Comparative Thermal Performance Analysis Of Segmental Baffle Heat Exchanger with Continuous Helical Baffle Heat Exchanger using Kern method

Professor Sunilkumar Shinde *, Mustansir Hatim Pancha **
*(Department of Mechanical Engineering, Vishwakarma Institute Of Technology. Pune, India.)
** (M.Tech, Department of Heat Power Engg. Vishwakarma Institute Of Technology, Pune, India.)

ABSTRACT

Heat exchangers are one of the most important heat transfer apparatus that find its use in industries like oil refining, chemical engineering, electric power generation etc. Shell-and-tube type of heat exchangers (STHXs) have been commonly and most effectively used in Industries over the years. This paper analyses the conventional heat exchanger thermally using the Kern method. This is a proven method & has been verified by researchers. The paper also consists of thermal analysis of a heat exchanger with helical baffles using the Kern method, which has been modified to approximate results for different helical angles.

The results obtained give us a clear idea that the ratio of heat transfer coefficient per unit pressure drop is maximum in helical baffle heat exchanger, as compared to segmental baffle heat exchanger. The Helical baffle heat exchanger eliminates principle shortcomings in Conventional Heat Exchangers due to shell side zigzag flow, induced by Segmental baffle arrangement. The flow pattern in the shell side of the continuous helical baffle heat exchanger is rotational & helical due to the geometry of continuous helical baffles. This flow pattern results in significant increase in heat transfer coefficient, however the pressure drop reduces significantly in the helical baffle heat exchanger.

Keywords - **Kern method, helical baffle heat exchanger, helix angle, heat transfer coefficient, pressure drop, shell & tube heat exchanger.**

I. INTRODUCTION

Conventional shell and tube heat exchangers with segmental baffles have low heat transfer co-efficient due to the segmental baffle arrangement causing high leakage flow bypassing the heat transfer surface and high pressure drop that poses a big problem for industries as the pumping costs increases.

The hydrodynamic studies testing the heat transfer (mean temperature difference) and the pressure drop; with the help of research facilities and industrial equipment have shown much better performance of helical baffle heat exchangers as compared to the conventional ones. This results in relatively high value of shell side heat transfer coefficient, low pressure drop, and low shell side fouling. [1]

II. DESIRABLE FEATURES OF A HEAT EXCHANGER

The desirable features of a heat exchanger would be to obtain maximum heat transfer to Pressure drop ratio at least possible operating costs without comprising the reliability.

2.1 Higher heat transfer co-efficient and larger heat transfer area

A high heat transfer coefficient can be obtained by using heat transfer surfaces, which promote local turbulence for single phase flow or have some special features for two phase flow. Heat transfer area can be increased by using larger exchangers, but the more cost effective way is to use a heat exchanger having a large area density per unit exchanger volume, maintaining the Integrity of the Specifications.

2.2 Lower Pressure drop

Use of segmental baffles in a Heat Exchanger result in high pressure drop which is undesirable as pumping costs are directly proportional to the pressure drop within a Heat Exchanger. Hence, lower pressure drop means lower operating and capital costs.

III. DEVELOPMENTS IN SHELL AND TUBE EXCHANGER

The developments for shell and tube exchangers focus on better conversion of pressure drop into heat transfer i.e higher Heat transfer co-efficient to Pressure drop ratio, by improving the conventional baffle design. With single segmental baffles, most of the overall pressure drop is wasted in changing the direction of flow. This kind of baffle arrangement also leads to more grievous undesirable effects such as dead spots or zones of recirculation which can cause increased fouling, high leakage flow that bypasses the heat transfer surface giving rise to lesser heat transfer co-efficient, and large cross flow. The cross flow not only reduces the mean temperature difference but can also cause potentially damaging tube vibration [2].

