Analytical and CFD analysis of Shell and tube heat Exchanger with Segmental and Helical Baffles

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Abstract— **The development and performance optimization of shell and tube heat exchanger is an issue of great challenge and part of emerging nascent technology. The performance optimization would serve a great contribution to placate the inflated operating costs as well as energy crisis. This paper showcases all the empirical results obtained from the real time system analysis in various working conditions. Further it represents comparison for several shell-and-tube heat exchangers with segmental baffles as well as helical baffles with baffle angle parametric variation. The system identification has been carried out by two methods viz. Theoretical analysis and CFD analysis. The combined results with respect to same shell-side flow rate show that, the heat transfer coefficient of the heat exchanger with helical baffles is higher than that of the heat exchanger with segmental baffles while the shell-side pressure drop of the former is even much lower than that of the latter. Further enhancement techniques should be incorporated in order to enhance shell-side heat transfer based on the same flow rate. The comparative analysis of heat transfer coefficient per unit pressure drop shows that the Segmental Baffle Heat exchanger have significant performance advantage over Segmental Baffle Heat exchanger for the same geometrical configurations. The performance enhancement of heat exchanger with helix baffle angle optimization could be considered as an innovation.**

Keywords—*CFD analysis, helical baffles, shell-side flow rate, heat transfer coefficient, pressure drop.(keywords)*

I.INTRODUCTION

A heat exchanger is equipment built for efficient heat transfer from one medium to another. The media may be separated by a solid wall to prevent mixing or they may be in direct contact. They have numerous applications and are widely used in space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, natural gas processing, and sewage treatment.

The performed work is based on the analysis of Shell and tube heat exchanger, that contains two separated fluids at different temperatures flowing through the heat exchanger: one through the tubes (tube side) and the other through the shell around the tubes (shell side). Several design parameters and operating conditions influence the optimal performance of a shell-and-tube heat exchanger.

The baffle configuration is selected on the basis of size, cost, and ability to lend support to the overhung tube bundles. In the presented work helical baffles are considered over segmental baffles for numerous advantages such as:-

- Increased heat transfer rate/ pressure drop ratio.
- Reduced bypass effects.
- Reduced shell side fouling.
- Prevention of flow induced vibration.
- Reduced maintenance

Helical baffles are advantageous because high pressure drop occurs since the segmental baffles make fluid perpendicularly impact the shell wall and the tubes, leading to an increased power load which is overcome by helical baffles.

II. PROBLEM STATEMENT

Comparative Analysis of Shell and Tube Heat Exchangers with Segmental Baffles and Helical Baffle Configurations in reference to Heat Transfer Co-efficient and Pressure Drop using Analytical and CFD analysis and identifying the most suitable Baffle angle Configuration for Industrial Application.

III. LITERATURE REVIEW

Literature review for the present study includes the guidelines which are as follows:-

1. Mustansir Hatim Pancha et al,"Comparative Thermal Performance Analysis of Segmental Baffle Heat Exchanger with Continuous Helical Baffle Heat Exchanger using Kern method"Pub no.ISSN/2248/9622.Pub Date-July-august 2012:- With segmental baffles, most of the overall pressure drop is wasted in changing the direction of flow, while helical baffle focus on better conversion of pressure drop into heat transfer that is, higher Heat transfer co-efficient to Pressure drop ratio. Also the undesirable effects such as dead spots/zones of recirculation causes fouling, high leakage flow and large cross flow, are avoided.
2. Qiuwang Wang et al,"Shell and tube heat exchanger with helical baffles"Pub no.US 2011/0094720.Pub. Date-Apr.28,2011:- Power load present in segmental baffle can be reduced by helical baffles. The invention provides 2 methods of manufacturing of continuous helical baffles. The flow pattern in helical reduce fouling and increase the service life.
3. B.Peng et al, "An Experimental study of shell and tube heat exchanger using continuous helical baffles, "Journal of Heat transfer Volno.-129/1425,October 2007:- Helical baffles prevent the flow induced vibration. The use of continuous helical baffles results in nearly 10% increase in heat transfer coefficient compared with those with the conventional segmental baffles for the same shell side pressure drop.

