International Research Journal of Innovations in Engineering and Technology (IRJIET)

ISSN (online): 2581-3048

Volume 2, Issue 3, pp 15-26, May-2018

Enhance the Efficiency of Heat Exchanger with Helical Baffle

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Abstract - In this project work the analyze of heat exchanger been done by ANSYS FLUENT. Shell and tube heat exchanger has been widely used in many industrial applications such an electric power generation, Refrigeration and environmental protection and Chemical Engineering. Baffle is a shell side component of shell and tube heat exchanger. Helical baffle have an effective performance of increasing heat transfer performance. The desirable features of heat exchanger obtain maximum heat transfer coefficient and a lower pressure drop. From the numerical experimentation results of heat exchanger is increased in helical baffle, which in turn decrease the time for cooling the product with lesser time.

Keywords: Heat Transfer Coefficient, Helical Baffle, Fluent Ansys.

I. INTRODUCTION

A Heat exchanger is a device used for efficient heat transfer from one fluid to other fluid a typical heat exchanger is shell and tube heat exchanger .They consist of series of finned tubes in which one of the fluid run over the tube to be heated or cooled. Advantage of configuration of shell and tube heat exchanger provide larger surface area with smaller shape .This type of heat exchanger is good mechanical layout and good for pressurized operation .Shell and tube heat exchanger is easy to clean.

The helical baffle heat exchanger is otherwise known as a Helix changer solution that removes many of deficiencies of other heat exchanger .Helical flow provide the necessary characteristics to reduce flow dispersion .The shell side flow configuration offers a very high conversion of pressure drop to heat transfer. The major drawback of shell and tube heat exchanger first it cause a larger pressure drop secondly it is a bit costly for initial process but it is very economical for later uses .The helical baffle depend upon the helix angle which determines the pressure drop on shell side.

In this project work the work done is on analysis of helical baffle in a heat exchanger. They show the higher heat transfer and lower pressure drop in a helical baffle. A virtual model of helical baffles was created in Solid works and analyzed in a fluent ansys. Based on the performance of parameters such as pressure drop heat transfer coefficient, baffle spacing and pitch angle we will select the best one in Baffles.

II. MAJOR FACTORS AFFECTING SHELL AND TUBE HEAT EXCHANGER PERFORMANCE

The major factors affecting the performance of shell and tube exchanger are turbulence pressure drop, heat transfer coefficient, fouling ratio of flow rates on the tube side to shell side, length of heat exchanger and type of baffles.

The ratio of heat transfer coefficient to pressure drop increases as length of heat exchanger decreases .Therefore more turbulence, lower pressure drop, higher heat transfer coefficient as well as less fouling.

III. COMPARISON BETWWEN HELICAL BAFFLES AND SEGMENTAL BAFFLES

Helical baffles as compared to segmental baffles, gives a better performance. If segmental and helical

ISSN (online): 2581-3048

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baffles are compared for use in shell tube heat exchanger, helical baffle serve as a more promising technology because of having less shell-side pressure drop, better heat transfer coefficient per unit drop pressure.

Helical baffles give spiral flow, which eliminates leakage dead zones and causes a decrease in fouling as well. Although helical baffles require high capital investment. But they are economical for long term usage. The effectiveness of heat exchangers with two layers will give more effectiveness than single layered helical baffle.

Figure-1: Helical baffles

IV. DESIGN OPTIMATION OF BAFFLE

a) Baffle Inclination Angle

Experimented in different helical angles (20^{O}, 30^{O}, 40^{O} and 50^{O}). but the helical baffles with an inclination angle of 40^{O} gives higher heat transfer rate for same pressure drop .we also increased the baffle inclination angle up to 45^{O} and performance decreased with the angle greater than 45^{O}.So it comes to the conclusion that prandtl number is large enough the heat exchanger with smaller baffle tilt comes out to be the optimal choice.