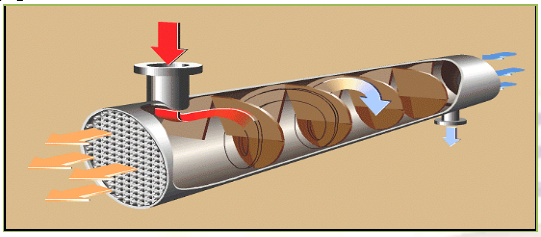


Fig. 1 Helical Baffle Heat Exchanger

3.1 Helical baffle Heat Exchanger

The baffles are of primary importance in improving mixing levels and consequently enhancing heat transfer of shell-and-tube heat exchangers. However, the segmental baffles have some adverse effects such as large back mixing, fouling, high leakage flow, and large cross flow, but the main shortcomings of segmental baffle design remain [3]

Compared to the conventional segmental baffled shell and tube exchanger Helixchanger offers the following general advantages. [4]

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

Figure 2 Helical baffle Heat Exchanger

3.2 Research aspects

Research on the helixchanger has forced on two principle areas.

- Hydrodynamic studies and experimentation on the shell side of the Heat Exchanger
- Heat transfer co-efficient and pressure drop studies on small scale and full industrial scale equipment.

3.3 Design aspects

An optimal design of a helical baffle arrangement depends largely on the operating conditions of the heat exchanger and can be accomplished by appropriate design of helix angle, baffle overlapping, and tube layout.

The original Kern method is an attempt to co-relate data for standard exchangers by a simple equation analogous to equations for flow in tubes. However, this method is restricted to a fixed baffle cut of 25% and cannot adequately account for baffle-to-shell and tube-to-baffle leakages. Nevertheless, although the Kern equation is not particularly accurate, it does allow a very simple and rapid calculation of shell side co-efficients and pressure drop to be carried out and has been successfully used since its inception. [5]

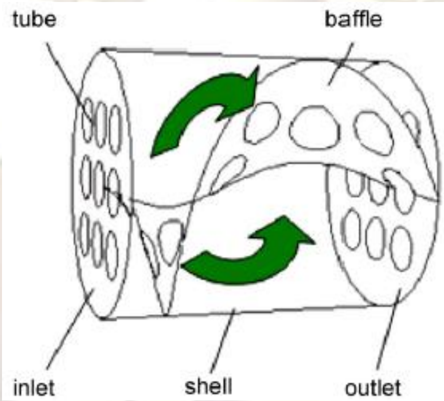


Figure 3 Helical Baffle Heat Exchanger pitch

3.4 Important Parameters

- Pressure Drop (ΔPS)
- Helical Baffle pitch angle (ϕ)
- Baffle spacing (LB)
- Equivalent Diameter (DE)
- Heat transfer coefficient (αo)

In designing a helical Baffle Heat Exchanger, the pitch angle, baffle's arrangement, and space between the two baffles with the same position are some of the important parameters. Baffle pitch angle (ɸ) is the angle between the flow and perpendicular surface on exchanger axis and LB is the space between two corresponding baffles with the same position.

Optimum design of helical baffle heat exchangers is dependent on the operating conditions of the heat exchanger. Consideration of proper design of Baffle pitch angle, overlapping of baffles and tube's layout results in the optimization of the Heat Exchanger Design. In segmental heat exchangers, changing the baffle space and baffle cut can create wide range of flow velocities while changing the helix pitch angle in helical baffle system does the same. Also, the overlapping of helical baffles significantly affects the shell side flow pattern.

IV. THERMAL ANALYSIS OF SEGMENTAL BAFFLE HEAT EXCHANGER & HELICAL BAFFLE HEAT EXCHANGER

In the current paper, thermal analysis has been carried out using the Kern's method. The thermal parameters necessary to determine the performance of the Heat Exchanger have been calculated for Segmental baffle heat Exchanger following the Kern's method, and suitable modifications made to the method then allow us to apply it for the helical baffle Heat Exchanger which is the subject area of interest. Also, the comparative analysis, between the thermal parameters of the two Heat exchangers has been carried out, that clearly indicates the advantages and disadvantages of the two Heat Exchangers.

