HEAT TRANSFER ENHANCEMENT USING DIFFERENT BAFFLE SPACING ARRANGEMENT

Prepared By

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Dedication

It is with our deepest gratitude and warmest affection that we dedicate this thesis to our professor

Dr. Shamsuddin Ahmed

Who has been a constant source of knowledge and inspiration.

Acknowledgement

In the name of Allah,the Most Gracious and the Most Merciful.

All praises to Allah and His blessing for the completion of this thesis.We thank God for all the opportunities,trials and strength that have been showered on us to finish writing the thesis.We experienced so much during this process,not only from the academic aspect but also from the aspect of personality.Our hublest gratitude to the holy Prophet Muhammad (peace be upon him) whose way of life has been a continuous guidance for us.

First and foremost,we would like to sincerely thank our supervisor,

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Our deepest gratitude goes to our family members.It would not be possible to complete this project without the support from them.We would like to thank our dearest parents.

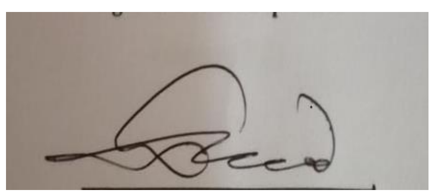
We would sincerely like to thank all our beloved friends who were with us and support us through thick and thin.

May The Almighty shower the above cited personalities with success and honor in their life.

Certificate of Research

The project titled **“HEAT TRANSFER ENHANCEMENT USING DIFFERENT BAFFLE SPACING ARRANGEMENT’** submitted by **Md. Muhibur Rahman(141434)** has been accepted as satisfactory in partial fulfillment of the requirement of the degree of Bachelor of Science in Mechanical Engineering on March 2021.

Signature of the Supervisor



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Candidate’s Declaration

It is hereby declared that this project or any part of this has not been submitted elsewhere for the award of any degree or diploma.

Signature of the Candidate

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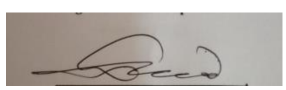

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Abstract

To enhance the heat transfer rate the feature of a shell and tube heat exchanger was developed by considering the baffle spacing and baffle cut.The research in this project was carried by manual calculations and was performed for a cross-flow heat exchanger with water as tube side fluid and methyle alcohol as shell side fluid with four different numbers of baffle spacing and 35% of baffle cut. Shell and tube heat exchangers in their various construction modifications are probably the most widespread and commonly used basic heat exchanger configuration in the process industries. There are many modifications of the basic configuration which can be used to solve special problems. Baffles serve two functions: Most importantly, they support the tubes in the proper position during assembly and operation and prevent vibration of the tubes caused by flow-induced eddies,and secondly, they guide the shell-side flow back and forth across the tube field, increasing the velocity and the heat transfer coefficient. Our designed baffle spaces are :

For case A: 380 mm

For case B: 420 mm

For case C: 470 mm

For case D: 520 mm

In this project we acquired sell side calculations using Kern method and after observing the shell side calculations,the overall heat transfer coefficient was decreased by 37.78% .

Keywords

Shell and tube heat exchanger,design,baffles.