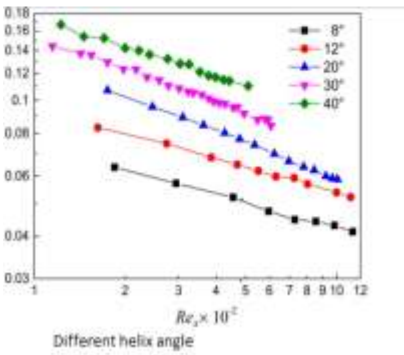

Figure-2: Baffle Inclination Angle

b) Baffle Spacing

With increasing baffle space, pressure gradient decreases. Increasing the baffle space at same mass flow rate reduces heat transfer coefficient. While increasing the baffle spacing at same pressure drop increases heat transfer coefficient. Baffle spacing also depends upon the flow process. If pressure drop is important factor, longer baffle spacing is required. Which lowers pressure drop and reduces the power consumption as well? So there is no precise criterion for selection of optimum baffle spacing. Although baffle spacing is very important factor which affects the capital as well as operating cost of heat exchanger.

c) Baffle Shape

Baffle shape is one of the prominent factors affecting the performance of shell tube heat exchanger. From the experiment of three physical models with baffles of different shapes i.e. trisection and quadrant sector, quadrant showed better comprehensive heat transfer than tri-sectional in helical shell and tube heat.

V. CALCULATION

Heat exchanger efficiency calculation:

Shell and tube heat Exchanger is designed by trial and error calculations. The main steps of design following the Kern method.

Given data:

Heat Capacity, Q = 9,35,557 Kcal /hour = 1122.66KW

Tube side: (6 passes)

Mass flow rate, m_w= 142000 Kg /hour = 39.44 Kg /s

Inlet temp, $t_i = 32\ ^oC$

Outlet temp, $t_o = 39\ ^oC$

Design temp, $\Delta t = 100\ ^0 C$

Design pressure, $\Delta p = 2\ Kg / cm^2$

Length, l = 3 m

Shell Side:

Mass flow rate, m_h =57960 kg/hour = 16.1 kg/s

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Inlet temp, $T_i = 90\ ^0C$

Outlet temp, $T_0 = 50\ ^0C$

Design temp, $\Delta T_s = 120\ ^0C$

Design pressure, $\Delta P_s = 5\ kg/cm^2$

Shell outer diameter, $D_s = 0.7$ m

-To find efficiency

$\eta = Q_{actual}/Q_{optimum}$

$Q_{optimum} = U_{op} \times Area \times (\Delta T_{AHD})$

$\Delta T_{AHD} = (T_I + T_O/2) - (t_o + t_i/2)$

Calculate heat transfer area (A) required

Heat transfer area , A = no of tubes × outer tube area

$= n \times \pi \times d_o \times l$

$= 420 \times \pi \times 0.7 \times 3$

$= 2770.884\ m^2$

Assuming tube of 1inch outer diameter and 1 ¼ inch square pitch of fixed plate heat Exchanger with n=420 tubes and wall thickness is measured in terms of BWG (Birmingham wire gauge).

No of tubes is taken as 420 as the shell diameter is 0.7 m i.e. 27.6 inches and as each tube is taken of 3/4-inch outer diameter so to accommodate the tubes inside the shell we take only 420 tubes so that we can ensure that 3 meter tubes fit inside the shell of 4.3 m, and the rest area or portion is provided for the hot thermo medium to flow over the tubes.

$d_o = 3/4$inch = 0.01905 m
$P_t = 1\ ¼$ inch = 0.03175 m
N = 420
Calculate tube side fluid velocity:

$$u = \frac{4m(n_p/n_t)}{\pi \rho d_i^2}$$

n_p = no of passes = 6

n=420

m= 39.44 kg/s

ρ = *density of water at* $(t_o + t_i/2)$ i.e. 35.5^0C = 994 kg /m^3

d_i = inner diameter of tube

$= d_0 - 14BWG$

14BWG= 0.083 inch = 0.0021082 m

=0.01905 – 0.0021082

= 0.01694 m

$$U = \frac{4 \times 39.44 \times (6/420)}{\pi \times 994 \times 0.01694^2}$$

U= 0.0426 m/s

Reynolds number:

$$Re = \frac{4\dot{m}(n_p/n_t)}{\pi d_i \mu} \geq 10^4$$

μ = dynamic viscosity of water at 35.5^0 C = 0.000710 kg /m-s

$$R_e = \frac{4 \times 39.44 \times (6/420)}{\pi \times 0.01694 \times 0.000710}$$

$R_e = 59645.43 \geq 10^4$
So, by this assumption of the tube diameter is correct.

Determine inside shell diameter (D_s) that can accommodate the calculated number of tubes n_t .

Select the type of Baffle (segmental, doughnut etc.), its size (i.e. percentage cut, 25% Baffles are widely used), spacing (B) and number.

