Mathematical Modeling and Simulation of Multi-component Condenser for the condensation of Hydrocarbon vapour mixture using MATLAB

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***ABSTRACT*: Condensers are important class of equipment in process industry. Condensation is the process of changing vapour into liquid. Design of heat exchanger to partially or totally condense a vapour mixture is frequently necessary in the process industry. The term vapour mixture actually comprehends an entire spectrum of situation. The limit of the spectrum is a mixture in which all the component present have boiling point above the maximum coolant temperature; in this case the mixture is totally condensed.(Bell & Ghaly et al).In the present work, vertical shell and tube condenser has been designed for the condensation of hydrocarbon vapour mixture containing Propane, Butane, Hexane ,Heptanes and Octane in the mole composition of 0.15, 0.25,0.05,0.30& 0.25 with the help of MATLAB using Bell & Ghaly. Heat transfer area has been calculated for the different heat load by Bell and Ghaly and Kern's design method. It is found that heat transfer surface area obtained by Kern's method is always lesser than that obtained by Bell & Ghaly's method. The reason for this is that Kern's method omits the mass transfer resistance in the vapour phase. Further, the developed code has been used to study the effect of operating variable on heat transfer area, overall heat transfer co-efficient, tube side and shell side pressure drops. The overall heat transfer coefficient was found to increase with coolant flow rate and decrease with an increase in coolant inlet temperature, vapour inlet temperature and vapour flow rate. The shell-side pressure drop was found to increase with vapour flow rate where as other variables have nominal effects. Validation of the developed code has been done using the work of S. Bandyopadhyay et al. and is be to be found in good agreement. Therefore, it may be concluded that the developed code may be used for the design of a multi-component condenser where all the components can be totally condensed.**

Keywords: **Mathematical Model, Multi- Component Condenser, MATLAB, Simulation**

INTRODUCTION

Condenser are an important class of heat transfer equipment. It is a device in which heat removed in to process of converting vapour to liquid is transferred to a coolant. Condenser in use today may fall in two categories: refrigerated or non-refrigerated. They are frequently used prior to control device (eg. in incinerators, absorber or absorber) .Refrigerated condensers are used as air pollution control device for treating emission stream with high VOC (Volatile Organic Compound).Many condensation process occurs on the outside or inside of horizontal tubes. [S.Bandyopadhyay et al., 2006]

In many process industry applications, the vapour to be condensed is multi-component in nature. Such cases include those where the single condensing vapour is mixed with an incondensable gas and also those where the condensate itself is as a multi-component condenser mixture. The presence of incondensable gases can have profound effects on condensation giving a great reduction in effective heat transfer co-efficient.

Principle of Condensation: When a hot vapour stream contacts a cooler surface, heat is transferred from the hot gases to the cooler surface. As the temperature of the vapour stream is lowered the average kinetic energy of gas molecule is reduced. Also, the volume that these vapours occupy the reduced. Ultimately the gas molecules are slowed down and crowded together to closely that the attractive force between the molecules cause them to condense to a liquid.

Two condition aid condensation: low temperature (so that the kinetic energy of the gas   molecule is low) and high pressure (so that the molecule are brought close together). The actual conditions at which a particular gas molecule will condense depend on its physical or chemical properties.

Condenser Configuration: The several possible condensers are discussed in this section

 (a)- Condenser Inside Tubes: Vertical downflow. In this type, the tube diameter normally are 19-25 mm and up to 50mm to minimize critical pressure drops.

(b)- Condenser Inside Tubes: Vertical Upflow. This mode is used to primarily for refluxing purpose when return of a hot condensate is required. A possible disadvantage is the like hood of flooding with condensate at the lower end of the tubes.

(c) Condensation Outside Vertical Tube. This type of arrangement requires careful distribution of coolant to each tube , and requires a sump and a pump for return to a cooling tower or other source of coolant. Advantage is the high coolant side heat transfer coefficient and the ease of cleaning. The free draining of condensate is a disadvantage with wide range mixtures.

(d) Condensation Inside Horizontal Tube. This mode is employed in air coolers where it is the only feasible mode. As condensation proceeds, liquid tends to build up in the tubes, then slugging and oscillating flow can occur.

(e) Condensation Outside Horizontal Tubes. This type promotes side to side flow of vapour. The tubes may be controlled partially flooded to ensure desired sub cooling of the condensate or for control of upstream pressure by regulating the rate of condensation.

Modes of Condensation.

Condensation may occur in variety of modes. The modes are described as follows.

Filmwise Condensation. Here, the condensed liquid forms a continuous liquid film on the heat transfer surface. The surface can be a flat plate or the outside or inside of a round tube. In design of condensers, it is usually assumed that the condensation will be in the filmwise mode.

Dropwise Condensation. If the heat transfer surface has poor wetting characteristic , the condenser may be formed as droplet , which are initially stationary on the surface but which are initially stationary on the surface, but which ultimately grow large enough to move across the surface under the influence of gravity or vapour shear.

Fig 1.1 Filmwise & Dropwise Condensation

Condensation Of mixture.

In many process industry application, the vapour to be condensed is multi-component in nature having different boiling point which condense over a wide temperature range, either in presence or absence of non-condensing material. The three word "multi-component vapour mixture" covers a wide range of situation. One limit of this range is one in which all component have boiling point above maximum coolant temperature; in this case the mixture can be totally condensed. The other limit is mixture in which at least one component in the initial vapour stream has a boiling point lower than the minimum coolant temperature. Example of such component includes the nitrogen and helium. An intermediate case is typified by a mixture of light hydrocarbon in which the lightest members often can not be condensed as pure components at the temperature encountered in the condenser, but instead will dissolve in the heavier components. [S. Bandyopadyay et al., 2006]

In each of the cases, the vapour mixture may form partially or completely immiscible condensate. As the composition of the vapour and liquid change throughout the condenser their physical properties vary. The heavy component must diffuse through the lighter components to reach the condensing surface. The rate of condensation will be governed by the rate of diffusion, as well as the rate of heat transfer. [Heat Exchanger design Handbook G. F.Hewitt]

Multi-Component Condenser Design

Design of a heat exchanger to a partially or totally condense vapour mixture is frequently necessary in process industries. Condensation of "mixture of vapour" covers three related situations of practical interest discussed in article 1.4. The feature common to all these cases that need to be considered in any design method are as following [Bell and Ghaly., 2006]:

Multi-component condensation is always non-isothermal. As the heavier component is preferentially condensed, the remaining mixture has a lower dew point and the temperature of condensation decrease monotonically.

