COMPARATIVE THERMAL ANALYSIS OF HELIXCHANGER WITH SEGMENTAL HEAT EXCHANGER USING BELL-DELAWARE METHOD

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ABSTRACT

Heat exchangers are important heat transfer apparatus in oil refining, chemical engineering, environmental protection, electric power generation etc. The present work modifies the existing Bell-Delaware method used for conventional heat exchanger, taking into consideration the helical geometry of Helixchanger. Thermal analysis was carried out to study the impacts of various baffle inclination angles on fluid flow and heat transfer of heat exchangers with helical baffles. The analysis was conducted for conventional shell and tube heat Exchanger and Helixchanger for five baffle inclination angles. Analysis results indicate that continual helical baffles can reduce or even eliminate dead regions in the shell side of shell-and-tube heat exchangers. The pressure drop varies drastically with baffle inclination angle and shell-side Reynolds number. The variation of the pressure drop is relatively large for small inclination angle. However, for $\alpha > 35^0$, the effect of α on pressure drop is very small. Compared to the segmental heat exchangers, the heat exchangers with continual helical baffles have higher heat transfer coefficients to the same pressure drop. The detailed knowledge on the heat transfer and pressure drop across the shell side will provide further basis flow for further optimization of shell-and-tube heat exchangers.

KEYWORDS: *Bell-Delaware method, helical baffles, helix angle, heat transfer coefficient per unit pressure drop, Shell & tube heat exchanger.*

I. INTRODUCTION

Conventional heat exchangers with segmental baffles in shell side have some shortcomings resulting in the relatively low conversion of pressure drop into a useful heat transfer.
Both hydrodynamic studies and testing of heat transfer and the pressure drop on research facilities and industrial equipment showed much better performance of helically baffled heat exchanger when compared with conventional ones. These results in relatively high value of shell side heat transfer coefficient, low pressure drop, and low shell side fouling [1].

1.1 Desirable Features of Heat Exchangers:

In order to obtain maximum heat exchanger performance at the lowest possible operating and capital costs without comprising the reliability, the following features are required of an Exchanger:
(1) Higher heat transfer coefficient and larger heat transfer area.
(2) Lower pressure drop.
The objective of the present work is to determine the pressure drop and heat transfer on shell side of Helixchanger analytically. A comprehensive experimental investigation on heat transfer and pressure drop on Helixchanger is very expensive. The paper discusses developments in the Helixchanger and its design and research aspects. Important geometrical parameters have been discussed while calculating the thermal parameters. For calculating the pressure drop and heat transfer on shell side of conventional as well as helical baffle heat exchanger, Bell Delaware method with suitable modification has been used. Results and discussions shows that for all the helical baffle heat

exchangers studied, the ratios of heat transfer coefficient to pressure drop are higher than those of a conventional segmental heat exchanger. This means that the heat exchangers with helical baffles will have a higher heat transfer coefficient, when consuming the same pumping power.

1.2 Developments in shell and tube exchanger:

The developments for shell and tube exchangers center around better conversion of pressure drop into heat transfer by improving the conventional baffle designs. With single segmental baffles, a significant proportion of the overall pressure drop is wasted in changing the direction of flow. This baffle arrangement also leads to other undesirable effects such as dead spots or zones of recirculation which can cause increased fouling, high leakage flow and large cross flow. The cross flow not only reduces the mean temperature difference but can also cause potentially damaging tube vibration [2].

1.2.1 Helical baffle Heat Exchanger or Helixchanger:

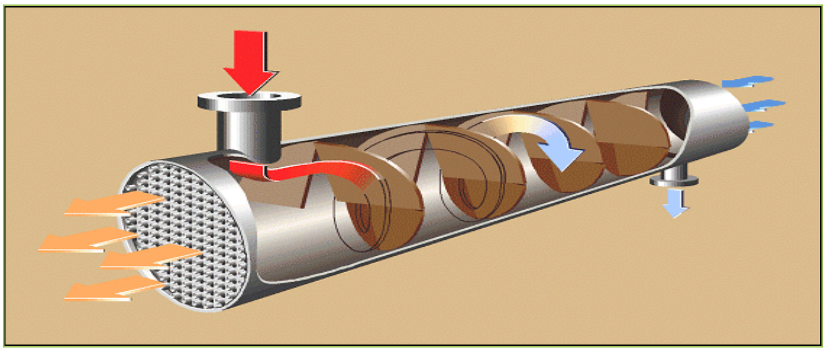


Figure 1: Helical baffle heat exchanger

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 [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.