Now,

Shell inner diameter, D_s= 0.7 m

P_T = 3.175 cm = 0.03175 m

Assuming segmented Baffle, with 25% cut,

Baffle spacing, B = 0.5 × D_s = 0.5 × 0.7

= 0.35 m

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Determine the tube side film heat transfer coefficient (h_i) using the suitable form of Sieder-Tate equation in laminar and turbulent flow regimes. Estimate the shell-side film heat transfer coefficient (h_o) from:

$$j_H = \frac{hoDe}{k}\left(\frac{c\mu}{k}\right)^{-1/3}\left(\frac{\mu}{\mu w}\right)^{-0.4}$$

Consider $\frac{\mu}{\mu w} = 1.0$

Tube material = copper

K= thermal conductivity of the tube material = 400 W/m-k

μ = dynamic viscosity of water at 35.5^0 C = 0.000710 kg /m-s

C_w= specific heat of water at 35.5^0 C = 4.18 KJ/Kg = 4180 W/s

D_o= 0.01694 m

Variation of j h -factor with the modified Reynolds number

$jh= 0.023 \times (Re)^{-0.2}$

$jh= = 0.023 \times (59645.43)^{-0.2}$

$jh= 2.550 \times 10^{-3}$

$$j_H = \frac{h_o D_e}{k}\left(\frac{c\mu}{k}\right)^{-\frac{1}{3}}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

You may consider, $\frac{\mu}{\mu_w} = 1.0$

$2.21x\ 10^{-3}$ =(h_o x 0.01694 x $(4180 \text{ x } 0.0710)^{-1/3}$) / 400 x $(400)^{-1/3}$

h_o= 10.178 W/m-k

D_e = Equivalent diameter of shell side

$$D_e = \frac{4\left(P_T^{\,2} - \frac{\pi}{4} d_o^{\,2}\right)}{\pi d_o}$$

d_o = 3/4 inch = 1.905 cm = 0. 01905 m

P_t = 1 ¼ inch = 3.175 cm = 0. 03175 m

$$D_e = \frac{4(0.03175^2 - \pi/4 \times 0.0254^2)}{\pi \times 0.01905}$$

D_e = 0.04832 m

Shell side cross flow area,

$$a_s = \frac{CBD_S}{P_T}$$

C= Tube Clearance

$C = P_T - d_o$

C = 0.03175 - 0.01905

C= 0.0127

Baffles spacing, B= 0.5 D_s

B = 0.35 m

$$a_s = \frac{0.0127 \times 0.35 \times 0.7}{0.03175}$$

a_s = 0.098 m^2

Mass velocity,

$$G_S = \frac{m_g}{a_s}$$

m_g = mass flow rate of shell side = m_h=16.1 kg/s

$$G_s = \frac{16.1}{0.098}$$

G_s =164.2857 kg / m^2 s

Reynolds number

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$$\mathrm{Re} = \frac{D_e G_S}{\mu_g}$$

D_e = 0.2610 m

G_s = 164.2857 kg / m^2 s

μ_g= dynamic viscosity of water at 35.5^0 C = 0.0007191 kg/m-s

R_e = 0.261 x 164.2857 /0.0007191

R_e = 59628.101

Variation of j h -factor with the modified Reynolds number

jh= 0.023 × $(R_e)^{-0.2}$

jh= 0.023 × $(59628.101)^{-0.2}$

jh= 2.5505 × 10^{-3}

Now for the shell side,

$$j_H = \frac{h_o D_e}{k_g}\left(\frac{\mu_g C_g}{k_g}\right)^{-1/3}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

$\frac{\mu}{\mu_w}$ = 1 is considered for the shell side fluid

Shell material = Carbon steel

K= thermal conductivity of the tube material = 36 W/m-k

μ = dynamic viscosity of water at 35.5^0 C = 0.00806 kg /m-s

C_w= specific heat of water at 35.5^0 C = 3.93 KJ/Kg = 3930 W/s

D_e = 0.261 m

jh= 2.5505× 10^{-3}

$$2.5505\times10^{-3} = \frac{h_o \times 0.261 \times (3930 \times 0.00806)^{-1/3}}{36\times (36)^{-1/3}}$$

h_o = 0.3371W/m^{2}k

Overall heat transfer co-efficient $U_{O\ cal}$

Fouling factor, *Rdg* =0.02 h ft2 °F Btu-1 for thermic oil and *Rdk* = 0.03 h ft2 °F Btu-1 for water is taken for this service.