Introduction

In most industrial fields,cooling and heating fluids are needed and the device which is utilized to execute the heat transfer between two different streams is known as heat exchanger. In most heat exchangers ,the fluids are separated and usually they do not mix. It is used in many applications such as space heating and airconditioning,chemical processing,waste heat recovery[1,2]. Basically they are classified according to heat transfer process,process- mechanism, flow-arrangements, construction etc[3]. While the simple double-pipe exchanger is inefficient for flow rates that cannot be easily managed in a few tubes, the modulating single-pipe exchanger is able to accommodate flow rates up to five times greater. If several double pipes are used in parallel,the weight of metal required for the outer tubes becomes large. The shell-andtube construction, such as that shown in Fig.-1,where one shell serves for many tubes, is more economical.This exchanger, because it has one shell-side pass and one tube-side pass, is a 1-1 exchanger. The most common type of heat exchanger that is utilized is the shell and tube heat exchanger as they are consist of structural simplicity and design flexibility This heat exchanger has a good mechanical layout and can operate under pressure[2]. It is made of different types of material among which selected materials are used for operating pressure and temperature. It has got many applications in power generation, petroleum refinery and chemical industries [3]. They are used as oilcooler,condenser,feed water heater etc. The shell and tube heat exchangers are having issues that are affecting the overall performance of the device as they were designed and manufactured.The common problem is fouling and it has to be considered by the engineers as it is an aggregation of undesirable materials on heat exchanger surfaces that decreases the heat transfer rate and increases the resistance to flow causing high pressure drop[3]. The principal components of a shell and tube heat exchanger are shell, shell cover, tubes, channel, channel cover, tube sheet and baffles[3]. The main component on which this project is focusing on, is baffle. Baffles are plates that are installed on the shell side of the tube bundle to support the tubes, maintain the spaces between them, and direct the shell side fluid flow across the tube bundle in a specific direction[3]. The most common type of baffle is segmental baffle. They improve heat transfer by increasing fluid turbulence on the shell side by causing the shell side fluid to flow in a zigzag pattern, as well as increasing pressure drop. As a result of rising electricity consumption, high pumping power will be required[4]. We investigated the effects of four different central distances among four different designs of baffles. We discovered that increasing the baffle spacing can improve the heat transfer rate, which is one of the most common issues with this device. The shell-side and tube-side heat-transfer coefficients are equally important in an exchanger, and both must be large to achieve a satisfactory overall coefficient. The shell-side liquid's velocity and turbulence are just as important as the tube-side liquid's. There must be a minimum distance between the tubes to prevent the tubes from weakening. It is not possible to space the tubes so close together that the path outside the tubes has the same area as the path inside the tubes. The velocity on the shell side is low in comparison to the tube side if the two streams are of comparable magnitude. Baffles are installed in the shell to reduce the cross section of the shell-side liquid and force it to flow across rather than parallel to the tube bank. The increased turbulence in this type of flow raises the shell-side coefficient even more. The shell-side and tube-side heat-transfer coefficients are equally significant in an exchanger, and both must be high to achieve a satisfactory overall coefficient. The shell-side liquid's velocity and turbulence are just as significant as the tube-side liquid's. There must be a minimum gap between the tubes to prevent the tubes from weakening. It is not possible to space the tubes so close together that the path outside the tubes has the same area as the path inside the tubes. The velocity on the shell side is poor in contrast to the tube side if the two streams are of equal magnitude. Baffles are mounted in the shell to reduce the cross section of the shell-side liquid and force it to flow across rather than parallel to the tube bank. The increased turbulence in this form of flow boosts the shell-side coefficient even further. Kern provided a straightforward method for measuring the pressure drop and heat transfer coefficient on the shell side. This form, however, is limited to a fixed baffle cut (35%) and cannot account for baffle to shell and tube to baffle leakage. When the shell side Reynolds number is less than 2000, the Kern method is also inapplicable in laminar flow. Although the Kern equation is not very precise, it does allow for a fast and easy measurement of the shell side heat transfer coefficient and pressure decrease. Centered on a comprehensive investigation, after a series of studies, the Bell-Delaware method [2] emerged as a superior method. Correction factors for baffle leakage effects are implemented using this approach, which is based on comprehensive experimental evidence. This is a commonly used and suggested form. TEMA “X” is the shell type (or shell configuration).

Which is a pure cross-flow shell, where the shell side fluid enters at one side of the shell, flows across the tubes, and exits from the opposite side of the shell. The flow may be introduced through multiple nozzles located strategically along the length of the shell in order to achieve good distribution. The pressure drop will be extremely low,in fact, there is hardly any pressure drop in the shell, and whatever pressure drop exists is virtually in the nozzles. Full support plates can be located if needed for structural integrity; they do not interfere with the shell side flow because they are parallel to the flow direction. The failure of most shell and tube heat exchangers is due to "flow-induced vibration." X (cross-flow) type is absolutely free from flow-induced vibration. In this type of shell and tube heat exchanger, there is no segmental-type of baffle, and the shell side fluid allows just one trip through the package. The bundle usually has adequate entry and exit flow areas, but in some cases,it may be necessary to provide a full-length perforated distributor plate at the inlet to aid flow distribution. Segmental or cross-flow baffles are standard. Single, double, and triple segmental baffles are used. The single segmental baffle shape is the most common. The segment sheared off must be less than half of the diameter in order to insure that adjacent baffles overlap at least one full tube row. The double segmental baffle reduces cross-flow velocity for a given baffle spacing. The triple segmental baffle, also known as the "window-cut" baffle, decreases both cross and long-flow velocities. The single segmental baffle configuration creates an unacceptably high shell-side pressure drop for several high-velocity gas flows. The double segmental baffle is one way to maintain the structural advantages of the segmental baffle while lowering the pressure drop. The centerline-to-centerline difference between neighboring baffles is known as baffle spacing. In STHE design, it is the most important parameter[4]. To measure a near-accurate overall heat transfer coefficient of the chosen heat exchanger in this analysis. Calculate the heat transfer coefficients on the tube and shell sides, as well as the contribution of the tube wall to the resistance and the necessary fouling resistances.They are most commonly used in STHE applications, where they are placed underneath the tubes to preserve their distance from each other and to assist in the cross flow formation, resulting in better heat transfer. Generally speaking, the target of this objective is not met due to a departure from cross flow. And as a result of the number of leaks and bypass flows needed, there are additional flow requirements for the building.As in, of the exchanger Reducing the cost of heat exchanger operations is the primary objective of this initiative.Performing investment and operation of the exchange One shell and tube heat exchanger feature is that it is designable. By applying optimization, one can choose the optimum inter-baffle spacing and the baffle cut[4].indicated that there might be a gap in the space between the baffles with the upper limit varying from 20% of the inner space The diameter of the shell is exactly the same as the inner diameter of the shell. On the advice of, baffle how wide of a slice the cutter has to make can vary between a minimum of 15% of the inner shell diameter and a maximum of 45% ,the inner shell diameter is extended.