4.1 Heat Exchanger Data at the shell side

Table 1. Input data – Shell Side

S. No.	Quantity	Symbol	Value
1.	Shell side fluid		Water
2.	Volume flow rate	$(\dot{Q}_s)$	40 to 80 lpm.
3.	Shell side Mass flow rate	$(\dot{m}_s)$	0.67 to 1.33 kg/sec
4.	Shell ID	(D_{is})	0.153 m
5.	Shell length	(L_s)	1.123 m
6.	Tube pitch	(P_t)	0.0225 m
7.	No. of passes		1
8.	Baffle cut		25%
9.	Baffle pitch	(L_B)	0.060 m
10.	Shell side nozzle ID		0.023 m
11.	Mean Bulk Temperature	(MBT)	30 ˚C
12.	No. of baffles	(N_b)	17
13.	Shell side Mass velocity / mass flux	$(\dot{M}_F)$	kg / $(m^2 s)$

4.2 Heat Exchanger data at the tube side

Table 2. Input data – Tube side

S. No.	Quantity	Symbol	Value
1.	Tube side fluid		Water
2.	Volume flow rate	$(\dot{Q}_t)$	40 to 80 lpm.
3.	Tube side Mass flow rate	$(\dot{m}_t)$	0.67 - 1.33 kg/sec
4.	Tube OD	(D_{ot})	0.153 m
5.	Tube thickness		1.123 m
6.	Number of Tubes		0.0225 m
7.	Tube side nozzle ID		1
8.	Mean Bulk Temperature	(MBT)	30 ˚C

4.3 Fluid Properties

Table 3: Fluid properties

Property	Symbol	Unit	Cold Water (Shell)	Hot Water (Tube)
Specific Heat	Cp	KJ/kg. K	4.178	4.178
Thermal Conductivity	K	W/m. K	0.6150	0.6150
Viscosity	µ	kg/m. s	0.001	0.001
Prandtl's Number	Pr	-	5.42	5.42
Density	P	1 kg/m^3	996	996

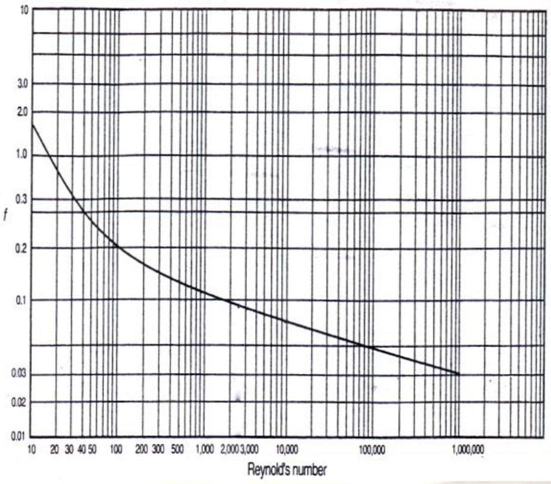


fig. 4 plot of 'f' as a function of shell-side Reynold's number

4.4 Thermal analysis of Segmental Baffle Heat Exchanger

1. Tube Clearance (C')

$C' = P_t - D_{ot}$

$= 0.0225 - 0.012$

$= 0.0105$

2. Cross-flow Area (A_S)

$A_S = (D_{is}\ C'\ L_B) / P_t$

$= (0.153 \cdot 0.0105 \cdot 0.06) / 0.0225$

$= 4.284\ E^{-3}$

3. Equivalent Diameter (D_E)

$D_E = 4\,[\,(P_t^2 - \pi \cdot D_{ot}^2 / 4) / (\pi \cdot D_{ot})$

$= 4\,[\,(0.0225^2 - \pi \cdot 0.012^2 / 4) / (\pi \cdot 0.012)\,]$

$= 0.04171$ m.