1.2.2 Research aspects

Research on the Helixchanger has forced on two principle areas.

- Hydrodynamic studies on the shell side and
- Heat transfer and pressure drop studies on small scale and full industrial scale equipment.

1.2.3 Design aspects:

An optimally designed helical baffle arrangement depends largely on the heat exchanger operating conditions and can be accomplished by appropriate design of helix angle, baffle overlapping and tube layout.
In the original method for conventional shell and tube Heat exchanger, an ideal shell-side heat transfer coefficient is multiplied by various correction factors for flow distribution and the non-idealities such as leakage streams, bypass stream etc. are taken into consideration. For Helical baffle geometry it is suggested that some correction factor are not required and suitable modification is done in the Bell Delaware method [4].

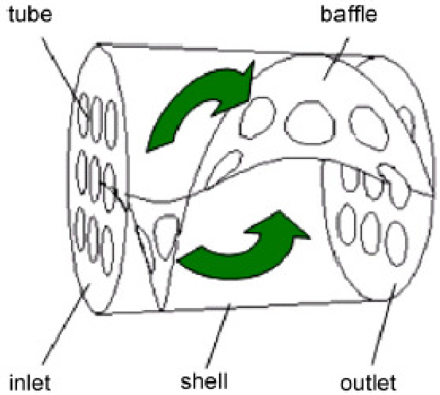


Figure 2: Helixchanger pitch

1.2.4 Important Parameters:

- Pressure Drop (ΔP_S)
- Baffles pitch(Helix angle) angle (α)
- Baffle space (B)
- Surface area (A)
- Heat transfer coefficient (h_o)

In designing of Helixchanger, pitch angle, baffle arrangement, and the space between two baffles with the same position are important parameters. Baffle pitch angle (α) is the angle between flow and perpendicular surface on exchanger axis and *B* is space between two following baffles with the same situation.

Changing the pitch angle in helical baffle system can create wide range of flow velocities as compared to the baffle space and baffle cut in traditional heat exchangers. Moreover, overlapping of helical baffles is a parameter that can affect significantly on shell side flow pattern.

A comprehensive study on heat transfer coefficient and pressure drop characteristics of Segmental baffle Shell & Tube heat exchanger and Helixchanger has been done. The performance of Helixchanger for different helix angles are calculated and compared with Segmental heat exchanger and result is tabulated.

II. HEAT TRANSFER COEFFICIENTS AND PRESSURE DROP CALCULATIONS

Heat transfer coefficients and pressure drop calculations are the main part of design of heat exchangers with a given duty. In traditional approaches such as Kern and Bell–Delaware methods are used to calculate the heat transfer coefficient & pressure drop.

For conventional tubular Heat Exchangers, Kern method, which was an attempt to correlate 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 (25%) and cannot adequately account for baffle-to-shell leakages. Nevertheless, although the Kern equation is not particularly accurate, it does allow a simple & rapid calculation of shell side coefficients and pressure drop to be carried out and has been successfully used since its inception.

The next stage of development of shell side calculation methods was that commonly described as Bell-Delaware method (Bell-1963). In this method, correction factors for baffle leakage effects, etc., are introduced based on experimental data. This method is widely used and is the basis of the approach recommended, in the Heat exchanger design handbook [5-7].

2.1 Thermal analysis of Segmental baffle heat exchanger & Helixchanger

In the present work, Bell Delaware method has been used for comparing the thermal performance of Segmental baffle heat exchanger & Helixchanger. Firstly, thermal parameters have been calculated for Segmental baffle heat exchanger using various steps mentioned by Bell-Delaware. Then, suitable modifications have been carried out in different steps for calculating the thermal parameters of Helixchanger. Then comparative analysis has been done between the two heat exchangers. Following data has been assumed and the geometrical parameters are kept constant for both types of heat exchangers.

Table 1: Input Data.