2-Because the condensation is non- isothermal, there are both liquid and vapour phase sensible heat effect. The condensate already laid down must be further cooled and the remaining vapour must be cooled down to remain in near-equilibrium at the interface where further condensation is taking place. Sensible heat removal from the vapour is particularly significant in design because of the low heat transfer co-efficient commonly associated with this process.

3-Both vapour and condensate phase composition change during the condensation, raising the question of evaluation of physical properties for the mixture. This is relatively minor problem for mixture of closely related components, for example, homogeneous series, but some spectacular effect occur in condenser handling something like a boron trichloride- hydrogen mixture.

4-Since composition of both phases are continuously changing through preferential transfer of the heavier remaining component from the vapour to the condensate, there are diffusion(mass transfer) resistance in both the vapour and the liquid phase that are coupled to the heat transfer process. This part of the problem-especially for the vapour phase is so important and so formidable as to constitute the crux of the matter. The problem can be formulated in fundamental terms fairly readily, but the lack of the applicable mass transfer data and the enormity of the computational procedure for any but the most simple case render the exact solution inaccessible at this time for the design purpose

5-Since the local the thermodynamics parameter (condensing temperature, latent heat, etc) and transport rate are directly related to the local vapour and liquid composition ,it is essential that the relative flow paths of liquid and vapour through the condenser be known .For a wide condensing range vapour, this result in serious lowering   of the condensing temperature and may render the condenser incapable of meeting process specification no matter how much surface is provided.

Method for designing Heat Exchanger

Method for designing heat exchanger can be classified into two basic methods: equilibrium methods & non equilibrium method. Equilibium method such as those proposed by Kern (1971)and Bell and Ghaly (1972)and the differential or non-equilibrium methods that have been developed following the work of Colburn and Drew(1937), Colburn and Hougen(1934) and later Colburn and Edison(1941) formulated in fairly rigorous from the equation and design procedure for condensation of a pure component from a completely insoluble gas in either co-current and counter- current flow.

Kern proposed a general-purpose design method based on the equilibrium model. He suggested how to employ the nonlinear condensation temperature curve instead of the log mean temperature difference but did not discuss how to handle a multipass coolant quantitatively. Bell and Ghaly proposed an approximate generalized design method based on the equilibrium method for multi-component partial condensers. They compensated the error introduced by neglecting the mass transfer resistance by overestimating the heat transfer resistance. After Bell and Ghaly a little work has been done on the equilibrium model. To design a condenser using film theory methods requires calculations of the local heat and mass transfer rates and integrating these local rates over the condenser length using differential mass and energy balances. This was proposed by Krishna and Panchal(1977).

Literature Review

Computer Aided Design (CAD) of a Multi-component Condenser

Design of Heat Exchanger to partially or totally condense a vapour mixture is frequently necessary in process industries. Existing method for designing heat exchanger to condense a multi-component mixture are of two basic kind: Equilibrium method, such as those of Kern (1950), Silver (1947) and Bell and Ghaly (1972) and the differential method or non-equilibrium methods that have develops following the original work of Colburn and Drew (1937) , Colburn and Hougen (1934). They are both film based models in which the resistance of a series of layers are added, but the former is essentially a heat transfer model in which the mass transfer rate is approximated whereas later are provides an accurate description of heat and mass transfer processes. Several authors have discussed the problem proposed general purpose design method.

Coulburn A.P. & Hougen O.A. (1934).Outlined a method of computing the required condenser surface for condensing vapours from mixture of vapours with non condensing gases. In condensing vapours from mixture of vapours with non-condensing gases, the gas film and overall heat transfer coefficients vary widely from point to point in the apparatus, and also the change with temperature. The value of UΔt at any point in the condenser is obtained, through trial and error by equating the heat transferred by the sensible cooling of the uncondensed gas and the latent heat equivalent of the vapour transferred by diffusion and condensation.

Kern Method (1950). Kern method recommends ignoring any vapour phase resistance to heat transfer for fully condensable mixture. The full heat load ΔQ in each section of area ΔA can be expressed

$$\Delta Q = \Delta A U \Delta T$$

Where,

$$U = \frac{h^c h^l}{\left(h^c + h^l\right)}$$

Bell and Ghaly (1972) proposed an approximate generalized design for multicomponent/ partial condenser. Bell and Ghaly did not ignore the heat transfer resistance in vapour phase. They adopted a very rough approximation that the mass transfer resistance was proportional to the sensible heat transfer in the vapour phase and try to overestimate the heat transfer resistance to compensate for the error introduced by neglecting the mass transfer resistance. The sensible heat load of the vapour is defined as

$$\Delta Q_s = -V\, C_p^v \Delta T^v$$

and

$$Z = \frac{\Delta Q_s}{\Delta Q}$$

A very rough and ready approximation was made that the mass transfer resistance is proportional to the sensible heat transfer resistance in vapour phase. Therefore the effect of mass transfer resistance in vapour phase can be replaced by a conservative (high) estimate of the heat transfer resistance in the vapour. The equilibrium method is used in industry because this approach does not require diffusivity data and it is simple and rapid in computation. The deficiencies are related to the use of condensate curve based on the assumption of the equal between the two place at the bulk vapour, phase temperature.

Krishna R. & Panchal. C.B. (1977) developed simultaneous heat and mass transfer process during condensation process of binary mixture in the presence of non condensable gas is analysed using matrix formulation of interfacial mass transfer rate reaction. A film model for multicomponent mass transfer based on exact solution of Maxwell – Stefan equation is used to calculate the mass transfer coefficient in the ternary vapour phase with the aid of computational example involving condensation of methanol and water vapour in the presence of air .

Krishna R. (1979) showed problem in calculating the condensing fluxes Ni for 'n' component vapour mixture, taking amount of diffusional interaction is vapour phase. A simplified method for the flux calculation is suggested requiring the assumption that the matrix of multicomponent mass transfer coefficient. This assumption allows the calculation of the steady state flux Ni in an explicit manner without iteration.

Webb D.R. And Sardesai R.G. (1983): provided experimental evidence of the applicability of the following theories to multi component condensation: (The Krishna- Standart method ,Toor's linearised theory), An effective diffusivity model based on Colburn & Hougen . It is found that all three models adequately quantify the vapour-side diffusional process for the experimental systems and conditions investigated but the multicomponent diffusional effects can be important. Physical models of the process of multicomponent condensation can be classified into two categories:

(1) Models such as those of Silver and Bell & Ghaly which assume that the multicomponent condensation process follows the condensation curve, an approach which is extensively used in industrial practice to design multicomponent condensers. The reasons for this widespread use are simplicity, .rapidity in computation, and no requirement to know intermediate vapour compositions or to obtain diffusivity data.