Nomenclature:

A_c	Cross sectional area of tube	m
A_s	Cross flow area at shell centerline	m^2
B	Baffle spacing	m
CF	Cleanliness Factor	
D_e	Equivalent diameter	m
D_i	Shell inside diameter	m
d_i	Tube inside diameter	m
d_o	Tube outside diameter	m
f_o	Friction factor on shell side	
G_s	Shell side mass velocity	$Kg/m^2 \cdot s$
H_i	Tube side convection heat transfer coefficient	$W/m^2 \cdot k$
H_o	coefficient Shell side convection heat transfer	$W/m^2 \cdot k$
j_h	Heat transfer factor	
k	Thermal conductivity of tube material	W/m.k
L	Tube Length	m
N_b	Number of baffles	
N	Number of tubes	

Nu_i	Nusselt number of tubeside fluid	
Nu_o	Nusselt number of shell side fluid	
OS	Over design percentage	
P	Temperature effectiveness ratio	
ΔP_s	Pressure drop on shell side	Pa
ΔP_{total}	Total pressure drop on tube side	Pa
P_T	Tube pitch size	
$\dot{Q}$	Heat transfer rate	W
R	Heat capacity rate ratio	
$R_{f,total}$	Total fouling resistance	$m^2.k/W$
$T_{c,i}$	Cold fluid inlet temperature	^0C
$T_{c,o}$	Cold fluid outlet temperature	^0C
$T_{h,i}$	Hot fluid inlet temperature	^0C
$T_{h,o}$	Hot fluid inlet temperature	^0C
ΔT_{ln}	Logarithmic mean temperature difference	^0C
U_c	Overall heat transfer coefficient for clean surface	$W/m^{2\cdot}k$
U_f	Overall heat transfer	$W/m^{2\cdot}k$
	coefficient for fouled surface	
μ_o	Shell side fluid viscosity	Pa.s
Re_i	Tube side fluid Reynolds number	
Re_o	Shell side fluid Reynolds number	
μ_i	Tube side fluid viscosity	Pa.s
$\dot{m}_o$	Shell side fluid mass flow rate	kg/s
$\dot{m}_i$	Tube side fluid mass flow rate	kg/s
k_i	Thermal conductivity of tube side fluid	$W/m^{2\cdot}k$
k_o	Thermal conductivity of shell side fluid	$W/m^{2\cdot}k$
	coefficient for fouled surface	
μ_o	Shell side fluid viscosity	Pa.s
Re_i	Tube side fluid Reynolds	

	number	
Re_o	Shell side fluid Reynolds number	
μ_i	Tube side fluid viscosity	Pa.s
$\dot{m}_o$	Shell side fluid mass flow rate	kg/s
$\dot{m}_i$	Tube side fluid mass flow rate	kg/s
k_i	Thermal conductivity of tube side fluid	$W/m^2 \cdot k$
k_o	Thermal conductivity of shell side fluid	$W/m^2 \cdot k$