Sr.no	**Parameter**	**Shell side**	**Tube side**
1	Fluid	Water	Water
2	Volume flow rate (Q_s)	60 lpm	10 lpm
3	Mass flow rate (m_s)	1Kg/s	0.17 Kg/s
4	Shell ID (D_{is})	0.153 m	
5	Shell length	1.123m	
6	Tube pitch	0.0225 m	
7	No of passes	1	
8	Baffle cut	25% =0.25	
9	Baffle pitch	0.060 m	
10	Nozzle ID	0.023 m	0.023 m
11	No of baffles	17	
12	Mean Bulk Temperature	30°C	30°C
13	Tube OD		0.012 m
14	Tube thickness		0.0014 m
15	No of tube		24
16	Helix angle	25^0	
17	No of baffles for Helixchanger	5	

Fluid property:

Table 2: Fluid properties

Property	**Unit**	**Cold Water (Shell Side)**	**Hot Water (Tube Side)**
Cp	kJ/kg. K	4.178	4.178
K	W/m. K	0.6150	0.6150
μ	Kg.s/m^2	0.001	0.001
Sp Gr.		0.996	0.996
Pr		5.42	5.42
ρ	1 kg/m3	996	996

Leakage and bypass clearances:

i) Tube to baffle clearance (δ_{bt}) = 0.0004 m.
ii) Baffle to shell clearance (δ_{bs}) = 0.001 m
iii) Shell to bundle clearance (δ_{sbd}) = 0.01428 m.
iv) Shell outer tube limit (D_{otl}) = 0.1387 m
v) Shell central tube limit (D_{ctl}) = $(0.1387 - 0.012)$.

= 0.1267 m

1) Cross flow area: Area between central baffle spaces

$$Sm = B\left[(Ds - Dotl) + \frac{(Dotl - Do)}{Pteff}(Pt - Do)\right] \quad (1)$$

Sm = 0.004405m^2

2) Reynolds number

$$\mathrm{Re} = \rho V max \frac{Do}{\mu} \quad (2)$$

$Re = 2725$

3) Heat transfer coefficient:

$$h_o = h_{ideal}\,(J_c\, J_L\, J_B) \quad (3)$$

$$ho = 2570.82 W/m2k$$

4) Window flow area: Area between the baffle cut & shell wall.

$$S_w = Ds^2\left[\cos^{-1}\frac{(Ds-2Lc)}{Ds} - \frac{(Ds-2Lc)}{Ds}\sqrt{1-\left(\frac{(Ds-2Lc)}{Ds}\right)^2}\right]\frac{1}{4} - nt(1-Fc)\pi\frac{Do^2}{8} \quad (4)$$

$$= 0.003139 m2$$

5) Shell side pressure drop

$$\Delta P_s = [(n_b - 1)\times \Delta P_c \times R_B + n_b \Delta P_w]R_L + 2\Delta P_c(1 + N_{cw}/N_c)R_B \quad (5)$$

$$= 2209.6 Pa$$

2.2 Thermal analysis of Helixchanger

1) Cross flow area:

$$Sm = \frac{B}{2}\left[(Ds - Dotl) + \frac{(Dotl - Do)}{Pteff}(Pt - Do)\right] \quad (6)$$

$$= 0.008224 m^2$$

In Helixchanger, baffles cover half of the shell diameter so cross flow area reduces to half.

B=∏ D_S tanα ------ [8]

2) Reynolds number:

$$Re = \rho Vmax\frac{Do}{\mu} \quad (7)$$

$$= 1459$$

3) Heat transfer coefficient:

$$h_o = h_{ideal}\,(J_c\, J_L\, J_B) \quad (8)$$

$$= 1639.65 W/m^2$$

4) Window flow area:

As the helical baffle runs over the diameter window area does not exists.

