International Journal of Engineering and Technical Research (IJETR)
ISSN: 2321-0869 (O) 2454-4698 (P) Volume-9, Issue-5, May 2019

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

Prakhar Mishra, Ravendra Singh

***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.**

In the case of multi-component condensation the 'condensing vapour contains a mixture of components having different boiling points, which condense over a wide temperature range, either in presence of or absence of non-condensing material. 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. Based on this algorithm, an in-house computer code has been developed. This code was used for the design of the condenser for the condensation of a hydrocarbon vapour mixture. A code was also developed for the Kern's method for the condenser design. It was 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 one-phase heat transfer coefficient during vapour sensible heat transfer. The effects of 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. The results are useful for the design of multi-component condensation.

***Index Terms*—CAD, multi-component condensation, design, vapour, heat transfer**

I. INTRODUCTION

A condenser is a device in which the heat removed in the process of converting a vapour to liquid is transferred to a coolant. An indirect or surface condenser has a thin wall separating the coolant from the vapour and its condensate; the heat passes through this wall. The surface used in indirect condensers may be plates or tubes and these surfaces can be plain, extended with fins, enhanced by passive or active augmentation techniques. The physical arrangement of the surfaces can take many forms and affects the two-phase flow patterns of the vapour-condensate mixture and the flow pattern of the coolant, thus influencing the heat transfer rates.

The shell side condensation plays an important role in a variety of engineering applications including electric power, refrigeration, and chemical process industries. Nowadays, significant insight has been gained into the two-phase flow pattern and heat transfer phenomenon that occurs on the shell side of a surface condenser.

In the case of multi-component condensation, the 'condensing vapour' contains a mixture of components having different boiling points, which condense over a wide temperature range, either in presence or absence of non-condensing material. The three words 'multi-component vapour mixture' cover a wide range of situations. One limit of this range is one in which all components have boiling points above maximum coolant temperature; in this case the mixture can be totally condensed. The other limit is a mixture in which at least one component in the initial vapour stream has a boiling point lower than the minimum coolant temperature and, also is negligibly soluble in the liquid condensate formed from the other components and hence cannot be condensed at all. An intermediate case is typified by a mixture of light hydrocarbons in which the lightest members often cannot be condensed as pure components at the temperature encountered in the condenser, but instead will dissolve in the heavier components. In each of these cases, the vapour mixture may form partially or completely immiscible condensate.

Existing methods for designing heat exchangers to condense multi-component mixtures can broadly be classified into two basic methods: equilibrium methods, such as those proposed by Kern,[1] and Bell and Ghaly[2] and the differential or non-equilibrium methods that have been developed following the work of Colburn and Drew [3] Colburn and Hougen [4] and later Colburn and Edison [5] formulated in fairly rigorous form the equations and design procedure for condensation of a pure component from a completely insoluble gas in either co-current or counter-current flow.

Kern1 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 [2] 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 [2] 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.

Prakhar Mishra, Mechanical Engineering, M.Tech Student, Department of Mechanical Engineering, Faculty of Engineering & Technology, Rama University, Kanpur, India.

Mr. Ravendra Singh, Assistant Professor, Department of Mechanical Engineering, Faculty of Engineering & Technology, Rama University, Kanpur, India

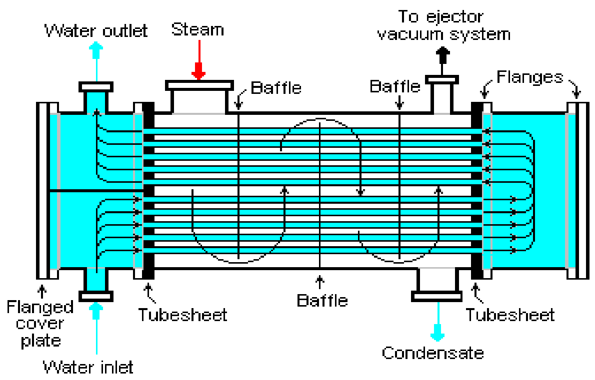


Figure 1: Basic Components of Condenser [3]

This was proposed by Krishna and Panchal. [6] In this work, an in-house CAD algorithm of a multi-component condenser based on the equilibrium model has been developed to investigate the effects of operating variables viz. vapour flow rate, coolant flow rate, vapour inlet temperature and coolant inlet temperature on the weighted mean temperature difference, the heat transfer area, overall heat transfer coefficient, and shell side pressure drop of condenser.