Design Methodology

In this project there are different calculations for shell side and tube side fluid. The design of a heat exchanger is referred to as the sizing problem[5]. In a broad sense, it means the determination of exchanger construction type, flow arrangement, tube and shell material, and physical size of an exchanger to meet the specified heat transfer and pressure drop. Heat transfer mode in a shell-and-tube heat exchanger usually involves convection in each fluid and conduction through the wall separating the two fluids. In the analysis of shell-and-tube heat exchangers, it is convenient to work with an overall heat transfer coefficient *U* that accounts for the contribution of all these modes at heat transfer. The rate of heat transfer between the two fluids at a location in a heat exchanger depends on the magnitude of the temperature difference at that location, which varies along the shell-andtube heat exchanger [6,7] In this design methodology, Tubular Exchangers Manufacturers Association (TEMA) standards have been followed strictly. The inputs to the sizing problem are as follows:

• Flow rates of both streams
• Inlet and outlet temperatures of both streams
• Allowable pressure drop of both streams

Calculation Procedure for Tube side fluid

The Nusselt number in turbulent flow is related to the friction factor through the **Chilton–Colburn** analogy,expressed as,

$$Nu=0.125f\,RePr^{1/3}\text{……………..}(1)$$

Once the friction factor is available, this equation can be used conveniently to evaluate the Nusselt number for both smooth and rough tubes[8].For fully developed turbulent flow in smooth tubes, a simple relation for the Nusselt number can be obtained by substituting the simple power law relation

$$f=0.184Re^{-0.2}$$

for the friction factor into Eq.(1)

For the tube side calculation we are going to find the value of convective heat transfer co-efficient for the tube side fluid,H_i and the equation that is used[1],

$$Nu_i=\frac{H_i\,d_i}{k_i}\text{ ....}(2)$$

To find the tube side Reynolds number, Most pipe flows encountered in practice are turbulent. Laminar flow is encountered when highly viscous fluids such as oils flow in small diameter tubes or narrow passages.For flow in a circular tube, the Reynolds number is defined as we are using equation[3],

$$Re_i=\frac{4\dot{m}_i}{\pi d_i \mu_i}\text{…………}(3)$$

and from the mean temperature of $T_{c,i}$ and $T_{c,o}$ the prandtle number,Pr_i
;

dynamic viscosity, μ_i for tube side fluid can be found [1].To find the H_i we have to find the value of Nu_i and to find that the **Chilton–Colburn** equation is used. Heat transfer correlations for turbulent fluid flow in the tubes are commonly used in the design and performance calculations of heat exchangers [1-4]. The value of heat transfer coefficient significantly affects the value of the thermal stress [5]. Time changes in optimum fluid temperature determined from the condition of not exceeding the stress at a point lying on the inner surface of the pressure component, very strong depend on the heat transfer coefficient [6-9]. Therefore, continuous experimental research is carried out to find a straightforward and accurate heat transfer correlations. The empirical correlation of Dittus-Boelter [10-12] has gained widespread acceptance for prediction of the Nusselt number with turbulent flow in the smoothsurface tubes that is,

$$Nu_i = 0.023 Re_i^{0.8} Pr_i^{\frac{1}{3}} \ldots\ldots\ldots\ldots (4)$$

After finding the Nusselt number from (3) tube side heat transfer co-efficient can be determined from (1).The tube-side calculation is only used to determine the value of the tube side convecteive heat transfer coefficient,H_i.As these equations are not dealing with baffle spacing,the values from equation (1),(2),(3) will remain constant for four different baffle spacing arrangement as different baffle spacing are only affecting the shell side calculations.

Calculation Procedure for Shell side fluid

For shell side calculation,we have to determine all the values using the Kern Method[5]. This method was based on experimental work on commercial exchangers with standard tolerances and will give a reasonably satisfactory prediction of the heat-transfer coefficient for standard designs. The prediction of pressure drop is less satisfactory, as pressure drop is more affected by leakage and bypassing than heat transfer[11]. The shell-side heat transfer and friction factors are correlated in a similar manner to those for tube-side flow by using a hypothetical shell velocity and shell diameter. As the cross-sectional area for flow will vary across the shell diameter, the linear and mass velocities are based on the maximum area for cross-flow: that at the shell equator.First,we need to calculate the cross-flow area A_s which has an equation,dealing with baffle spacing,l_B,that is,

$$A_s = \frac{(P_t - d_o) D_s l_B}{P_t} \ldots\ldots\ldots\ldots (5)$$

Now we are going to calculate the shell side mass velocity G_s,with the equation,

$$G_s = \frac{\dot{m}_o}{A_s} \ldots\ldots\ldots\ldots (6)$$

After finding the value of G_s we will go for the[6] value for the equivalent

diameter,d_e,that is,

$$d_e = \frac{1.27}{d_0}\left({P^2}_t - 0.785 d_0^2\right) \ldots\ldots\ldots (7)$$

Now we are going to determine the value of shell side Reynold number[7],that is,

$$Re_0 = \frac{G_s d_e}{\mu_0} \ldots\ldots\ldots\ldots\ldots (8)$$

The shell side prandtle number,Pr_0 can be calculated from the average temperature of the shell side inlet and outlet temperature.Then we have to find the value of shell side heat transfer factor, j_h from Reynolds number vs shell side heat transfer factor(j_h) Friction factor chart[10] for determining the value of Nusselt number,Nu_0 for the shell side fluid.