$$U_{o.cal} = \left[\frac{1}{h_o} + R_{dg} + \frac{A_0}{A_i}\left(\frac{d_0 - d_i}{2k_w}\right) + \frac{A_0}{A_i}\left(\frac{1}{h_i}\right) + \frac{A_0}{A_i}R_{dk}\right]^{-1}$$

$U_{O\ cal}$ = (1/5.5 + 0.02 + 3.66 x10^{-5}+ 2.36 × 10^{-6}+ 0.03595)$^{-1}$

$U_{O\ cal}$ =4.205Kw/m^{2}k

$Q_{optimum}$ = U_{op} ×Area × (ΔT_{AHD})

$Q_{optimum}$ = 4.205 × 11.17 × 35.5

$Q_{optimum}$ = 1667.22 KW

Heat Exchanger Efficiency

η= $Q_{actual}/Q_{optimum}$

η= 1122.66 / 1667.22

η= 67.33

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ANSYS ANALYSIS

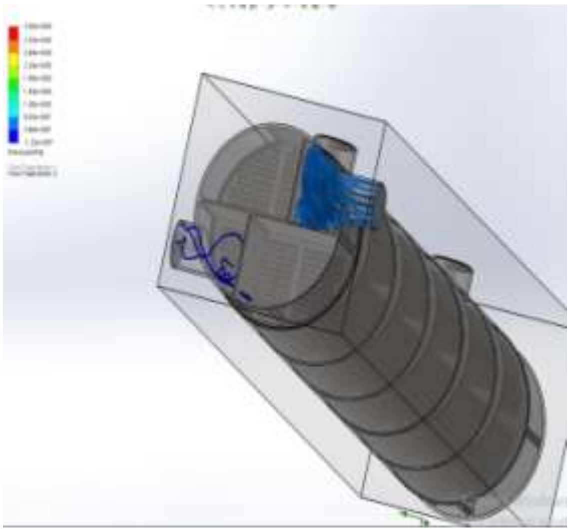

Figure-3: Helical baffle heat exchanger showing inlet and outlet water flow

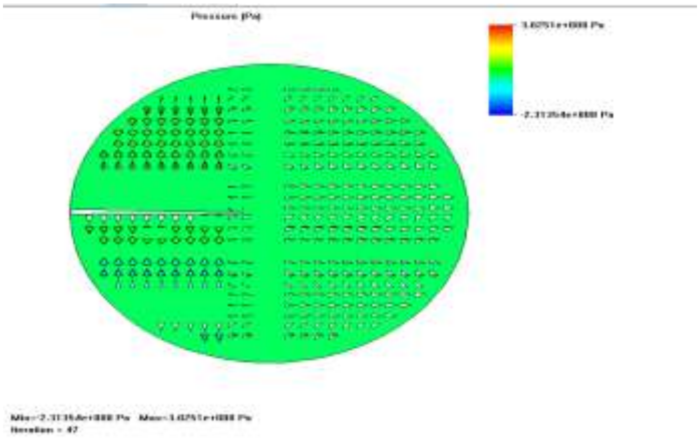

Figure-4: Pressure in tubes and shell side pressure in 47 iteration

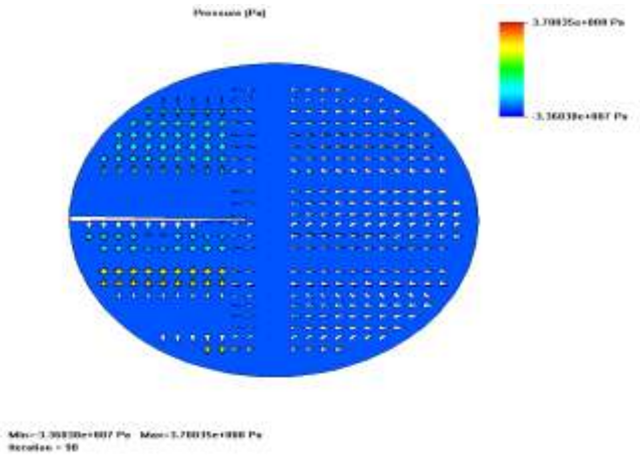

Figure-5: Pressure in tube and shell side pressure in 90 iteration

INPUT DATA

Initial Mesh Settings

Automatic initial mesh: On

Result resolution level: 3

Advanced narrow channel refinement: Off

Refinement in solid region: Off

Geometry Resolution

Evaluation of minimum gap size: Automatic

Evaluation of minimum wall thickness: Automatic

Computational Domain

Size

X min	-1.521 m
X max	-0.705 m
Y min	0.783 m
Y max	1.684 m
Z min	0.723 m
Z max	4.351 m