IV. ANALYTICAL CALCULATIONS

This section represents all the equations and formulae used in designing the shell side, tube side, segmental baffles and helical baffles heat exchanger. The values obtained are listed below and sample calculation for one Configuration (25^o baffle angle) for helical baffle heat exchanger is shown.

Geometrical parameters
Shell
Shell diameter (inner diameter)=92.5mm
Thickness=1.25mm
Tube
Outer Diameter=12.7mm
Length=245mm
Thickness=0.37mm
Pitch= 27.5mm
Segmental baffles
Helical baffle angles- 15^o , 25^o , 35^o , 45^o

Boundary conditions
Ambient air temperature (inlet air temp)=25^oC
Water inlet temp=55^oC
Inlet mass flow rate of air=0.1025kg/sec
Inlet mass flow rate of water=0.052kg/sec

General Calculations
1. Heat Transfer Rate

$$Q = m_a C_{pa} \Delta T_a \qquad ...(1)$$

Q =3714.14 KJ/hr

2. Outlet Temp Of Water

$$Q = m_w C_{pw} \Delta T_w \qquad ...(2)$$

$$Q_{air} = Q_{water} \qquad ...(3)$$

$$Q / m_w C_{pw} = \Delta T_w$$

3. Logarithemic Mean Temp Difference

R=2

$$R = \frac{T_{s1} - T_{s2}}{T_{t2} - T_{t1}} \qquad ...(4)$$

S=0.25

$$S = \frac{T_{t2} - T_{t1}}{T_{s1} - T_{t1}} \qquad ...(5)$$

$$(\Delta T)_{lmtd} = \frac{(T1 - t2) - (T2 - t1)}{\ln\left(\frac{(T1 - t2)}{(T2 - t1)}\right)} \qquad ...(6)$$

$(\Delta T)_{lmtd} = 12.33\ ^0C$

$$\Delta T = F_t * \Delta T_{lmtd} \qquad ...(7)$$

F_t =Correction factor

Heat transfer rate, Outlet temperature of water and LMTD are the parameters which forms the base for the calculations of both shell and tube side primary calculations.

2.1 Calculation for shell side pressure drop [1]	2.2 Calculation for tube side pressure drop [1]
1. Flow area $a_s = \frac{ID \times C' \times B}{144 \times P_T}$ ...(8) a_s =0.02725 ft^2 2.Mass Velocity $G_s = M_s / A_s$ ...(9) G_s =29851.77 lb/hr ft^2 3. Reynold's Number $R_{e_s} = \frac{D_s G_s}{\mu}$ ...(10) R_{e_s} =140486.98 4. Pressure Drop $\Delta P_s = \frac{f G_s^2 D_s [N+1]}{5.22 \times 10^{10} D_e S \phi_s}$ ...(11) ΔP_s =7.31 psi Allowable Pressure drop=0.00731 kpsi	1.Flow Area $a_t = \frac{N_t \times a'_t}{144n}$ ...(12) a_t =0.0042 ft^2 2. Mass velocity $G_t = \frac{M_t}{a_t}$ ...(13) G_t =92909.07 lb/hr ft^2 3. Reynold's Number $R_{e_t} = \frac{D_t G_t}{\mu}$ ...(14) R_{e_t} =2994.58 4.Tube Side Pressure Drop $\Delta P_t = \frac{f G_t^2 L n}{5.22 \times 10^{10} D_t S \phi_t}$ ...(15) ΔP_t =0.0310 psi 5.Return Pressure Drop $\Delta P_r = \frac{4nV^2}{s \times 2g'} \frac{62.5}{144}$ ...(16) ΔP_r =0.00258 psi 6.Total Pressure Drop $\Delta P_T = \Delta P_t + \Delta P_r$ ...(17) ΔP_T =0.03358 psi Allowable Pressure Drop=0.03358 psi

4. Sample baffle calculation for baffle angle 25^0 is shown below:

1. Tube Clearance (C')

$$C' = P_t - D_{ot} \qquad ...(18)$$

C'=0.0148 m

2. Baffle Spacing (Lb)

$$L_b = \pi \cdot D_{is} \cdot \tan \phi \qquad ...(19)$$

L_b=0.1355 m

3. Cross-flow Area (AS)

$$A_S = (D_{is}\,C'\,L_B) / P_t \qquad \ldots(20)$$

$A_S = 0.006745\ m^2$

4. Equivalent Diameter (D_E)

$$D_E = 4\,[\,(P_t^2 - \pi\,.D_{ot}^2 / 4\,) / (\pi\,.D_{ot}) \qquad \ldots(21)$$