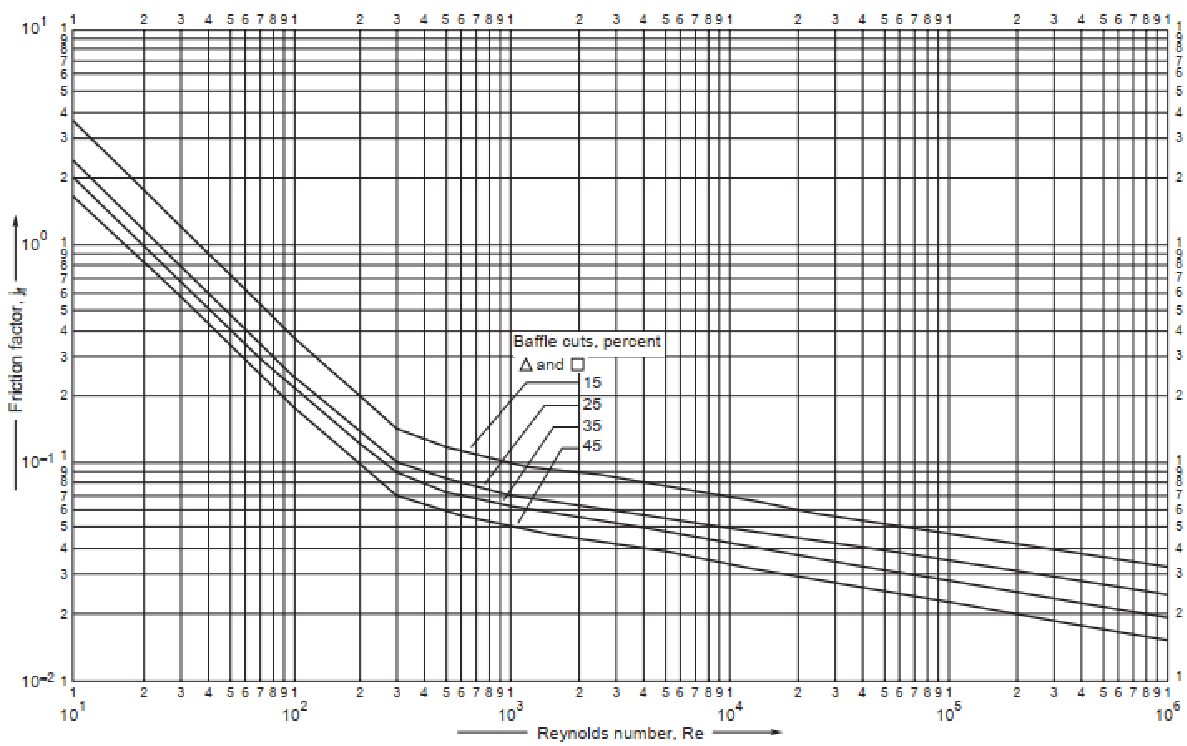


Fig-1: Reynolds number vs Friction factor chart

After finding the Re_0 , Pr_0 , j_h we will find the value of Nusselt number,using the baffle cut percentage,from the[9] chart above.

$$Nu_0 = j_h Re_o Pr_o^{\frac{1}{3}} \left(\frac{\mu_o}{\mu_w}\right)^{0.14} \ldots\ldots\ldots\ldots(9)$$

After finding the nusselt number for shell side fluid, the shell[9] side convective heat transfer coefficient, H_0 , can be determined,

$$Nu_o = \frac{H_o d_e}{k_o} \ldots\ldots\ldots\ldots\ldots\ldots(10)$$

Then we need to calculate the value for the overall heat transfer co-efficient for clean surface[10],that is,

$$U_c = \frac{1}{\frac{d_0}{d_i h_i} + \frac{r_o \ln(\frac{r_o}{r_i})}{k} + \frac{1}{h_o}} \ldots\ldots\ldots\ldots\ldots(11)$$

After finding U_c,we will go for U_f using the equation[10],

$$U_f = \frac{1}{\frac{1}{U_c} + R_{f,total}} \ldots\ldots\ldots(12)$$

After finding U_c and U_f we can calculate the value of the cleanliness factor percentage[11],

$$CF = \frac{U_f}{U_c} \text{ x } 100\% \ldots\ldots.(13)$$

For determining the overdesign percentage,we will use the equation[11],

$$OS = \left[\frac{U_c}{U_f} - 1\right] \times 100\% \quad ....(14)$$

Before starting the design calculation we need to tabulate the values of the properties that will remain constant throughout the calculation of all the four different baffle spacing arrangement.