II. CONDENSERS

As already mentioned, condenser is an important component of any refrigeration system. In a typical refrigerant condenser, the refrigerant enters the condenser in a superheated state. It is first de-superheated and then condensed by rejecting heat to an external medium. The refrigerant may leave the condenser as a saturated or a sub-cooled liquid, depending upon the temperature of the external medium and design of the condenser. Figure 1.11 shows the variation of refrigeration cycle on T-s diagram. In the figure, the heat rejection process is represented by 2-3'-3-4. The temperature profile of the external fluid, which is assumed to undergo only sensible heat transfer, is shown by dashed line.

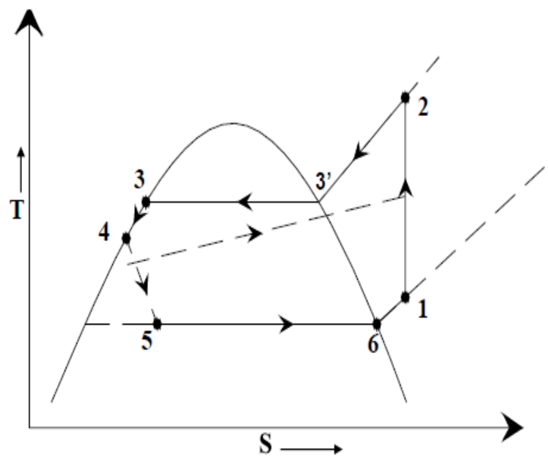


Figure 2: Refrigeration cycle on T-s diagram

It can be seen that process 2-3' is a de-superheating process, during which the refrigerant is cooled sensibly from a temperature T_2 to the saturation temperature corresponding condensing pressure, T_3 Process 3'-3 is the condensation process, during which the temperature of the refrigerant remains constant as it undergoes a phase change process. In actual refrigeration systems with a finite pressure drop in the condenser or in a system using a zeotropic refrigerant mixture, the temperature of the refrigerant changes during the condensation process also. However, at present for simplicity, it is assumed that the refrigerant used is a pure refrigerant (or an azeotropic mixture) and the condenser pressure remains constant during the condensation process. Process 3-4 is a sensible, sub cooling process, during which the refrigerant temperature drops from T_3 to T_4.

III. LITERATURE REVIEW

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 $U\Delta t$ 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.

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. 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 places at the bulk vapour, phase temperature.

D.R. Webb (2002): This paper makes a direct comparison of the popular silver – bell method, termed as equilibrium method and the film methods due to Colburn and Hougen. the silver approach is more commonly used in design of multicomponent condenser. With unspecified mixtures, where only a cooling curve is available, it must be used. Its advantage is that no further vapour-liquid equilibrium calculations are needed, irrespective of geometry. It has been corrected and improved by use of more physically realistic film model. but it is shown above that substantial discrepancies between the methods can occur, Even when all published correction factors are included. The success of the Silver method in industrial practice must be ascribed to the fact that industrial design is extremely conservative. The two methods are not in agreement at a Lewis number of unity, but at a value somewhat below, typically 0.60-0.80. At ***Le*** = 1 the equilibrium method is unsafe in predicting the gas film heat transfer coefficient by up to 50%. As Lewis number increases above unity the equilibrium method is increasingly unsafe.

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.

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

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.

IV. Multi-Component Condenser Design

To design a multi-component condenser, one requires a suitable method to evaluate the overall heat transfer coefficient, which is useful for determining the heat transfer area. Evaluation of the overall heat transfer coefficient needs suitable equations to predict the heat transfer rates i. e. heat transfer coefficients. In this work, the equilibrium method proposed by *Bell* and *Ghaly* [2] was considered for predicting the surface area required for heat transfer because of its robustness, speed, and reliability.