From geometry Window flow area,

$$S_w = 0 \quad (9)$$

5) Shell side pressure drop:

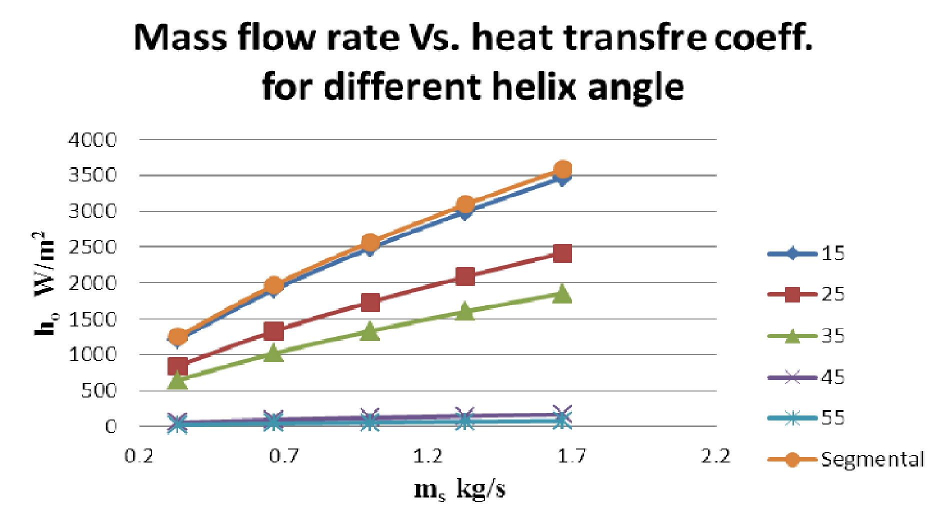
$$\Delta P_s = [(n_b - 1)\times \Delta P_c \times R_B + n_b \Delta P_w]R_L + 2\Delta P_c(1 + N_{cw}/N_c)R_B \quad (10)$$

$$= 420.5 Pa$$

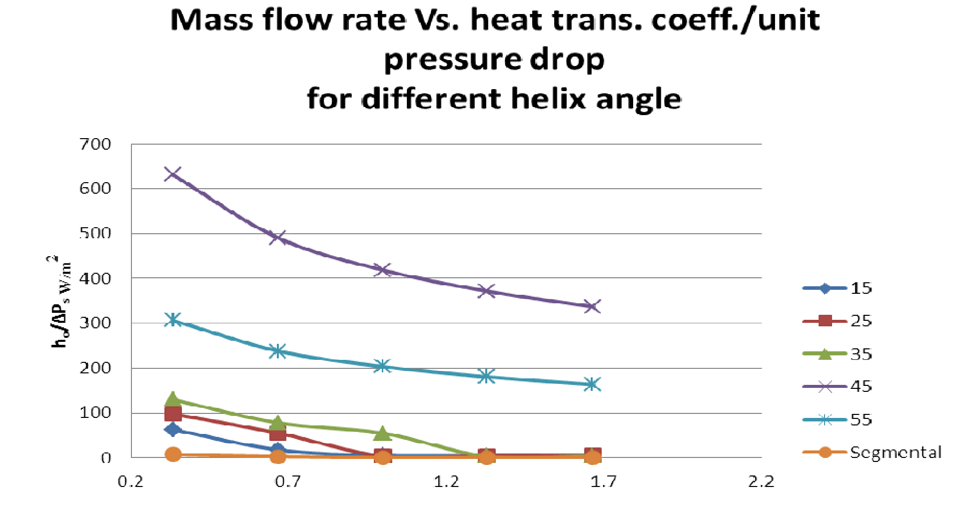
III. RESULT AND DISCUSSION

Table 3

Sr. No.	Parameter	Segmental baffle Heat Exchanger	Helixchanger 15^0	25^0	35^0	45^0	55^0
1	$S_m(m^2)$	0.004405	0.004729	0.008224	0.012353	0.480665	0.686437
2	R_e	2725	2538	1459	972	25	9
3	$h_O(W/m_2.k)$	2570.82	2487.395	1639.65	1331.379	122.7894	59.657
4	$\Delta P_C(Pa)$	112.22	106.7728	95.83	5.4248	0.064337	0.054337
5	$\Delta P_W(Pa)$	102.24	0.2989	0.1719	0.1144	0.002941	0.002341
6	$\Delta P_S(Pa)$	2209.6	654.5	420.5	24.211	0.2934	0.2534
7	$h_O/\Delta P_s$	1.16	3.80	3.90	53.24	420	236



Graph 1



Graph 2

It shows that, from table 3, the pressure drop for Segmental baffle heat exchanger is 2.209 KPa and for Helixchanger (15^0) it is 0.645 KPa and it is minimum for all helix angles. Also the, ratio of heat transfer coefficient to pressure drop for Helixchanger (25^0) is 3.90 & for Segmental baffle heat exchanger it is 1.16. i.e. this ratio for Helixchanger is about three times higher than Segmental baffle heat exchanger.