Available design data: (Table-1)

Data	Tube side	Shell side
Fluid name	Water	Methanol
Mass flow rate(kg/s)	48.7	125
Inlet Temperature(^{0}C)	25	70
Outlet Temperature(^{0}C)	37	40
Data	Tube side	Shell side
Fluid name	Water	Methanol
Mass flow rate(kg/s)	48.7	125
Inlet Temperature(^{0}C)	25	70
Outlet Temperature(^{0}C)	37	40

Some constant values of tube side and some properties of both fluids.

Table-2:

Total fouling resistance,$R_{f,total}$(m^2k/W)	4.7×10^{-4}
Thermal conductivity of tube material,k(W/m.k)	63.67
Number of tubes,N_t	108
Tube Pitch(m)	30×10^{-3}

Calculation For Case D:

At D,we find the baffle space , l_B =520mm.

Tube side calculation:

From (2) we find, $Re_i = \frac{4\dot{m}_i}{\pi d_i \mu_i}$

$$= \frac{4\times48.7}{\pi\times21\times7.7922\times10^{-4}\times10^{-3}}$$

$$= 3.7893\times10^6$$

From (3) we find the equation for nusselt number,

$$Nu_i = 0.023Re_i^{0.8}Pr_i^{\frac{1}{3}}$$

$$= 0.023\times(3.7893\times10^6)^{0.8}\times (5.276)^{\frac{1}{3}}$$

$$= 7.334\times10^3$$

Then from (1), we find,

$Nu_i=\frac{H_i\, d_i}{k_i}$

Or, $H_i = \frac{Nu_i k_i}{d_i}$

$$= \frac{7.334\times10^3\times0.6166}{21\times10^{-3}}$$

$$= 2.1534\times10^5 W/m^2k,$$

which will remain constant for other three different baffle spacing arrangements.

Shell side calculation:

From equation (4) we find,

$$A_s = \frac{(P_t - d_o)D_i l_B}{P_t}$$

$$= \frac{[(30\times10^{-3})-(25\times10^{-3})]890\times10^{-3}\times520\times10^{-3}}{30\times10^{-3}}$$

$$=7.71333\times10^{-2} m^2$$

Shell side mass velocity,

$$G_s = \frac{\dot{m}_o}{A_s}$$

$$= \frac{125}{7.7133\times10^{-2}}$$

$$=1620.5774 kg/m^2 s$$

Shell side equivalent diameter,

$$d_e = \frac{1.27}{d_0}(P_t^2 - 0.785 d_0^2)$$

$$= \frac{1.27}{25\times10^{-3}}[(30\times10^{-3})^2 - 0.785\times(25\times10^{-3})^2]$$

$$=2.0796\times10^{-2} m$$

Reynolds number,

$$Re_0 = \frac{G_s d_e}{\mu_0} = \frac{1.6206\times10^{3}\times2.0796\times10^{-2}}{4.201\times10^{-4}}$$

$=8.02237\times10^{4}$

Taking 35% baffle cut and with the value of Re_0 ,we find,

$j_h=0.79$

From (3) we find,nusselt number of shell side fluid,

$Nu_0=j_h Re_o Pr_o^{\frac{1}{3}}(\frac{\mu_o}{\mu_w})^{0.14}$

$=0.79\times8.02237\times10^{4}\times5.2355^{0.333}\times(\frac{3.721}{5.04})^{0.14}$

$=1.0899\times10^{5}$

Again, $H_0=\frac{Nu_0 k_0}{d_e}$

$$= \frac{1.0899\times10^{5}\times0.1961}{2.0796\times10^{-2}}$$

$= 9.9908\times10^{5} W/m^{2}.k$

The overall heat transfer co-efficient for clean surface can be determined from,

$$U_c = \frac{1}{\frac{d_0}{d_i H_i} + \frac{r_o \ln(\frac{r_o}{r_i})}{k} + \frac{1}{H_o}}$$

$$= \frac{1}{\frac{25}{21 \times 2.1534 \times 10^5} + \frac{12.5 \times \ln(\frac{25}{21}) \times 10^{-3}}{205} + \frac{1}{9.9908 \times 10^5}}$$