Boundary Conditions

2D plane flow	None
At X min	Default
At X max	Default
At Y min	Default
At Y max	Default
At Z min	Default
At Z max	Default

Physical Features

Heat conduction in solids: On

Heat conduction in solids only: Off

Radiation: Off

Time dependent: Off

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Gravitational effects: Off

Rotation: Off

Flow type: Laminar and turbulent

Cavitation: Off

High Mach number flow: Off

Default roughness: 0 micrometer

Default outer wall condition: Adiabatic wall

Initial Conditions

Thermodynamic parameters	Static Pressure: 101325.00 Pa Temperature: 293.20 K
Velocity parameters	Velocity vector Velocity in X direction: 0 m/s Velocity in Y direction: 0 m/s Velocity in Z direction: 0 m/s
Solid parameters	Default material: Steel (Mild) Initial solid temperature: 293.20 K
Turbulence parameters	

Material Settings

Fluids

Water

Olive oil

Solids

Steel (Mild)

Fluid Subdomains

Fluid Subdomain 1

Fluids	Water
Faces	Face<1>
Coordinate system	Face Coordinate System
Reference axis	X
Thermodynamic Parameters	Static Pressure: 101325.00 Pa Temperature: 293.20 K
Velocity Parameters	Velocity in X direction: 0 m/s Velocity in Y direction: 0 m/s Velocity in Z direction: 0 m/s
Turbulence parameters type:	Turbulence intensity and length
Intensity	2.00 %
Length	0.008 m
Flow type	Laminar and Turbulent
Cavitation	Off

Fluid Subdomain 2

Fluids	Servo therm oil
Faces	Face<1>
Coordinate system	Face Coordinate System
Reference axis	X
Thermodynamic Parameters	Static Pressure: 101325.00 Pa Temperature: 293.20 K
Velocity Parameters	Velocity in X direction: 0 m/s Velocity in Y direction: 0 m/s Velocity in Z direction: 0 m/s
Flow type	Laminar Only

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Boundary Conditions

Inlet Velocity 1

Type	Inlet Velocity
Faces	Face<3>@Part2-1
	-solid1
Coordinate system	Face Coordinate System
Reference axis	X
Flow parameters	Flow vectors direction: Normal to face Velocity normal to face: 0.042 m/s Fully developed flow: No
Thermodynamic parameters	Temperature: 305.00 K
Turbulence parameters	Boundary layer parameters
Boundary layer type: Turbulent	

Environment Pressure 2

Type	Environment Pressure
Faces	Face<4>@Part2-2-solid1
Coordinate system	Face Coordinate System
Reference axis	X
Thermodynamic parameters	Environment pressure: 101325.00 Pa Temperature: 312.00 K
Turbulence parameters	Boundary layer parameters

Inlet Mass Flow 2

Type	Inlet Mass Flow
Faces	Face<5>@helical baffleinlet-1-solid3
Coordinate system	Face Coordinate System
Reference axis	X
Flow parameters	Flow vectors direction: Normal to face Mass flow rate: 0.2040 kg/s Inlet profile: 0
Thermodynamic parameters	Temperature: 363.00 K
Turbulence parameters	Boundary layer parameters
Boundary layer type: Turbulent	

Environment Pressure 1

Type	Environment Pressure
Faces	Face<6>@helical baffleoutlet-1-solid3
Coordinate system	Face Coordinate System
Reference axis	X
Thermodynamic parameters	Environment pressure: 101325.00 Pa Temperature: 293.20 K
Turbulence parameters	Boundary layer parameters
Boundary layer type:	

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Turbulent

Calculation Control Options

Finish Conditions

Finish Conditions	If one is satisfied
Maximum travels	4
Goals convergence	Analysis interval: 5.000000e-001

Solver Refinement

Refinement: Disabled

Results Saving

Save before refinement	On

Advanced Control Options

Flow Freezing

Flow freezing strategy	Disabled

VI. RESULTS

General Info

Iterations: 115

CPU time: 1668 s

Log

Mesh generation started	08:42:58 , Apr 04
Mesh generation normally finished	08:50:23 , Apr 04
Preparing data for calculation	08:50:25 , Apr 04
Calculation started 0	08:52:40 , Apr 04
Calculation has converged since the following criteria are satisfied: 114	09:18:19 , Apr 04
Goals are converged 114	
Calculation finished 115	09:20:47 , Apr 04