Condensation at shell side, in a vertical shell and tube heat exchanger was considered because of its large industrial practices and suitability towards the equilibrium method. For the tube arrangement, square, triangular and rotated square layouts were considered. Multi-component condensation takes place over a wide temperature range – from dew to bubble point. Therefore, it is necessary to determine the dew and bubble points before starting design calculations.

V. Numerical Analysis

Dew and Bubble Point Temperature

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

At dew point:

$$\sum_{i=1}^{n} x_i = 1 \quad (1)$$

At bubble point:

$$\sum_{i=1}^{n} y_i = 1 \quad (2)$$

At any intermediate temperature, the vapour fraction must satisfy the following equation:

$$z_{F,i} = y_i + (1-\theta)x_i \quad (3)$$

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

From eq. (3.19) the following equations are obtained for calculating the vapour and liquid mixture's composition.

$$x_i = \frac{Z_{F,i}}{1+\theta(K_i-1)} \quad (4)$$

$$x_i = \frac{K_i Z_{F,i}}{1+\theta(K_i-1)} \quad (5)$$

The shell-side heat-transfer coefficient was obtained from the correlation taken from the Heat Exchanger Design Handbook [7] while the pressure drop was obtained from the Martinelli equation [8]. The tube-side heat-transfer coefficient was obtained from the correlation given by Kutaleladze and Borishanskii [9] and pressure drop from the correlation taken from Coulson and Richardson. [10]

Heat transfer area

The basic equation for calculation of the heat transfer area is

$$\Phi_T = U_m A_o \Delta T_m \quad (6)$$

Where, U_m is the mean overall heat transfer coefficient, A_o is the total heat exchanger area, and ΔT_m is the mean temperature difference between the hot and the cold fluid stream.

The nature of the heat release curve in multi-component condensation is not linear. Therefore, it is unrealistic to take the mean temperature difference and mean overall heat-transfer coefficient over the whole exchanger. Here, the calculation was carried out at each point and then the results were integrated to obtain the total exchanger area. During condensation, there are three types of heat loads to be accounted for total heat load. These are latent heat load, liquid sensible heat load, and vapour sensible heat load.

$$\frac{d\Phi_T}{dA_o} = \frac{d\Phi_{sv} + d\Phi_p}{dA_o} = U_m(T_p - T_c) \tag{7}$$

Where,

$$U_m = \frac{1}{\frac{dA_o}{h_i dA_i} + R_{di}\frac{dA_o}{dA_i} + \frac{x_w}{k_w dA_i}R_{do} + \frac{1}{h_o}}$$

Where subscript 'i' and 'o' stand for the inside and outside of the tube, respectively,

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

$$\frac{d\Phi_{sw}}{dA_o} = h_{sw}(T_v - T_1) \tag{8}$$

Substitution of T_1 from eq. (7) into eq. (8) gives:

$$\frac{d\Phi_T}{U_m(T_v - T_c)} = \frac{dA_o}{1 + \frac{ZU_m}{h_m}} \tag{9}$$

Where, $Z = \frac{d\Phi_W}{d\Phi_T}$

Integration yields the following equation:

$$\int_0^{A_o} dA_o = \int_0^{\Phi_T}\left(\frac{1 + \frac{ZU_m}{h_{sv}}}{U_m(T_v - T)}\right) d\Phi_T \tag{10}$$

The overall heat transfer coefficient for multi-component condensation U_{mc} is calculated by the following equation:

$$\frac{1}{U_{mc}} = \frac{1}{U_m} + \frac{Z}{h_{sv}} \tag{11}$$

If the assumptions are taken that U_m and h_{sv} depend only upon the local vapour side condition, the generalized design equation for the multipass can be written as follows:

$$A_o = \int_0^{\Phi_T}\left(\frac{1 + \frac{ZU_m}{h_{sv}}}{U_m(T_v - T)}\right)\frac{d\Phi}{(T - T_m)} \tag{12}$$

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

The term (T- T_m) with in the integration limit is the mean temperature between the hot vapour and the coolant at any position. The integration is carried out numerically and the zones are insufficiently small, hence the logarithmic temperature difference at each interval was taken.