4. Maximum Velocity (V_{max})

$V_{max} = \dot{Q}_s / A$

$= 0.001 / (\frac{\pi}{4} \cdot D_{is}^2)$

…(since $\dot{Q}_s$ = 60 lpm = 3600 lph = 0.001 m^3/s)

$= 0.001 / (\frac{\pi}{4} \cdot 0.153^2)$

$= 0.0544$ m/s

5. Reynold's number (Re)

$Re = (\rho \cdot V_{max} \cdot D_E) / \mu$

$= (996 \cdot 0.0544 \cdot 0.04171) / 0.001$

$= 2259.948$

6. Prandtl's number (Pr)

Pr = 5.42

…(for MBT = 30°C and water as the medium)

7. Heat Transfer Co-efficient (α_o)

$\alpha_o = (0.36 \cdot K \cdot Re^{0.55} \cdot Pr^{0.33}) / R \cdot D_E$

(where $R = (\frac{\mu}{\mu w})^{0.14} = 1$ for water as medium)

$= (0.36 \cdot 0.6150 \cdot 2259.948^{0.55} \cdot 5.42^{0.33}) / 0.04171$

= 648.352 W/m²K

8. No. of Baffles (N_b)

$N_b = L_s / (L_b + \Delta_{SB})$

$= 1.123 / (0.06 + 0.005)$

≈ 17

9. Pressure Drop (Δ_{PS})

$\Delta P_S = [4 \cdot f \cdot \dot{m}_s^2 \cdot D_{is} \cdot (N_b + 1)] / (2 \cdot \rho \cdot D_E)$

…(f from graph and $\dot{m}_s = \dot{M} / A_s$)

$= (4 \cdot 0.09 \cdot 233.42^2 \cdot 0.153 \cdot 18) / (2 \cdot 996 \cdot 0.04171)$

$= 650.15$ Pa

= 0.65 KPa

4.5 Thermal analysis of Helical Baffle Heat Exchanger :

(Baffle Helix Angle 15°)

1. Tube Clearance (C')

$C' = P_t - D_{ot}$

$= 0.0225 - 0.012$

$= 0.0105$

2. Baffle Spacing (L_b)

$L_b = \pi \cdot D_{is} \cdot \tan \phi$
…(where ϕ is the helix angle = 15˚)

$= \pi \cdot 0.153 \cdot \tan 15$

$= 0.1288$

3. Cross-flow Area (A_S)

$A_S = (D_{is} \cdot C' \cdot L_B) / P_t$

$= (0.153 \cdot 0.0105 \cdot 0.1288) / 0.0225$

$= 9.196\ E^{-3}$

4. Equivalent Diameter (D_E)

$D_E = 4 [(P_t^2 - \pi \cdot D_{ot}^2 / 4) / (\pi \cdot D_{ot})$

$= 4 [(0.0225^2 - \pi \cdot 0.012^2 / 4) / (\pi \cdot 0.012)]$

$= 0.04171$ m.

5. Maximum Velocity (V_{max})

$V_{max} = \dot{Q}_s / A_s$

$= 0.001 / (9.169 \cdot E^{-3})$

…(since $\dot{Q}_s$ = 60 lpm = 3600 lph = 0.001 m^3/s

$= 0.1087$ m/s

6. Reynold's number (Re)

$Re = (\rho \cdot V_{max} \cdot D_E) / \mu$

$= (996 \cdot 0.1087 \cdot 0.04171) / 0.001$

$= 4515.74$

7. Prandtl's number (Pr)

Pr = 5.42

…(for MBT = 30˚C and water as the medium)

8. Heat Transfer Co-efficient (α_o)

$\alpha_o = (0.36 \cdot K \cdot Re^{0.55} \cdot Pr^{0.33}) / R \cdot D_E$

(where $R = (\frac{\mu}{\mu w})^{0.14} = 1$ for water as medium)

$= (0.36 \cdot 0.6150 \cdot 4515.74^{0.55} \cdot 5.42^{0.33}) / 0.04171$

= 948.98 W/m^2K

9. No. of Baffles (N_b)

$N_b = L_s / (L_b + \Delta_{SB})$

$= 1.123 / (0.1288 + 0.005) \approx 8$

10. Pressure Drop (ΔP_S)

$\Delta P_S = [4 \cdot f \cdot \dot{M}_F^2 \cdot D_{is} \cdot (N_b + 1)] / (2 \cdot \rho \cdot D_E)$