IV. Conclusion

In this paper, the thermal analysis for heat exchangers with different baffle inclination angles are performed to show the effects of baffle inclination angle on the heat transfer and pressure drop characteristics. The major findings are summarized as follows:

1) The flow pattern in the shell side with continual helical baffle is near-plug flow. Therefore, the dead region is eliminated and the heat transfer area is used more effectively.

2) As the shell-side Reynolds number is increased, the pressure drop increases for all the cases considered. For all helical baffle heat exchangers studied, the pressure drops are lower than those of the conventional segmental heat exchangers. The pressure drop decreases with the increase of baffle

inclination angle in all the cases considered. The change of the pressure drop is large in the small inclination angle region. However, the effects of baffle inclination angle on pressure drop are small when $\alpha>35^0$.

3) For all the helical baffle heat exchangers studied, the ratios of heat transfer coefficient to pressure drop are higher than those of a conventional segmental heat exchanger. This means that the heat exchangers with helical baffles will have a higher heat transfer coefficient, when consuming the same pumping power.

4) It can be concluded that proper baffle inclination angle will provide an optimal performance of heat exchangers. The detailed knowledge of the heat transfer and flow distribution provided in this investigation may serve as a basis for further optimization of shell-and-tube heat exchangers

5) Bell Delaware method available in the literature is only for the segmental baffle heat exchanger, since helical baffle heat exchanger is the recent development. Suitable modifications in Bell Delaware method can give the results for Helixchanger and can serve as a preliminary method for thermal analysis.

NOMENCLATURE

Symbol	Quantity
A	Area (m^2)
B	Baffle spacing (m)
Bc	Baffle cut (%)
C_P	Heat capacity at constant pressure (kJ/kg k)
D_{ctl}	Central tube limit diameter (m)
D_o	Outside diameter of tube (m)
D_{otl}	Outer tube limit diameter (m)
D_s	Inside diameter of shell (m)
F_c	Fraction of tubes in cross flow between baffle tips
H	Heat-transfer coefficient ($W/m^2.k$)
ho	Shell-side heat-transfer coefficient ($W/m^2.k$)
h_{ideal}	Ideal tube bank heat-transfer coefficient ($W/m^2.k$)
J_B	Heat-transfer correction factor for bundle bypass effects
J_C	Heat-transfer correction factor for effect of baffle window flow
J_L	Heat-transfer correction factor for baffle leakage effects
N_c	Number of tube rows crossed in flow between two baffle tips
N_{cw}	Effective number of tube rows crossed in flow through one baffle window
N_{ss}	Number of pairs of sealing strips
n_b	Number of baffles
n_t	Number of tubes in bundle
P	Pressure (kg/m^2)
Pr	Prandtl number
P_T	Tube pitch (m)
P'_T	Tube pitch parallel to flow direction (m)
R_B	Pressure-drop correction factor for bundle bypass effects
Re	Shell-side Reynolds number
R_L	Pressure-drop correction factor for baffle leakage effects
F_{bp}	Bundle bypass flow area (m^2)
S_m	Cross-flow area (m^2)
S_{sb}	Shell-to-baffle leakage area (m^2)
S_{tb}	Tube-to-baffle leakage area (m^2)
S_w	Window flow area (m^2)

Greek Letters

ΔP_C	Cross flow pressure drop (Pa)
$\Delta P_S(\Delta P_o)$	Total shell-side pressure drop (Pa)
ΔP_w(Pa)	Pressure drop in one baffle window, uncorrected for baffle leakage effects
δ_{sb}	Shell-to-baffle clearance (m)

δ_{tb}	Tube-to-baffle clearance (m)
μ	Viscosity (kg.s/m^2)
ρ	Fluid density (kg/m^3)
V_{max}	Maximum intertube velocity (m/s)
α	Helix angle (degree)

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