VI. Simulation Results

The surface area of the condenser calculated for various cumulative heat loads for condensation of hydrocarbon vapours using Kern's and Bell and Ghay's methods are shown in simulation result diagram. Simulation diagram of Kern's method under design the multi-component condenser, both the methods give the same area for superheating and sub-cooling zone but differ during condensation. This is because during multi-component condensation not only is there heat transfer from both the liquid and vapour phases to the coolant but also transfer of vapour molecules from the vapour phase to the liquid phase. Therefore, there is diffusional 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 Ghay's method is more reliable and used for the design.

Effect of the vapour flow rate

The variation of the overall heat transfer coefficient with the vapour flow rate for single and two tube pass condensers. As it can be seen from simulation diagram, the overall heat transfer coefficient decreases with the increase in vapour flow rate, irrespective of the tube pass. Typically, if the flow increases, the heat transfer coefficient also increases due to higher turbulence. However, in this investigation the case is different. In the case of a vertical condensation, the heat-transfer coefficient decreases with the increase in Reynolds number up to 2000, and then increases with the Reynolds number.10 Reynolds number in this investigation was calculated in every possible combination of temperatures and condensate flow rates, and the values of Reynolds numbers were well below 2000. This justifies the results obtained by the model. Also, the higher the vapour flow rate, the higher the heat load of the condenser, thereby the higher the required surface area. 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 variation of the shell-side pressure drop** with the vapour flow rate for a single and two-tube pass condenser, a close examination of simulation diagram depicts that the shell-side pressure drop increases with the increase in vapour flow rate, irrespective of the tube pass arrangement. Besides, for a given value of vapour flow rate the two-tube pass condenser gives a lesser pressure drop than the single- tube pass. The above features are obvious because the increase in flow rate increases the pressure drop. 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.

Effect of vapour inlet temperature

The variation in the overall heat transfer coefficient for a single and two-tube pass condenser, a close examination reveals 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, as we know the behaviour of vapour density and vapour dynamic viscosity with change in temperature from the fundamentals of intermolecular forces and collision phenomenon. With an increase in vapour temperature, the density of vapour decreases and dynamic viscosity increases. With a close examination of Nusselt model equation, it is clear that an increase in inlet vapour will adversely affect the heat transfer coefficient.10 Also, the above features can be explained by the fact that an increase in vapour inlet temperature increases the de-superheat zone heat load, therefore decreases the overall heat transfer coefficient.

In simulation result diagram a plot between the tube-side pressure drop for a single and two-tube pass condenser. This figure reveals 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. 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, as we know the be-haviour of vapour density and vapour dynamic viscosity with change in temperature from the fundamentals of intermolecular forces and collision phenomenon. With an increase in vapour temperature, the density of vapour decreases and dynamic viscosity increases. With a close examination of Nusselt model equation, it is clear that an increase in inlet vapour will adversely affect the heat transfer coefficient.10 Also, the above features can be explained by the fact that an increase in vapour inlet temperature increases the de-superheat zone heat load, therefore decreases the overall heat transfer coefficient.

The tube-side pressure drop for a single and two-tube pass condenser, this figure reveals 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.

Effect of the coolant flow rate

The overall heat transfer coefficient with the coolant flow rate for a single and two-tube pass condenser, it can be seen from the figure that the increase in coolant flow rate enhances the overall heat transfer coefficient irrespective of tube pass arrangement. Also, for a given coolant flow rate, the two-tube pass condenser offers higher overall heat transfer coefficients. The above features are due to the fact that the increase in flow rate decreases the heat transfer area and thereby increases the overall heat transfer coefficient.

In simulation diagram 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.

