plate HX possesses a few advantages over other types of plate and frame heat exchangers.

Heat transfer rate: The flat plate HX, has a low effective area and induces lower turbulence which results in a low heat transfer rate. . On the other hand, this curved plate design has better heat transfer than a flat plate.

Pressure drop: The spiral plate HX requires high pumping power due to the very high-pressure drop. It is also very heavy, and difficult to clean or repair. On the other hand, this curved plate design has a lower pressure drop than a spiral plate as the flow pass is optimized.

Miscellaneous: Since the plates are connected with gaskets, the number of plates can be changed depending on the heat exchange requirement. It is also easier to clean.

10. Drawbacks:

Our design also has some drawbacks. It will have a high-stress concentration on the corrugated edges, which may ultimately cause fatigue failures due to high-pressure fluid flow. Its shape is rather complex and so it will be difficult to manufacture. Moreover, it has high flow resistance and is also quite expensive.

11. Conclusion:

The curved plate HX model tries to maximize the benefits of spiral plate and flat plate type heat exchangers while reducing their shortcomings. This design can be improved upon through further analysis and simulation. We are hopeful that a curved plate heat exchanger will bring another dimension to the type of plate heat exchanger.

12. References:

Optimal design of plate-and-frame heat exchangers for efficient heat recovery in process industries- Olga P.Arsenyeva, Leonid L.Tovazhnyansky, Petro O.Kapustenko, Gennadiy L.Khavin

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