Warnings: A vortex crosses the pressure opening Boundary Condition : Environment Pressure 2 ; Inlet flow/outlet flow=0.516151

Calculation Mesh

Basic Mesh Dimensions

Number of cells in X	8
Number of cells in Y	8
Number of cells in Z	32

Number of Cells

Total cells	84361
Fluid cells	10871
Solid cells	9353
Partial cells	64137
Irregular cells	0
Trimmed cells	0

Maximum refinement level: 2

Goals

Name	Unit	Value	Progress	Use in convergence	Delta	Criteria

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Min/Max Table

Name	Minimum	Maximum
Pressure [Pa]	-1.22e+007	3.50e+008
Temperature [K]	305.00	617.98
Density (Fluid) [kg/m^3]	807.83	994.22
Velocity [m/s]	0	84.328
Velocity (X) [m/s]	-5.462	3.865
Velocity (Y) [m/s]	-6.276	6.748
Velocity (Z) [m/s]	-84.303	52.411
Mass Fraction of Water [ ]	1.0000	1.0000
Mass Fraction of Olive oil [ ]	1.0000	1.0000
Volume Fraction of Water [ ]	1.0000	1.0000
Volume Fraction of Olive oil [ ]	1.0000	1.0000
Temperature (Fluid) [K]	305.00	617.98
Temperature (Solid) [K]	315.13	617.98
Density (Solid) [kg/m^3]	7870.00	7870.00
Vorticity [1/s]	0	8431.475
Velocity RRF [m/s]	0	84.328
Velocity RRF (X) [m/s]	-5.462	3.865
Velocity RRF (Y) [m/s]	-6.276	6.748
Velocity RRF (Z) [m/s]	-84.303	52.411
Shear Stress [Pa]	0	14601.17
Relative Pressure [Pa]	-1.23e+007	3.50e+008
Heat Transfer Coefficient [W/m^2/K]	0	4140.628
Surface Heat Flux [W/m^2]	-864334.699	775272.681
Overheat above Melting Temperature [K]	-1358.019	-1055.168

Engineering Database

Solids

Steel (Mild)

Path: Solids Pre-Defined\Alloys

Density: 7870.00 kg/m^3

Specific heat: 472.0 J/(kg*K)

Conductivity type: Isotropic

Thermal conductivity: 51.9000 W/(m*K)

Electrical conductivity: Conductor

Resistivity: 1.7400e-007 Ohm*m

Radiation properties: No

Melting temperature: Yes

Temperature: 1673.15 K

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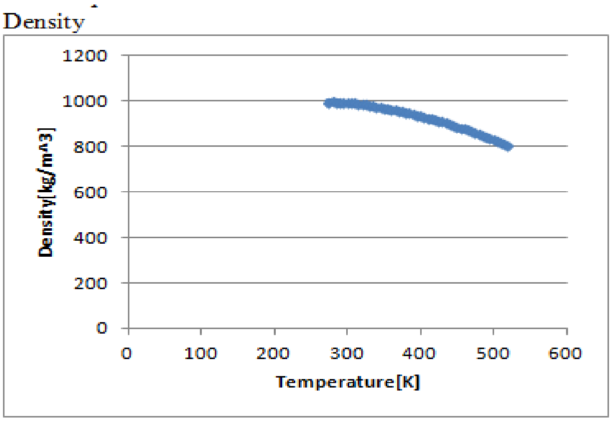
Liquids