(2) Models which are more physically realistic, Such models (film, penetration or boundary layer) lead to design procedures involving calculation of the local heat and mass transfer rates and their integration over the entire condenser area. The film theory of mass transfer forms the basis of this study. Multicomponent mass transfer has been described by an empirical generalisation of Fick's Law and by the Maxwell-Stefan (MS) equations. For isothermal, isobaric conditions of ideal vapours the MS equations are particularly useful..

Taylor et.al. (1986): Reviewed non-equilibrium models of multicomponent condensation with particular attention to the various ways in which the rates of condensation can be calculated. They proposed a new way of solving model equation which consists of mixed set of differential and algebraic equation. It was suggested that differential could be approximately by finite difference and the resulting set of algebraic equation solved simultaneously (using Newton Method) with the non linear equation representing the processes of interphase transport & interfacial equation.

Webb et.al (1996): made a direct comparison of the popular Silver-Bell & Ghaly (SBG) method termed the equilibrium method and non- equilibrium (rate based) methods due to Colburn and Hougen. The relationship between equilibrium methods of SBG and rate based methods of Colburn and co-worker has been studied and it is formed that SBG- method is very sustains to the Lewis number (Le = Sc/Pr) for the vapour phase. At Le =1 , the equilibrium method is unsafe in predicting the gas film heat to coefficient by upto 50%. The prediction are even less safe when Lewis number is greater than unity. Also they derived a correction factor that significantally reduces this error.

Burghardt Andrzej and Berezowski Marek :(1988) developed mathematical model has been developed which describes quantitatively the phenomena occurring during the condensation of multicomponent mixtures in which all constituents condense to form a homogeneous liquid phase. As a result of the analysis of the model equations two specific modes of the vapour mixture condensation have been defined and described: equilibrium condensation and total local condensation. The methods currently used in the design of heat exchangers in which multicomponent condensation takes place fall into two groups: the equilibrium methods employing the works of Kern, Silver and Bell and Ghaly and film method those based on the original paper by Colbum and Drew.

A. Cavallini et. Al.(2003) developed semi-empirical correlations to predict heat transfer during condensation and formed it to be quite inaccurate in some new applications, and consequently a renewed effort is made to the characterise the flow conditions and to determine associated predictive procedures for heat transfer and pressure drop of condensing vapours, even in the form of a zeotropic mixtures.

L. L. Tovazhnyansky (2004) has reviewed application of plate heat exchangers for the condensation of multi-component mixtures. A numerical simulation using semi-empirical equations of heat and mass transfer performance along the surface of plate condensers was carried out for different multi-component mixtures with non-condensable components. It is shown that the enhancement of heat and mass transfer in a plate condenser for the case of a four-component mixture gives the possibility of decreasing by 1.8–2 times the necessary heat transfer surface area comparatively with shell-and-tube unit for the same process parameters.

J. Thomas Schrodt (2004) analyzed multi-component vapour condensation in the presence of noncondensing gas using numerical solutions to a set of linearized rate equations that describe the simultaneous heat and mass transfer to the condensate film. The mass fluxes of the vapours vary with their respective mass transfer coefficients and local concentration driving forces. In the vertical tube condensers, the flux of the alcohols go through maximums. Gas phase saturation is approached as the mixtures cool

and the final traces of vapour become more difficult to condense. The techniques of solution and analysis have general application to the design of cooler-condensers in either vertical or horizontal configurations.

S. Bandyopadyay et al. (2007). In this work, a design algorithm for the condensation of a multi-component vapour mixture in shell side of a shell and tube vertical condenser has been developed using Bell and Ghaly's method. They have found that Kern's method provides a lesser heat transfer area because Kern's method does not consider the mass transfer resistance, nor does it take care of handling the sensible heat transfer during condensation. These facts have been incorporated in Bell and Ghaly's method by taking the single -phase heat transfer coefficient during operating variables viz. vapour flow rate, coolant flow rate, vapour inlet temperature, and coolant inlet temperature on the overall heat transfer coefficient, and shell side pressure drop have been studied for the wide range of parameters

M.M Awad et Al.(2010) examined Lockhart–Martinelli method for predicting two phase flow pressure drop is examined from the point of view of asymptotic modeling. Comparisons are made with the Lockhart–Martinelli method, the Chisholm method, and the Turner–Wallis method. An alternative approach for predicting two phase flow pressure drop is developed using superposition of three pressure gradients: single phase liquid, single phase gas, and interfacial pressure drop. This new approach allows for the interfacial pressure drop to be easily modeled for each type of flow regime such as bubbly, mist, churn, plug, stratified, and annular, based on the classical laminar-laminar, turbulent-turbulent, laminar-turbulent, and turbulent-laminar flow regimes proposed by Lockhart and Martinelli.

Dr. Sattar Jaber Habeeb Al-Jabair et al (2012) .The work is performed to evaluate the influence of the tube diameter, coil pitch, shell-side and tube-side mass flow rate, and inlet temperatures of tube-side over the axial temperature distribution of heat exchanger, effectiveness, modified effectiveness and heat transfer coefficient. The evaluation has been performed for the steady-state . The results indicate that the major effect on the axial temperature distribution of heat exchanger was the mass flow rate ratio (mr). Also the modified effectiveness decreased with increasing mass flow rate ratio. The main influence on the shell side heat transfer coefficient was coil pitch. Developed an equation to correlated, Nusselt Number as function of Reynolds Number, Prandtl Number, Dean Number and helical coil Number.

Brian M. Fronk & Srinivas Garimella (2013) revealed that the condensation of multi-component mixtures in tubes is critical to the function of many energy conversion processes. This paper reviews experimental and analytical studies of the coupled heat and mass transfer phenomena during condensation of different fluid mixtures in different in-tube geometries. Experimental studies on condensation of binary and higher order mixtures in vertical and horizontal smooth and enhanced tubes are discussed. Four categories of models are reviewed (1) Conservation equation (2) Non-equilibrium film theory (3) Equilibrium and (4) Empirical models.

Yusuf Ali Kara(2014). developed a computer code based on a simplified model for sizing a horizontal shell and tube refrigerant condenser. The model uses three-zone approach for condensing-side and overall approach for the coolant side of the condenser. The model has been experimentally validated by testing a shell-and-tube refrigerant condenser that water flows on tube-side as coolant while R-134a as refrigerant condenses on shell-side.

Mathematical Model : Model based on the equilibrium method as proposed by Bell and Ghaly has been considered for the multi-component condenser design. Vertical shell and tube condensers with condensation at shell side is considered because of large industrial practice and its suitability towards equilibrium method. Equations consists of heat balance, rate equation which is useful for determining the heat transfer area and expression for overall heat transfer coefficient. Multi–component condensation takes place over a wide temperature range that is from dew point to bubble point. Therefore, it is necessary to determine the bubble point and dew point of the mixture to be condensed before starting the design calculations.