$D_E = 0.06312$ m

5. Maximum Velocity (V_{max})

$$V_{\max} = \frac{M_s}{A_s} \qquad \ldots(22)$$

$V_{max} = 12.35$ m/sec

6. Reynolds number (R_e)

$$R_e = (\rho\,.V\max\,.D_E) / \mu \qquad \ldots(23)$$

$R_e = 46198.92$

7. Prandtl number (P_r)

Pr = 0.7038

8. Heat Transfer Co-efficient (α_o)

$$\alpha_o = (0.36\,.K\,.\,Re^{0.55}\,.\,Pr^{0.33}) / \ R.D_E \qquad \ldots(24)$$

$\alpha_o = 50.8764\ W/m^2K$

9. No. of Baffles (N_b)

$$N_b = Ls / (L_b + \Delta S_B) \qquad \ldots(25)$$

$N_b = 2.39 = 3$

10. Pressure Drop (ΔP_S)

$$\Delta P_S = [4.f.M_s^2.D_{is}.(N_b+1)]/(2.\rho.D_E) \qquad \ldots(26)$$

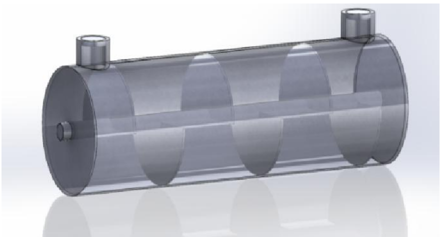
$\Delta P_S = 0.1196$ KPa

V. HEAT EXCHANGER AND BAFFLE CONFIGURATION

The Baffle Configurations used for Theoretical and CFD purpose are listed as follows:

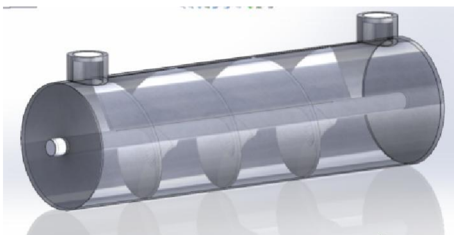
1. Segmental Baffle Heat Exchanger

Segmental Baffle Heat Exchanger is a type of shell and tube heat exchanger which has a Quadrant shaped baffle segments that are arranged at right angle (90^0) to the tube axis in a sequential pattern that guide the shell side flow over the tube bundle.



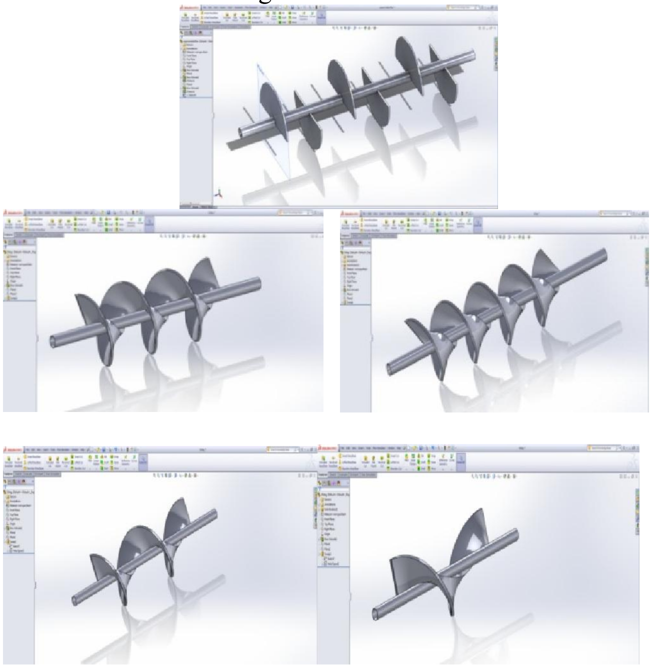
2. Helical Baffle Heat Exchanger

Helical Baffle Heat Exchanger is a type of shell and tube heat exchanger which has a Helical shaped baffle segment which are arranged at Helix angle (15^0,25^0,35^0&45^0) to the tube axis in a sequential pattern that guide the shell side flow over the tube bundle. The visual representation of the Helical Baffle over the tubes is similar to a spring wound around the rod/tube.