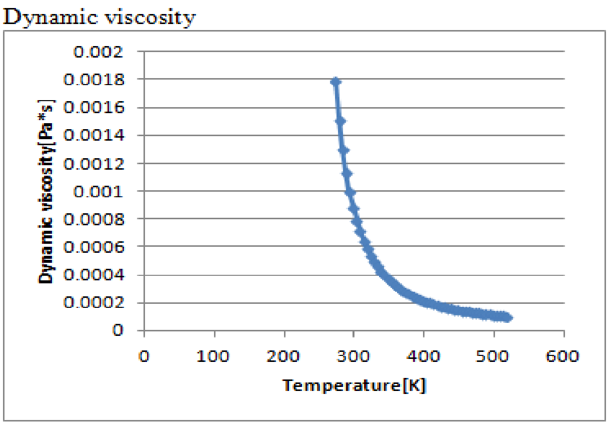
Water

Path: Liquids Pre-Defined

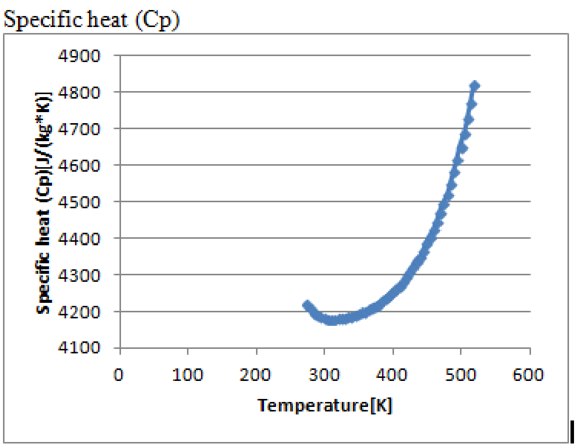
Density:



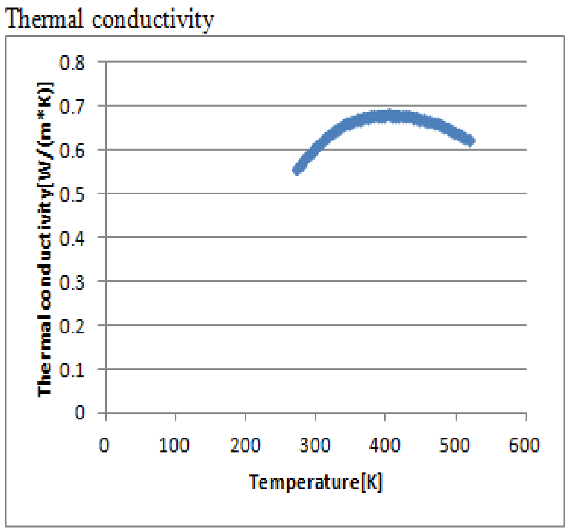
Dynamic viscosity:



Specific heat (Cp):



Thermal conductivity:



Cavitations effect: Yes

Temperature: 0 K

Saturation pressure: 0 Pa

Non-Newtonian/Compressible liquids

Servotherm oil

Path: Non-Newtonian Liquids Pre-Defined

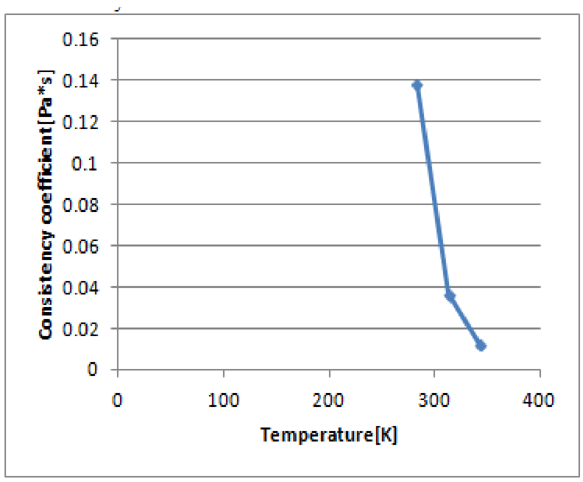
Density: 917.53 kg/m^3

Specific heat: 1971.8 J/(kg*K)

Thermal conductivity: 0.1700 W/(m*K)

Viscosity: Power-law model

Consistency coefficient



Set up maximum viscosity: No

Set up minimum viscosity: No

Power-law index: 1.0000000

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VII. CONCLUSION

Use of Helical baffles in heat exchanger reduces Shell side Pressure drop, pumping cost, weight, fouling etc as compare to segmental baffle for a new installation. The ratio of heat transfer rate is increased in helical baffle as compared to that of segmental baffle heat exchanger. Pressure drop is also lower in helical than segmental baffle heat exchanger. Helical baffle is much higher than the segmental baffle because of reduced by pass effect and reduced shell side fouling. Helical baffle is 3 times higher than segmental baffle.

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How to cite this article:

Sagar Kadu, Monash Mhatre, Aadhityasagar, Augussilvastar, Siddhart Nagi, “Enhance the Efficiency of Heat Exchanger with Helical Baffle”, in *International Research Journal of Innovations in Engineering and Technology (IRJIET)*, Volume 2, Issue 3, pp 15-26, May 2018.

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