Dew and Bubble point temperature.

The dew point of vapour corresponds with the onset of condensation and the bubble point corresponds with total condensation.

At Dew point

$$\sum_{i=1}^{n} x_i = 1$$

At Bubble point

$$\sum_{i=1}^{n} y_i = 1$$

At any temperature intermediate to the dew and bubble point, the compositions of the equilibrium liquid x_i and y_i can be determined by vapour liquid equilibrium consideration:

$$Z_{F,i} = y_i + (1 - \theta)x_i$$

Where, $Z_{F,i}$ is the vapour feed composition.

From equation following equations are obtained for calculating the vapour and liquid mixture's composition.

$$x_i = \frac{z_{F,i}}{1+\theta(K_i-1)} \quad \text{and}$$

$$y_i = \frac{K_i z_{F,i}}{1+\theta(K_i-1)}$$

Assumptions :

1. The design method is based on the following assumptions:
2. The liquid and vapour compositions are in equilibrium at the vapour bulk temperature.
3. Liquid and vapour enthalpies are those of the equilibrium phases at the vapour bulk temperature.
4. The sensible heat of the vapour is transferred from the bulk vapour to the vapour-liquid interface by a convective heat transfer process. The heat transfer coefficient is calculated from a correlation for the geometry involved, assuming only the vapour phase is present and using physical properties of vapour and local vapour flow rate.
5. The total latent heat of condensation and sensible heat of the cooling condensate are transferred through the entire thickness of the liquid film.
6. Fouling factor is negligible. No slip condition is present.

Design Equation. [19]

Heat transfer area :   The basic equation for calculation of the heat transfer,

$$Q = U_m A_0 \Delta T_m$$

Wall

Inside Dirt Film Coolant

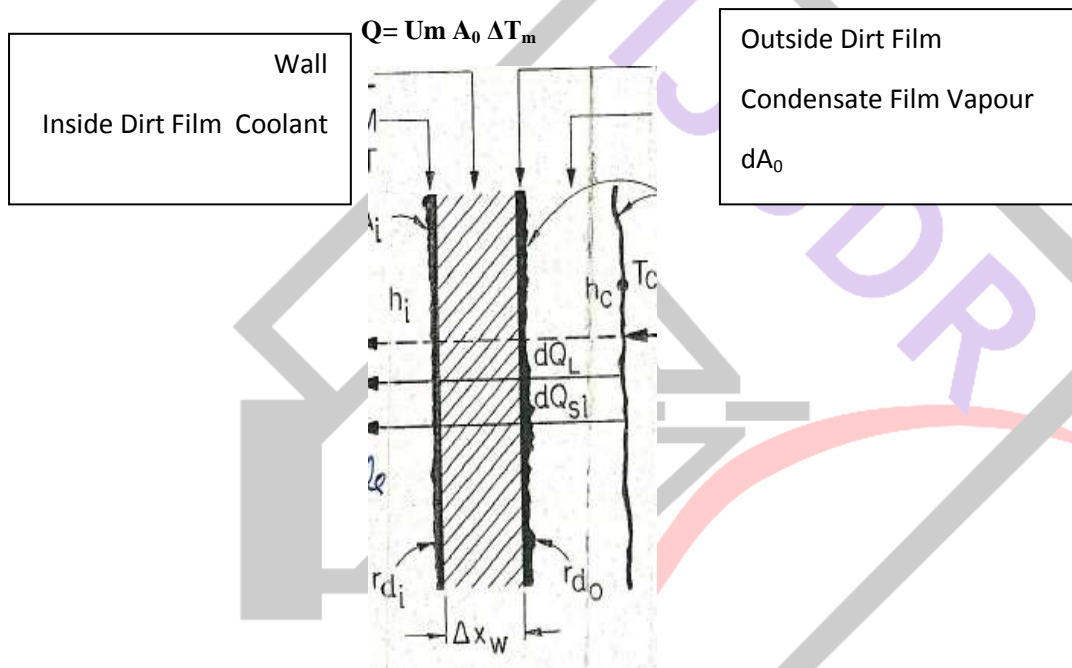Outside Dirt Film

Condensate Film Vapour

dA_0

Fig : Idealized model of multi component condensation

During condensation, there are three types of heat loads to be accounted for total heat load(figure 3.1). These are latent heat load, liquid sensible heat load, and vapour sensible heat load.

$$\frac{dQ_T}{dA_0} = \frac{dQ_{sv}+dQ_L+dQ_{sl}}{dA_0}$$

$$= U'_0 \, (T_c - t)$$

Where,

$$U'_0 = \frac{1}{\frac{1}{h_i}\left(\frac{dA_0}{dA_i}\right)+r_{di}\frac{dA_0}{dA_i}+\frac{\Delta x_w}{k_w}\left(\frac{dA_0}{dA_i}\right)+r_{do}+\frac{1}{h_c}}$$

The heat flux for the sensible heat removal from the vapour to the vapour-liquid interface is given as follows:

$$\frac{dQ_{sv}}{dA_0} = h_{sv} \, (T_v - T_c)$$

where, hsv , is calculated from a correlation for gas phase heat transfer for the geometry involved.

$$\frac{dQ_T}{dA_0} = U_0'\left[T_v - \frac{1}{h_{sv}}\left(\frac{dQ_{sv}}{dA_0}\right) - t \right]$$

Now define Z as the ratio of the local vapour sensible heat flux to the local heat flux

$$\frac{dQ_T}{dA_0} = \frac{U_0'\,(T_v - t)}{1 + \frac{ZU_0'}{h_{sv}}}$$

Substituting value of Z in equation the differential form of the design equation is obtained,

$$Z = \frac{dQ_{sv}/dA_0}{dQ_T/dA_0} = \frac{dQ_{sv}}{dQ_T}$$

Integration of equation yields the following equation:

$$\int_0^{A_0} dA_0 = \int_0^{Q_T}\left(\frac{1 + \frac{ZU_m}{h_{sv}}}{U_m(T_v - T)}\right) dQ_T$$

The overall heat transfer coefficient for multi-component condensation Umc is calculated by the following equation:

$$\frac{1}{U_{mc}} = \frac{1}{U_m} + \frac{Z}{h_{sv}}$$

If the assumption is taken that Um and hsv depend only upon the local vapour side condition, the generalized design equation for the multi-pass can be written as follows:

$$\int_0^{A_0} dA_0 = \int_0^{Q_T}\left(\frac{1 + \frac{ZU_m}{h_{sv}}}{U_m(T_v - T)}\right)\frac{dQ}{(T - T_m)}$$

$$\text{Where}, \ T_m = \sum_{k=1}^{N_{tp}}\frac{T_c^k}{N_{tp}}$$

The term (T- Tm) within the integration limit is the mean temperature between the hot vapour and the coolant at any position.