3. Baffle configurations:

Various baffle configurations are tested for optimum results and the baffle angle CFD models are as follows:



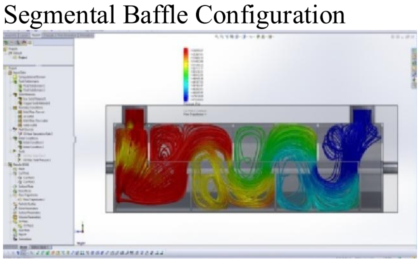
VI. CFD ANALYSIS

Computational fluid dynamics, abbreviated as CFD, is a branch of fluid mechanics which uses numerical methods and algorithms to solve and analyze problems that involve fluid flow. We have used software named SOLIDWORKS 2013for the analysis of the Helical Baffles in the Shell and Tube Heat Exchangers.

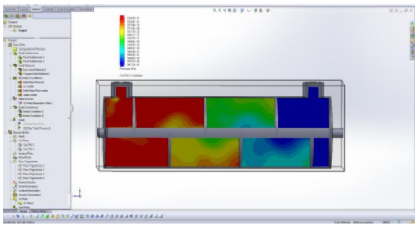
CFD analysis is performed on the developed solid model with the baffle angle 15^o, 25^o, 35^o, 45^o. CFD results for segmental and optimum helical angle (25^o) are shown below.

Pressure analysis:-

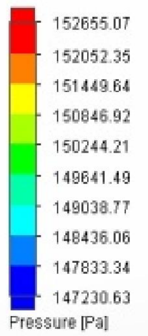
Segmental Baffle Configuration



Pressure Trajectory (segmental baffle)

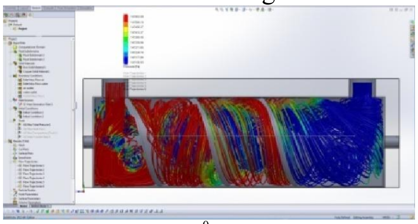


Pressure Contour (segmental baffle)

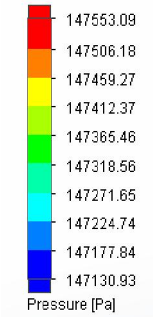


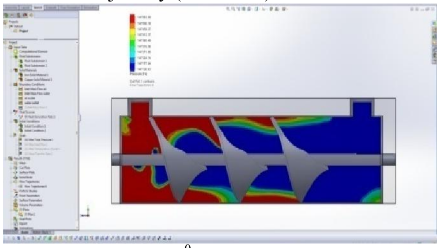
The pressure drop is 152655.07-147230.63 = **5420Pa**

25^0 Helical Baffle Configuration



Pressure Trajectory (25^0 baffle)





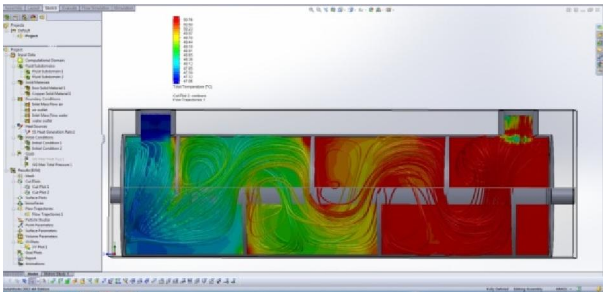
Pressure Contour (25^0 baffle)

The pressure drop is 147553.09-147130.93 = **423Pa**

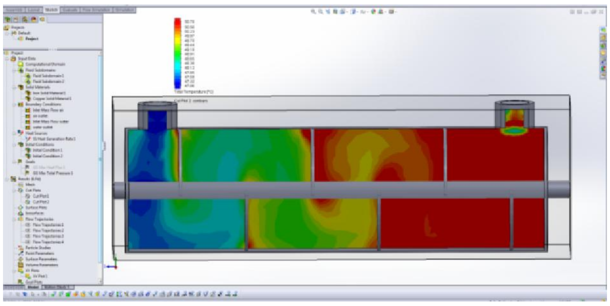
Thermal Analysis:-

Most effective temperature rise is studied which also contributes to the selection of the optimum baffle angle configuration from the following analysis.