…(f from graph and $\dot{M}_F = \dot{m}_s / A_s$)

$= (4 \cdot 0.09 \cdot 108.75^2 \cdot 0.153 \cdot 9) / (2 \cdot 996 \cdot 0.04171)$

$= 70.55$ Pa

= 0.07 KPa

4.6 Thermal analysis of Helical Baffle Heat Exchanger :

(Baffle Helix Angle 25˚)

1. C' = 0.0105

2. Baffle Spacing (L_b)

$L_b = \pi \cdot D_{is} \cdot \tan \phi$
…(where ϕ is the helix angle = 25˚)

$= \pi \cdot 0.153 \cdot \tan 25$

$= 0.2241$

3. Cross-flow Area (A_S)

$A_S = (D_{is} \cdot C' \cdot L_B) / P_t$

$= (0.153 \cdot 0.0105 \cdot 0.2241) / 0.0225$

$= 0.016\ m^2$

4. Equivalent Diameter

$D_E = 0.04171$ m.

5. Maximum Velocity (V_{max})

$V_{max} = \dot{m}_s / A_s$

$= 0.001 / (0.016)$

$= 0.0625$ m/s

6. Reynold's number (Re)

$Re = (\rho \cdot V_{max} \cdot D_E) / \mu$

$= (996 \cdot 0.0625 \cdot 0.04171) / 0.001$

$= 2596.44$

7. Prandtl's no.

Pr = 5.42

8. Heat Transfer Co-efficient (α_o)

$\alpha_o = (0.36 \cdot K \cdot Re^{0.55} \cdot Pr^{0.33}) / R \cdot D_E$

$= (0.36 \cdot 0.6150 \cdot 2596.44^{0.55} \cdot 5.42^{0.33}) / 0.04171$

$= 699.94$ W/m²K

8. No. of Baffles (N_b)

$N_b = L_s / (L_b + \Delta_{SB})$

$= 1.123 / (0.2241 + 0.005)$

≈ 5

9. Pressure Drop (ΔP_S)

$\Delta P_S = [4 \cdot f \cdot \dot{M}_F^2 \cdot D_{is} \cdot (N_b + 1)] / (2 \cdot \rho \cdot D_E)$

$= (4 \cdot 0.08 \cdot 62.5^2 \cdot 0.153 \cdot 6) / (2 \cdot 996 \cdot 0.04171)$

= 13.8 Pa

= 0.013 KPa

4.7 Thermal analysis of Helical Baffle Heat Exchanger :

(Baffle Helix Angle 55˚)

1. C' = 0.0105

2. Baffle Spacing (L_b)

$L_b = \pi \cdot D_{is} \cdot \tan \phi$
…(where ϕ is the helix angle = 55˚)

$= \pi \cdot 0.153 \cdot \tan 55$

= 0.6864

3. Cross-flow Area (A_S)

$A_S = (D_{is} \cdot C' \cdot L_B) / P_t$

$= (0.153 \cdot 0.0105 \cdot 0.6864) / 0.0225$

$= 0.049$ m²

4. $D_E = 0.04171$ m.

5. Maximum Velocity (V_{max})

$V_{max} = \dot{m}_s / A_s$

$= 0.001 / (0.049)$

= 0.02 m/s

6. Reynold's number (Re)

$Re = (\rho \cdot V_{max} \cdot D_E) / \mu$

$= (996 \cdot 0.02 \cdot 0.04171) / 0.001$

= 847.81

7. Pr = 5.42

8. Heat Transfer Co-efficient (α_o)

$\alpha_o = (0.36 \cdot K \cdot Re^{0.55} \cdot Pr^{0.33}) / R \cdot D_E$

$= (0.36 \cdot 0.6150 \cdot 847.81^{0.55} \cdot 5.42^{0.33}) / 0.04171$

$= 378.2$ W/m²K

8. No. of Baffles (N_b)

$N_b = L_s / (L_b + \Delta_{SB})$

$= 1.123 / (0.6864 + 0.005)$

≈ 2

9. Pressure Drop (ΔP_S)

$\Delta P_S = [4 \cdot f \cdot \dot{M}_F^2 \cdot D_{is} \cdot (N_b + 1)] / (2 \cdot \rho \cdot D_E)$