4.1 Validation of the developed Code

To Check the credibility of the developed code validation study has been carried out. For this, work done by S. Bandyopadhyay et. al. has been considered. Composition of hydrocarbon vapour mixture considered is as follows:

Table 4.1 Mass fraction and Boiling point of component [19]

Component	Mass Fraction	Boiling Point (K)
Porpane	0.15	231
Butane	0.25	272.5
Hexane	0.05	342
Heptane	0.30	371
Octane	0.25	398

condensation of the mixture vertical shell and tube heat exchanger, with condensation at shell side has been considered. The operating conditions are given below.

Table 4.2 Operating Variable of Condenser :

Operating Variable	Value
Vapour Flow Rate q_{ma}/kgs^{-1}	12.00
Vapour Inlet Temperature $T_{i,v}$/K	430.00
Vapour Outlet Temperture $T_{o,v}$/K	320.00
Coolant Inlet Temperature T_c/K	293.00
Coolant Flow Rate, q_{mc}/kg.s^{-1}	298.00
Pressure, P/kPa	506.66

To see the effect of these operating variable on overall heat transfer coefficient, shell side pressure drop, tube side pressure drop and heat transfer area, the range considred is given in Table.4.3.

Table 4.3 Range of variable :

Table : Range of Variable				
Variable	Vapour Flow Rate (q_m/kg s^{-1})	Vapour inlet Temperature (T_i /K)	Coolant flow rate (q_m/kg s^{-1})	Coolant inlet Temperature (T_i /K)
Range	8-18	430-480	150-600	287-308

Table 4.4 Geometrical variable of a condenser:

Geometrical Variables:	
Tube side diameter d_o,mm	25.4
Tube side inside diameter d_i,mm	19.86
Length of tube , l,m	2.384
Pitch arrangement	Equilateral
Pitch	Triangular Pitch
Number of tube, N	2
Baffle type	25% Cut, Segmental Baffle
Baffle Spacing, m	Ds/5
Tube sheet thickness	54
Tube roughness, mm	0.0016

Analysis of Result

The result of simulation are shown in table 4.5.There is comparison of output variable by S. Bandyopadhyay and present work is given below by Bell & Ghaly method:

Table 4.5 Output variable of the condenser:

Output Variables	S.Bandyopadhyay et al(2006)	Present work
Dew point of vapour mixture ,T(K)	429.72	410.23
Bubble point of vapour mixture ,T(K)	330.28	312.25
Total Heat load, Q	5885.05	6022.35
Weighted mean temperature difference, ΔT,K	77.0	75
Overall Heat Transfer Coefficient, U , $Wm^{-2}K^{-1}$	546.10	534.07
Heat Transfer Surface Area,m^2	139.82	150.35
Total number of tubes	752	791
Tube side pressure drop Δp, Pa	0.1677×10^5	02365×10^5
Shell Side pressure drop Δp, Pa	0.1123×10^4	0.235×10^4

Table 4.5 shows that dew point of the various mixture are obtained is 410.23K and the bubble point is 312. .therefore condensation range of the given hydrocarbon mixture is 410.23K to 312.25 K . This range has been divided into the five interval to obtain the heat load , weighted mean temperature difference and heat transfer surface. It can be seen that the result are good agreement with the published work.

Further the developed code has been used to study the effect of operating variable (like vapour flow rate ,vapour inlet temperature etc.) on heat transfer area, overall heat transfer area, tube side and shell side pressure drop. The result are as follows:

Effect of cumulative heat load on heat transfer surface area:
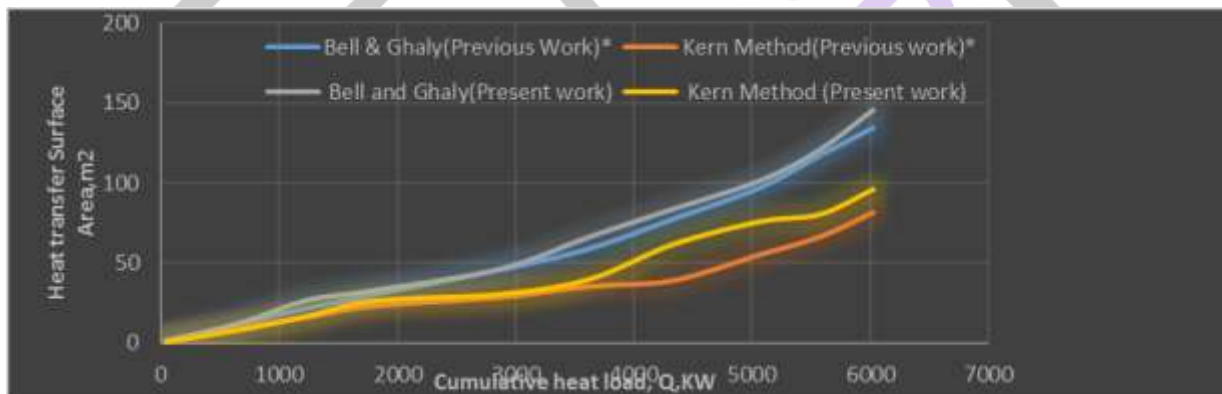

Fig (4.1) shows the variation of heat transfer surface area with various cumulative heat loads for condensation of hydrocarbon vapours. The surface area of the condenser has been calculated by Kern and Bell and Ghaly methods. It is observed that heat transfer area predicted by the Kern method at different heat load is always less then Bell & Ghaly method. There is diffusion mass transfer resistance along with heat transfer resistance during multi-component condensation. In Kern's method, the effect of mass transfer is not considered. Hence, Bell and Ghaly's method is more reliable and used for the design. The results obtained for the heat transfer surface area are in good agreement with the published result of S. Bandyopadhyay et al (2006).The maximum absolute difference between the present work and the published work for heat transfer area is 8% by Kern method and 5% by Bell and Ghaly method. This signifies that the developed code can be used to design the condenser for mixture having boiling point above the maximum coolant temperature the case where mixture can be totally condensed.