Segmental Baffle Configuration

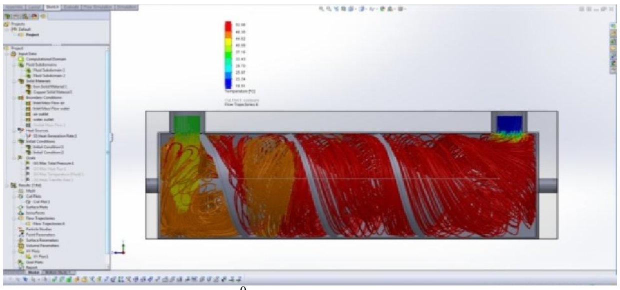


Temperature Trajectory (segmental baffle)

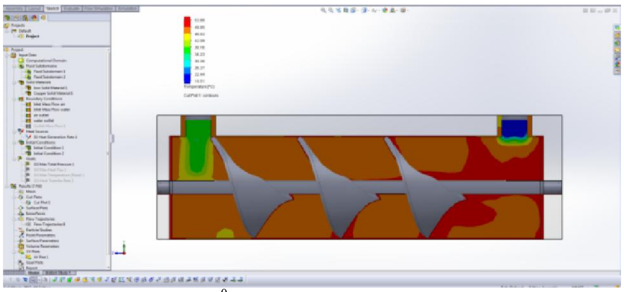


Temperature Contour (segmental baffle)

25^0 Helical Baffle Configuration



Temperature Trajectory (25^0 baffle)



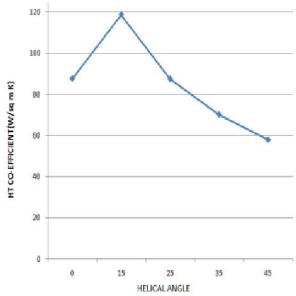
Temperature Contour (25^0 baffle)

VII. RESULTS AND VALIDATION

This section combines the results obtained from theoretical analysis and CFD analysis. The results are then compared by plotting graphs and final conclusions have been found which are shown below.

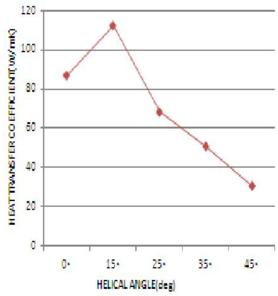
Comparative Results for Heat Transfer Coefficient

Theoretical Results



Graph1Heat Transfer Co-efficient vs. Helical Angle(Theoretical)

CFD Results



Graph2Heat Transfer Co-efficient vs. Helical Angle(CFD)

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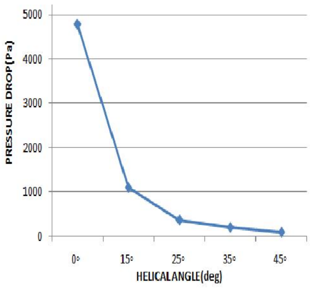
TABLE 1: COMPARISON OF HT COEFFICIENT

HELIX ANGLE	H.T. COEFFICIENT (W/sq m. K) Theoretical	H.T. COEFFICIENT (W/sq m. K) CFD	NUMBER OF REVOLUTIONS
0◦ (Segmental baffle)	87.65	86.993	6
15◦	118.77	112.471	4
25◦	87.47	68.250	3
35◦	70.15	50.827	2
45◦	57.906	30.548	1

The above result give us a clear idea that the Helical baffle heat exchanger has far more better Heat transfer coefficient than the conventional segmental Heat Exchanger.

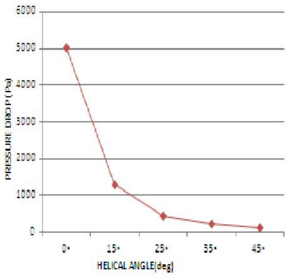
Comparative Results for Pressure Drop.