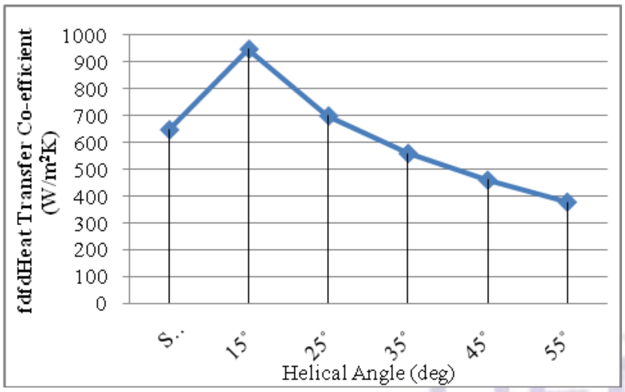
$= (4 \cdot 0.12 \cdot 20.4^2 \cdot 0.153 \cdot 3) / (2 \cdot 996 \cdot 0.04171)$

= 1.1 Pa **= 1.1 E^{-3} KPa**

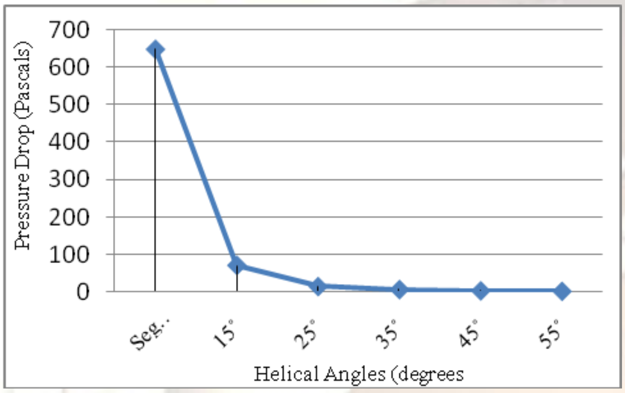
V. Results

Helix Angle	H.T Co-efficient α_o (W/m²K)	Pressure Drop Δ_{PS} (Pa)	$\alpha_o / \Delta P_S$
Segmental baffle	648.352	650	0.99
15˚	948.98	70.55	13.45
25˚	**699.94**	**13.81**	**50.68**
35˚	560.03	5.74	97.5
45˚	460.17	2.5	183.9
55˚	378.2	1.1	343.8

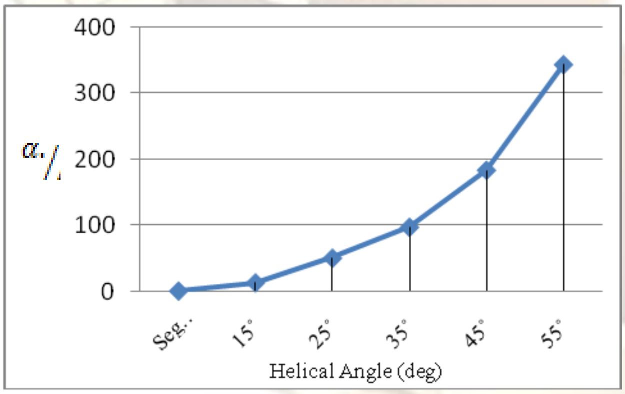
5.1 Graph Plots



Graph 1 : Heat Transfer co-efficient α_o vs. Varying Helical Angles ф



Graph 2 : Pressure drop ΔP_s vs. Varying Helical Angles ф



Graph 3 : Ratio of Heat Transfer co-efficient α_o and Pressure drop ΔP_s vs. Varying Helical Angles ф

VI. CONCLUSIONS

a. In the present study, an attempt has been made to modify the existing Kern method for continuous helical baffle heat exchanger, which is originally used for Segmental baffle Heat Exchangers.