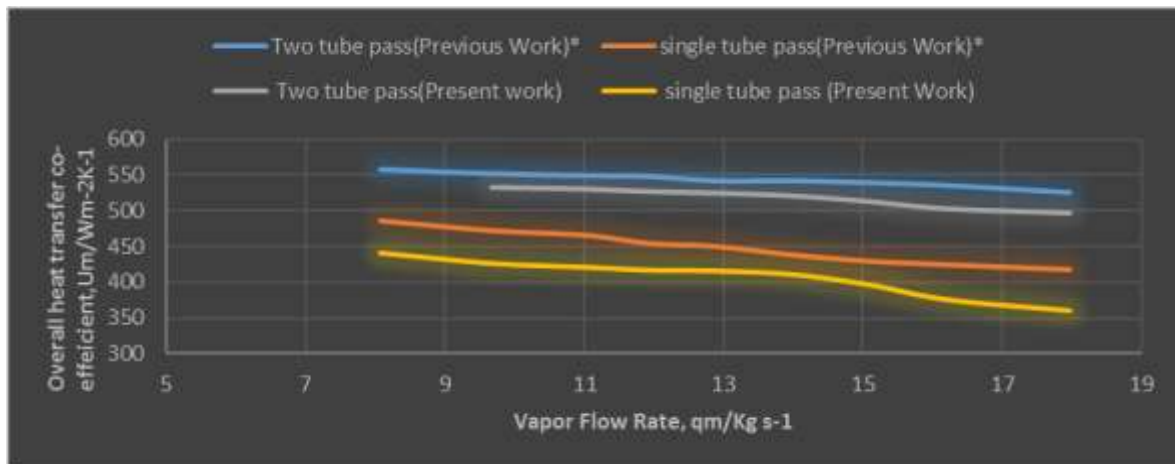**4.2.2 Effect of vapour flow rate (q$_m$) on overall heat transfer co-efficient(U):**

Fig. (4.2) shows the variation of the overall heat transfer coefficient with the vapour flow rate for single and two tube pass condensers. It can be observed that the overall heat transfer coefficient decreases with the increase in vapour flow rate, irrespective of the tube pass. We know that, if the flow increases, the heat transfer coefficient also increases due to increase in turbulence.. Therefore, the overall heat transfer coefficient decreases. For any given flow rate, the two-tube pass condenser gives a higher overall heat transfer coefficient .The trend obtained in the present work is in good agreement to the published work of S.Bandyopadyay.

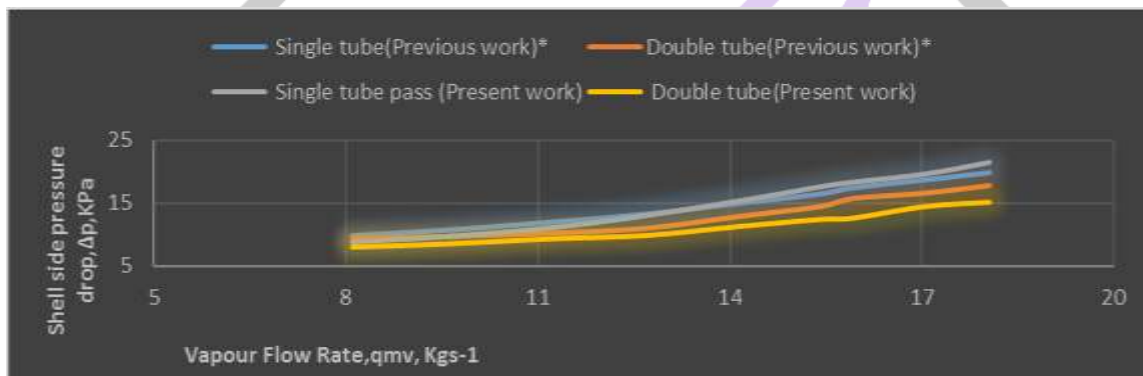**Effect of vapour flow rate on shell side pressure drop(Δp):**

Fig. (4.3) shows the variation of the shell-side pressure drop with the vapour flow rate for a single and two-tube pass condenser. As expected, the shell-side pressure drop increases with the increase in vapour flow rate, irrespective of the tube pass arrangement. Moreover, for a given value of vapour flow rate the two-tube pass condenser gives a lesser pressure drop than the single- tube pass. In any given geometry, to obtain more flow through a restriction (condenser is a restriction to vapour flow), one must have more pressure upstream of the restriction, i. e. pressure drop in the condenser increases. The trend obtained is in good agreement with the published work of S.Bandyopadhyay(2006).

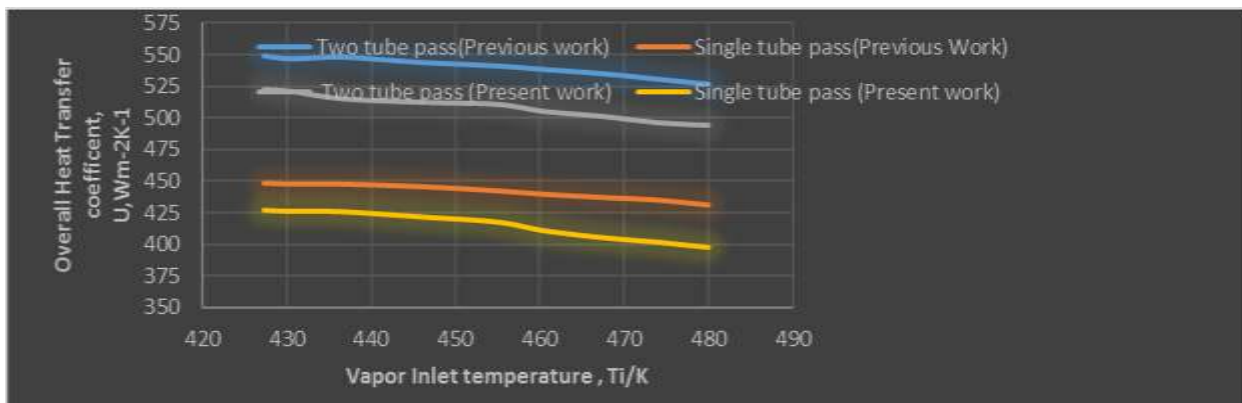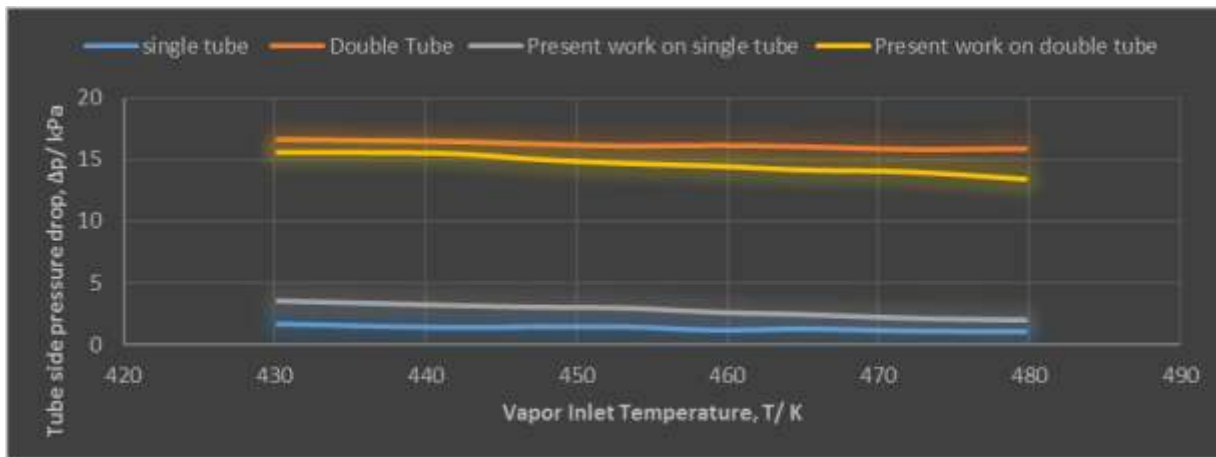**Effect of vapour inlet temperature(T$_i$) on overall heat transfer co-efficient(U):**