Theoretical Results



Graph3Pressure Drop vs. Helical Angle(Theoretical)

CFD Results



Graph4Pressure Drop vs. Helical Angle(CFD)

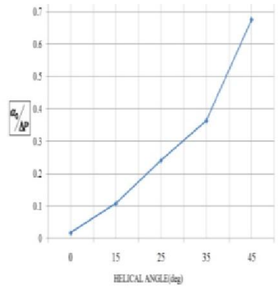
TABLE 2: COMPARISON OF PRESSURE DROP

HELIX ANGLE	PRESSURE DROP ΔP_S (Pa) Theoretical	PRESSURE DROP ΔP_S (Pa) CFD	NUMBER OF REVOLUTIONS
0◦(Segmental baffle)	4780	5420	6
15◦	1095.73	1270	4
25◦	360.84	423	3
35◦	192.8	204	2
45◦	85.51	98.72	1

The above result indicates that the pressure drop ΔPs in a helical baffle heat exchanger is appreciably lesser as compared to Segmental baffle heat Exchanger due to increased cross-flow area resulting in lesser mass flux throughout the shell, and also different baffle geometry.[Graph 3 and 4]

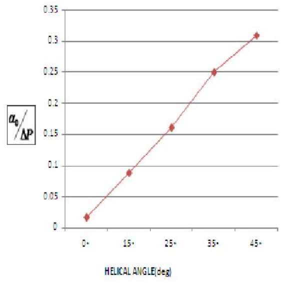
Comparative Results for Heat Transfer per unit pressure drop

Theoretical Results



Graph5 The ratio of Heat Transfer co-efficient per unit pressure drop Vs Helix angle(Theoretical)

CFD Results



Graph6 The ratio of Heat Transfer co-efficient per unit pressure drop Vs Helix angle(CFD)

TABLE 3:COMPARISON OF HEAT TRANSFER PER UNIT PRESSURE DROP

HELIX ANGLE	$\alpha_o/\Delta P_S$ Theoretical	$\alpha_o/\Delta P_S$ CFD	NUMBER OF REVOLUTIONS
0◦ (Segmental baffle)	0.0183	0.0174	6
15◦	0.1084	0.0886	4
25◦	0.2424	0.1613	3
35◦	0.3638	0.2500	2
45◦	0.6771	0.3090	1

The ratio of Heat Transfer co-efficient per unit pressure drop for helical baffle is higher as compared to segmental baffle heat exchanger. [Graph 5 and 6]

VIII. CONCLUSION

In this project, numerical simulations of Helical baffle heat exchanger with different helix angles are performed to reveal the effects of baffle helical angle on the heat transfer and pressure drop characteristics. This provides an optimal helix angle for the required range of heat transfer coefficient and available pumping power. The major findings are summarized as follows:

1. In the present work, an attempt has been made to modify the existing Kern method which is originally used for Segmental baffle heat exchangers, so as to use it for continuous helical baffle heat exchanger. The Kern method available in the literature is only for the conventional segmental baffle heat exchanger, but the modified formula used to approximate the thermal performance of Helical baffle Heat Exchangers give us a clear idea of their efficiency and effectiveness.

2. By **Theoretical** analysis the maximum heat transfer co-efficient obtained for segmental baffles is **87.65 W/sqmK** and for 15^o helical baffles the **maximum** obtained heat transfer coefficient is **118.77 W/sqmK** and by using **CFD** analysis the **maximum** heat transfer co-efficient obtained for segmental

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baffles is **86.993 W/sqmK** and **112.471 W/sqmK** for 15° helical baffles. By **Theoretical** analysis the pressure drop obtained for segmental baffles is **4780 Pa** and for 45° helical baffles the **minimum** pressure drop obtained is **85.51 Pa** and by using **CFD** analysis pressure drop is **5000 Pa** for segmental baffles and **minimum** pressure drop for 45° helical baffles being **98.72 Pa** respectively.

The values obtained by analytical writing and CFD analysis are in good agreement.

3. The optimum helical angle can be determined by comparing the values obtained from graph of Heat Transfer per unit pressure drop. With decrease in helical angle, though there is increase in heat transfer coefficient but this also leads to increase in the shell side pressure drop. Hence, **25^0 is considered to be the optimum helical angle for Industrial purpose heat exchangers.**

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