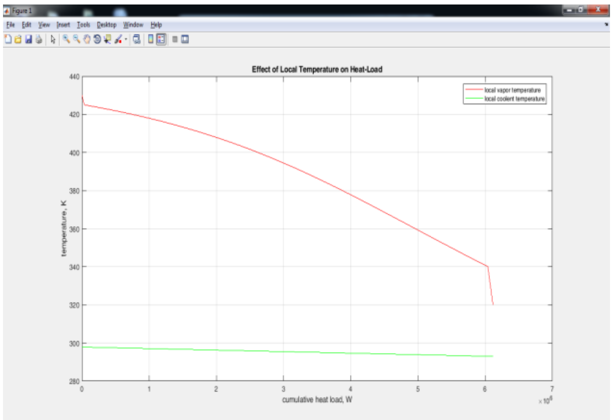
Effect of coolant inlet temperature

The variation of the overall heat transfer coefficient with the coolant inlet temperature for single and two-tube pass condensers, this figure shows 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.

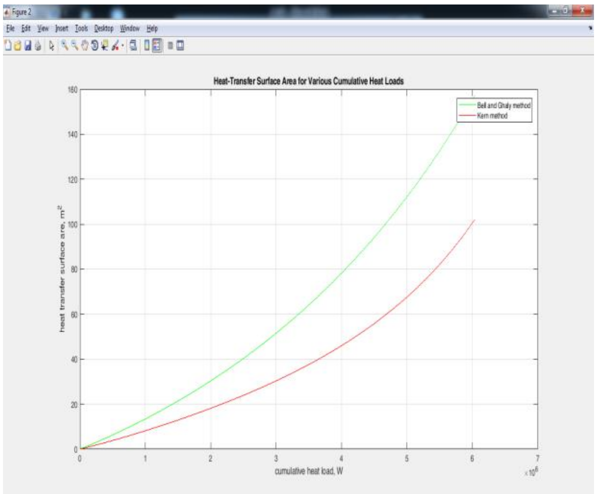
The condensation overall heat-transfer coefficient depends upon factors such as condenser type, layout, surface geometry, heat load and thermo-physical properties of the condensing vapour. The change in coolant inlet temperature will change the vapour temperature, which alters the thermo-physical properties of vapour. Hence, it is clear that an increase in inlet coolant temperature will adversely affect the heat transfer coefficient.

- **Results for Single Tube**

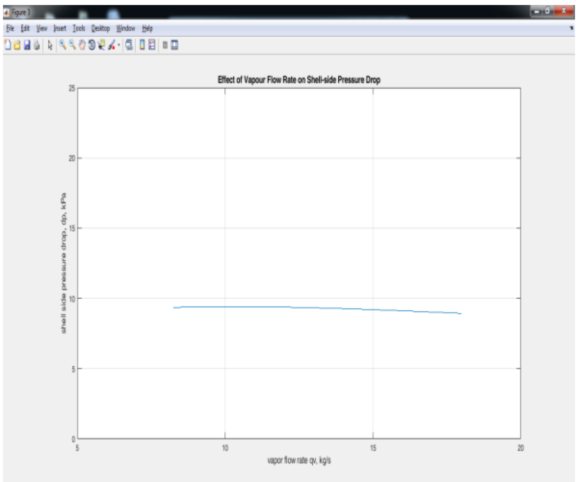
Simulation results of single tube with all parameters show in figure 3.



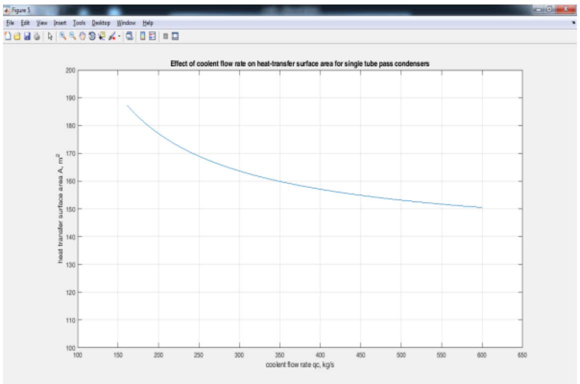
(a)



(b)



(c)