Fig. (4.4) shows the variation of vapour inlet temperature on overall heat transfer coefficient for a single and two-tube pass condenser. It can be observed that increase in vapour inlet temperature decreases the overall heat transfer coefficient, irrespective of the number of tube passes. Also, for a given vapour inlet temperature, the two-tube pass condenser provides a higher overall heat transfer coefficient. In general, the condensation, overall heat transfer coefficient depends upon factors such as condenser type, layout, surface geometry, heat load and thermo-physical properties of the condensing vapour. Since the change in inlet vapour temperature alters the thermo-physical properties of vapour, therefore the behaviour of vapour density and vapour dynamic viscosity will change in temperature. With an increase in vapour temperature, the density of vapour decreases and dynamic viscosity increases. It can be seen from Nusselt equation, that an increase in inlet vapour temperature will adversely affect the heat transfer coefficient. The trend obtained in the present work is in good agreement with the published work of S.Bandyopadhyay (2006).

Effect of vapour inlet temperature on tube side pressure drop:

Graph shows the variation of the tube-side pressure drop with the vapour inlet temperature for a single and two-tube pass condenser. This graph shows that an increase in vapour inlet temperature decreases the tube-side pressure drop. This is because the increase in vapour inlet temperature increases the heat-transfer surface area, and thereby the tube-side pressure drop. The trend obtained is in good agreement with the published work S.Bandyopadhyay(2006).

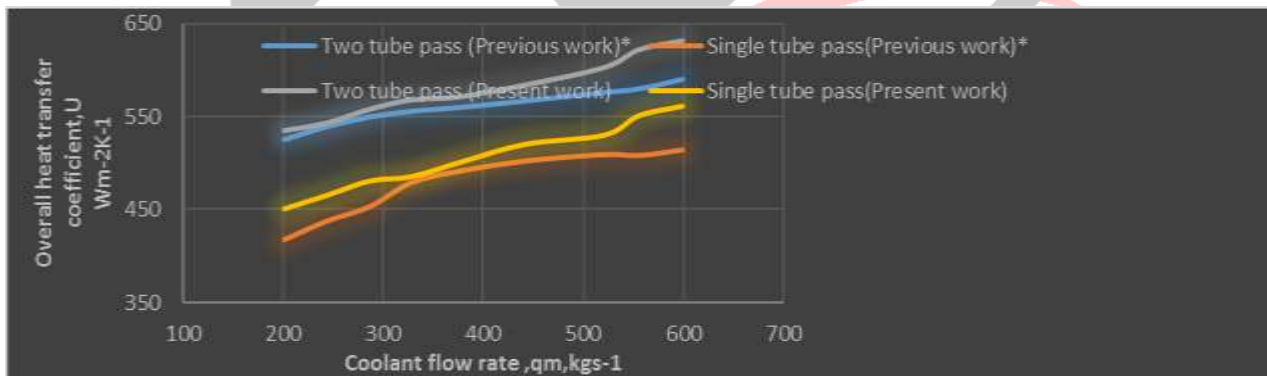**Effect of coolant flow rate on overall heat transfer co-efficient:**

Fig shows the variation of the overall heat transfer coefficient with the coolant flow rate for a single and two-tube pass condenser. . It is observed that increase in the coolant flow rate increases the overall heat transfer coefficient irrespective of tube pass arrangement. Further it can be seen that, for a given coolant flow rate, the two-tube pass condenser offers higher overall heat transfer coefficients.. The reason for behaviour observed is that increase in flow rate decreases the heat transfer area and thereby increases the overall heat transfer coefficient. The results obtained for coolant flow rate on overall heat transfer coefficient for single and two-tube pass condensers condenser is in good agreement with the published result of S. Bandyopadhyay et al (2006).

4.2.7 Effect of coolant flow rate(q_m) on heat transfer surface area(A) :
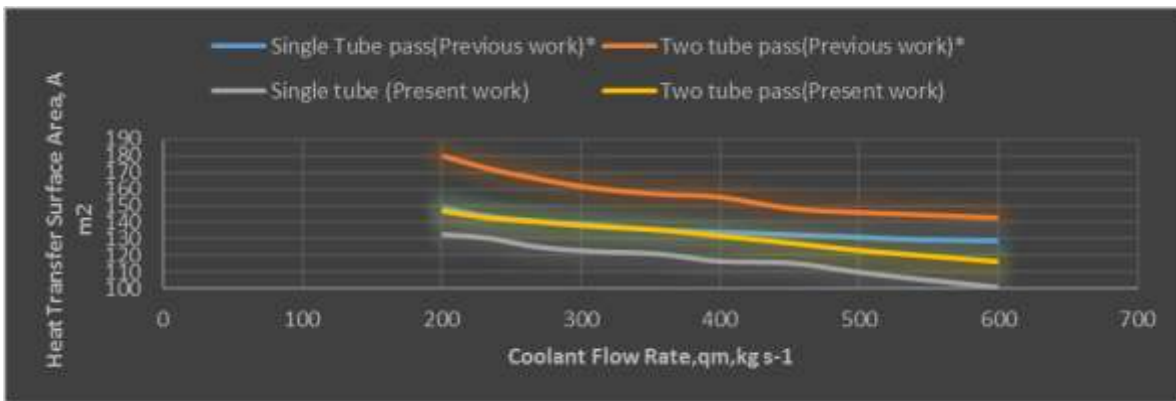

Fig. (4.7) is a plot between the heat transfer area for the coolant flow rate for single and two-tube pass condensers. This figure reveals that the increase in coolant flow rate decreases the heat-transfer surface area irrespective of tube pass arrangement. This is due to the fact that the increase in coolant flow rate decreases the heat transfer coefficient and thereby the surface area of the condenser. The results obtained for coolant flow rate on heat-transfer surface area for single and two-tube pass condensers condenser is in good agreement with the published result of S. Bandyopadhyay et al (2006).The maximum absolute difference between the present work and the published work for heat for coolant flow rate on heat-transfer surface area for single tube pass is 13% and two-tube pass is 17%.