b. The above results give us a clear idea that the Helical baffle heat exchanger has far more better Heat transfer coefficient than the conventional segmental Heat Exchanger.[Graph 1]

c. The above results also indicate that the pressure drop ΔP_s 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 2]

d. The ratio of Heat Transfer co-efficient per unit pressure drop is higher as compared to segmental baffle heat exchanger and **most desired in Industries for helical angle of 25°**.[Graph 3]

This helps reduce the pumping power and in turn enhance the effectiveness of the heat exchanger in a well-balanced way.

e. 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.

f. Suitable helix angle may be selected based upon the desired output and industrial applications. Helix angle of 15° may provide better heat transfer than the one with an angle of 25°, however at the expense of lesser pressure drop.

g. The ratio of Heat transfer co-efficient to Pressure drop is 50 for helix angle of 25° amongst all the other helical angles. This is the most desired result for industrial Heat Exchangers as it creates a perfect balance between the Heat transfer co-efficient and shell side pressure drop in a heat exchange.

NOMENCLATURE

Symbol	Quantity	Units
A_s	Shell Area	m^2
L_B	Baffle Spacing	m
C_p	Specific Heat	kJ/kgK
D_{ot}	Tube Outer Diameter	m
D_{is}	Shell Inner Diameter	m
D_E	Equivalent Diameter	m
α_o	Heat Transfer Co-efficient	$\frac{W}{m2 \cdot}$
N_b	Number of Baffles	-
Pr	Prandtl's No.	-
P_T	Tube Pitch	m
Re	Reynold's Number	-
ΔP_S	Total shell side pressure drop	Pa
μ	Dynamic viscosity	$Kg{\cdot}s/m^2$
ρ	Fluid Density	kg/m^3
V_{max}	Maximum Tube Velocity	m/s
ϕ	Helix Angle	-

ACKNOWLEDGEMENTS

The authors would like to acknowledge BCUD, University of Pune for providing necessary financial support (Grant no. BCUD/OSD/184/09).

REFERENCES

[1]. Andrews Malcolm, Master Bashir, *Three Dimensional Modeling of a Helixchanger Heat exchanger using CFD, Heat Transfer Engineering*, 26(6), (2005), 22-31.
[2]. Mukherjee R, Effective design Shell and Tube Heat exchangers, *Chemical Engg Progress*, (1998), 1-8.
[3]. Gang Yong Lei, Ya-Ling He, RuiLi, Ya-Fu Gao, Effects of baffle inclination angle on flow and heat transfer of a heat exchanger with helical baffles, *ScienceDirect-Chemical Engineering and Processing*, (2008), 1-10.
[4]. Peng B, Wang Q.W., Zhang C., An experimental Study of shell and tube heat exchangers with continuous Helical baffles, *ASME Journal of Heat transfer*, (2007).
[5]. Hewitt G.F, Shires G.L., Bott T.B., *Process heat transfer*, (CRC press), 275-285.
[6]. Master B.I, Chunagad K.S. Boxma A.J. Kral D, Stehlik P, *Most Frequently used Heat exchangers from pioneering Research to Worldwide Applications, vol. No.6*, (2006), 1-8.
[7]. Prithiviraj, M., and Andrews, M. J, Three Dimensional Numerical Simulation of Shell-and-Tube Heat Exchangers, *Foundation and Fluid Mechanics, Numerical Heat Transfer Part A - Applications, 33(8)*, 799–816, (1998).
[8]. Serth Robert W., *Process heat transfer principles and application*, ISBN:0123735882, 246-270, (2007).
[9]. Wadekar Vishwas *Enhanced and Compact Heat exchangers – A Perspective from process industry*, 5th international conference on *Enhanced and Compact Heat exchangers: Science, Engineering and Technology*, 35-41, (2005).
[10]. Wang Qui, Dong Chen Gui, Xu Jing, PengJi Yan, *Second-Law Thermodynamic Comparison and Maximal Velocity Ratio Design Of Shell-and-Tube Heat Exchanger With Continuous Helical Baffles, ASME journal*, 132/101801, 1-8, (2010).