$=5.8273\times10^4$ W/m^2.k

The overall heat transfer co-efficient including fouled surface we find the equation from (11),

$$U_f = \frac{1}{\frac{1}{U_c} + R_{f,total}}$$

$$= \frac{1}{\frac{1}{5.8232 \times 10^4} + 4.7 \times 10 - 4}$$

$=2.05267\times10^3$ W/m^2.k

Cleanliness Factor,

$$CF=[\frac{U_f}{U_c}\times 100\%]$$

$$=[\frac{2.05267\times 10^3}{5.8232\times 10^4}\times 100\%]=3.525\%$$

This was the design calculation of shell side fluid at baffle space ,B_4, l_B=520 mm.

Following the same procedure we can find these values for other three different baffle spaces and after calculating,the collected datas are tabulated below,

Table-3:

Summary results of different designs due to varying baffle spacing,

Parameter	Case A	Case B	Case C	Case D
Baffle Space(mm)	380	420	470	520
$A_s(m^2)$	5.6307×10^{-2}	6.2353×10^{-2}	6.9717×10^{-2}	7.7133×10^{-2}
$G_s(kg/m^2s)$	2.2173×10^{3}	2.0064×10^{3}	1.7929×10^{3}	1.6206×10^{3}
Re_0	1.0927×10^{5}	9.8863×10^{4}	8.8347×10^{4}	8.0224×10^{4}
j_h	.027	.026	.024	.023
Nu_0	4.9099×10^{3}	3.5879×10^{3}	3.526×10^{3}	3.056×10^{3}
$H_0(W/m^2.k)$	4.6529×10^{4}	3.3989×10^{4}	3.3403×10^{4}	2.8951×10^{4}
U_c $(W/m^2.k)$	2.6549×10^{4}	2.1934×10^{4}	2.1693×10^{4}	1.9723×10^{4}
U_f $(W/m^2.k)$	1.9698×10^{3}	1.9395×10^{3}	1.9376×10^{3}	1.9205×10^{3}
CF(%)	7.4195	8.8424	8.9319	9.7374

Discussion:

Baffle spacing is the centerline-to-centerline distance between adjacent baffles and identified as the most important geometric parameter affecting both pressure drop and heat transfer characteristics on the shell side design.Firstly it affected the cross-flow area then it affected the shell side Reynolds number and the nusselt number and that is how shell side heat transfer co-efficients have been changed.and if we check the graph below,