4.2.8 Effect of coolant inlet temperature(T_i) on overall heat transfer co-efficient:
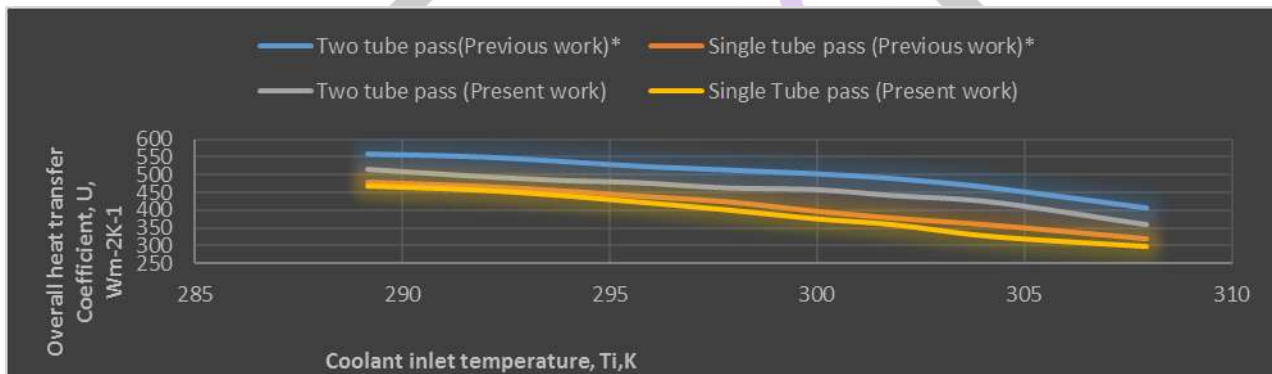

Fig 4.8 shows the variation of the overall heat transfer coefficient with the coolant inlet temperature for single and two-tube pass condensers. This figure reveals that the increase in coolant inlet temperature decreases the overall transfer coefficient irrespective of the tube pass. This is because the increase in coolant inlet temperature reduces the weighted mean temperature difference and increases the heat transfer area, thereby reducing the overall heat transfer coefficient. The condensation overall heat-transfer coefficient depends upon factors such as condenser type, layout, surface geometry, heat load and thermo-physical properties of the condensing vapour. The change in coolant inlet temperature change the vapour temperature, which alters the thermo-physical properties of vapour. The results obtained coolant inlet temperature and overall heat transfer in good agreement with published result of S. Bandyopadhyay et al(2006).The maximum absolute difference between the present work and the published work for single tube pass is 9% and two tube pass is 4%.

 Conclusion

Based on the results presented and discussed in the previous chapter following conclusions have been made: The code has been developed in MATLAB for the design of a multi-component condenser, based on the equlibirium method of Bell and Ghaly (1972). This program has been tested for a wide range of operating variable and the following   result are obtained:

- Heat transfer area has been calculated for the different loads by Bell and Ghaly and Kern's design method. It is formed that heat transfer surface area obtained by Kern's method is always lesser than that obtained by Bell & Ghaly's method. The reason for this is that Kern's method omits the mass transfer resistance in the vapour phase.

- The overall heat transfer coefficient was found to increase with coolant flow rate and decrease with an increase in coolant inlet temperature, vapour inlet temperature vapour flow rate.

- The pressure drop in the tube side was found to increase with an increase in coolant flow rate, and decrease with an increase in vapour flow rate and coolant inlet temperature.

- The shell-side pressure drop was found to increase with vapour flow rate where as other variables have nominal effects.

Validation of the developed code has been done using the work of S. Bandyopadhyay et al. (2006) and is to be found in good agreement. Therefore, it may be concluded that the developed code may be used for the design of a condenser for the multi-component mixture ; where all the components have the boiling point above the maximum coolant temperature that is the case where mixture can be totally condensed

Recommendations for Future Work

Following recommendations are suggested for the future work:

- On the basis of present work we recommend that while considering this topic in future one can perform variation in vapour flow rate, vapour inlet temperature with tube length, tube diameter.

- In future the model can be also extended for the mixture containing non-condensable gases.

NOMENCLATURE

Notations

A_o	Heat transfer area based on outside tube area, m^2.
A_i	Heat transfer area based on inside tube area, m^2.
b_s	Baffle Spacing.
C_p	Thermal Conductivity
D_o & D_i	Outer and inner diameter of tube
D_b	Bundle Diameter, m
d_e	Equivalent Diameter, m
d_s	Shell diameter,
dQ_L, dQ_{sl}, dQ_{sv}	Latent differential value of latent heat , liquid sensible heat load and vapour sensible heat load
h_i	Heat transfer coefficient inside the tube, $Wm^{-2} K^{-1}$.
h_o	Heat transfer coefficient outside the tube, $Wm^{-2} K^{-1}$.
h_{sv}	Sensible heat transfer coefficient, $W m^{-2} K^{-1}$.
K	Vapour – liquid equilibrium constant.
m_l & m_g	Liquid and gas phase mass flow rates
N	Number of components.
P_{tr}	Triangular pitch, m
Q.	Heat Load.
Q_m	Vapor mass flow rate, $kg\ s^{-1}$.
R	Gas Constant
R_{di}	Fouling resistance inside the tube, $m^2\ K\ W^{-1}$
R_{do}	Fouling resistance outside the tube, $m^2\ K\ W^{-1}$
T.	Temperature, K
T_l	Liquid phase temperature, K
T_m	Mean temperature, K

ΔT_m	Mean temperature difference between hot and cold fluid streams,
T_v	Vapour phase temperature, K
T_c	Coolant temperature, K
U_m	Mean overall design heat-transfer coefficient m^{-2} K^{-1}
U_{mc}	Overall heat-transfer coefficient for multi-component condenser,
V_g & V_l	Specific molar volume of gas and liquid.
X	Mole fraction in liquid phase, 1
Y	Mole fraction in vapour phase, 1
Z_F	Feed molar composition

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