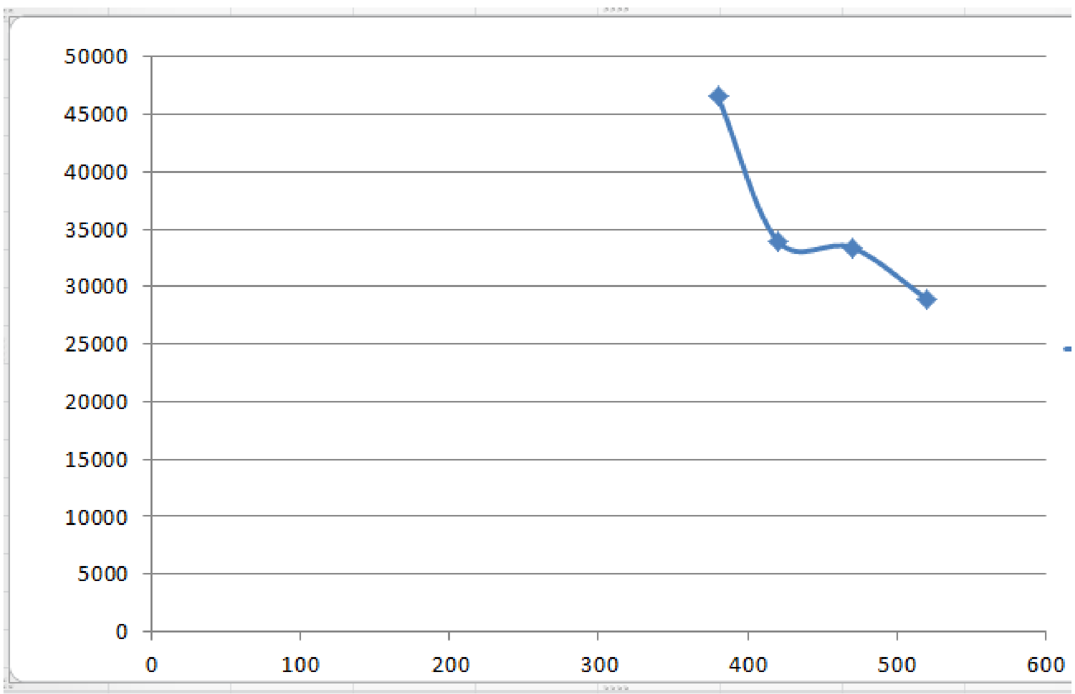
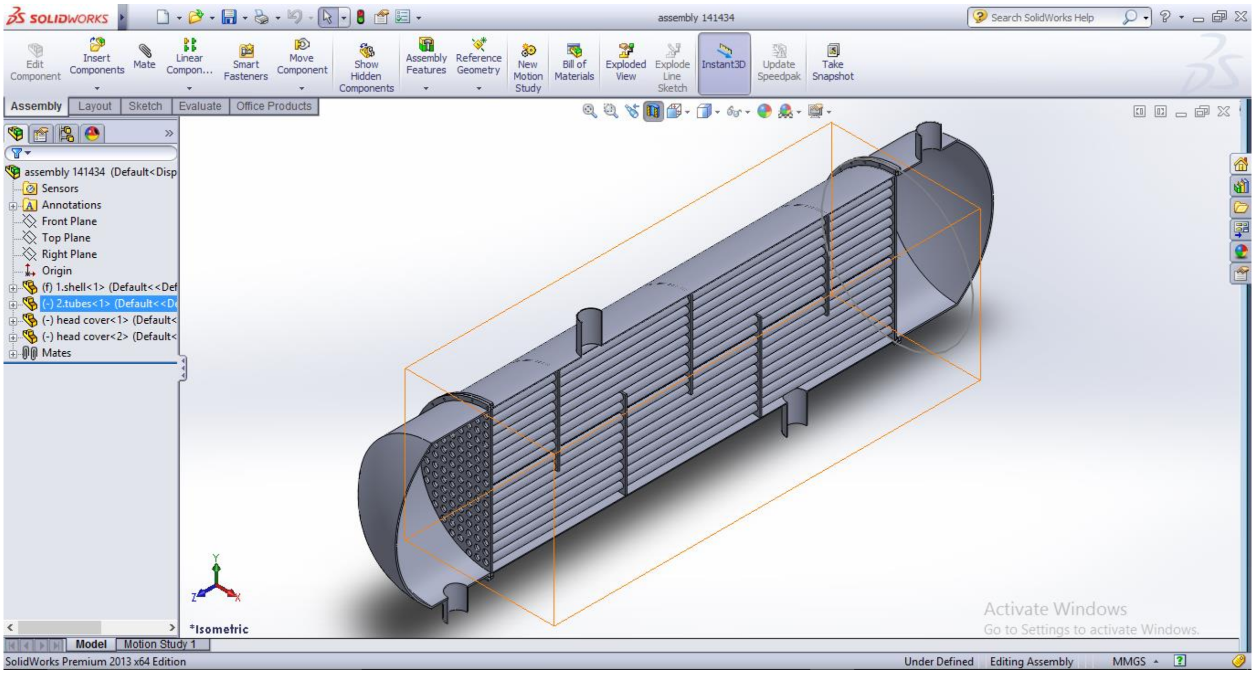


Fig: Baffle spacing vs Shell side heat transfer coefficient

In addition, higher baffle spacing makes the flow longitudinal which is less efficient than cross flow and large unsupported tube spans which will make the heat exchanger prone to tube failure due to flow-induced vibration. Reducing the baffle cut below 20% in order to increase the heat transfer coefficient or increasing it beyond 35% in order to decrease the pressure drop usually lead to poor design.The figure below is presenting the cross-sectional view of the shell and tube heat exchanger of design A. In order to achieve that, the designer should change the aspects of tube bundle geometry instead of reducing the baffle cut below 20% or increasing it beyond 35%.



the maximum and minimum values of shell side heat transfer coefficient are 46529 and 28951 W/m^2K, respectively. It's clear, with increasing baffle spacing the heat transfer coefficient decreases, and that because increasing central baffle distance means changing flow type.

The percentage decrease in the Reynolds number between design A and D is 26.58% in the turbulent region, and that percentage quite enough to change flow type. Therefore, this results in a decrease in heat transfer, and from fluid mechanics we know that the high turbulent fluid flow results in high heat transfer.

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