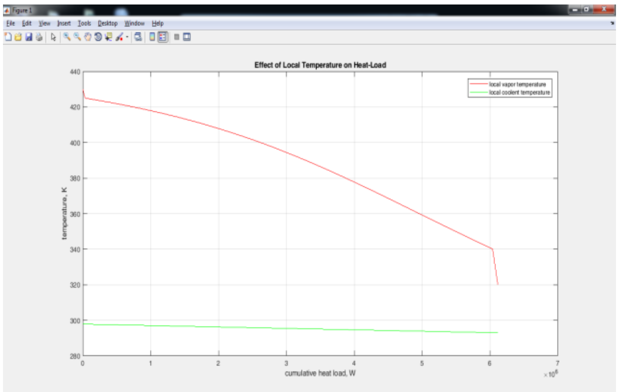


(d)

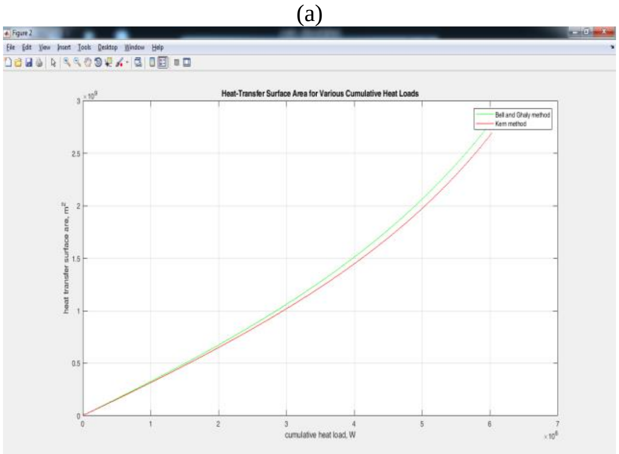
Figure 3: Simulation Results of Single Tube with all Parameters

- **Results for Double Tube**

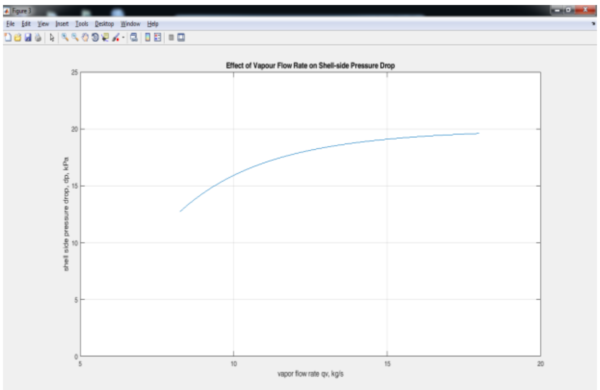
Simulation results of Double tube with all parameters show in figure 4.



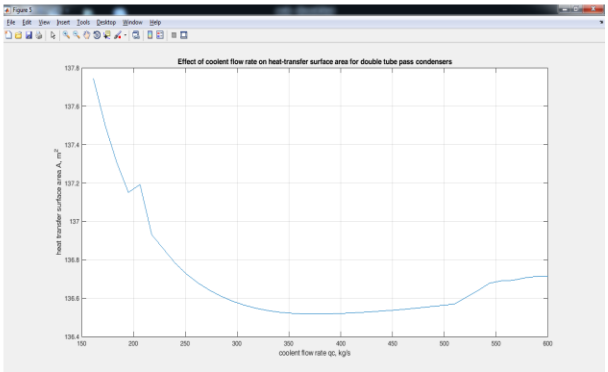
(a)



(b)



(c)

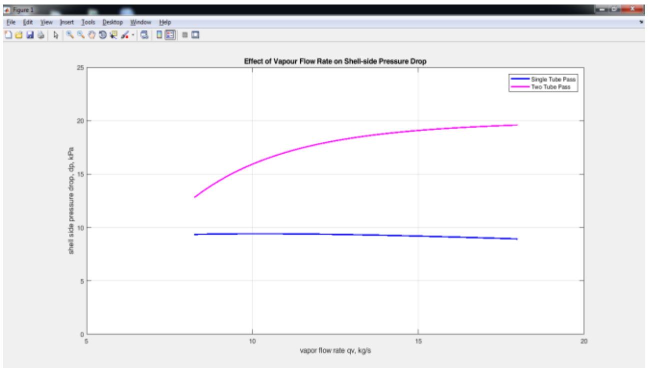


(d)

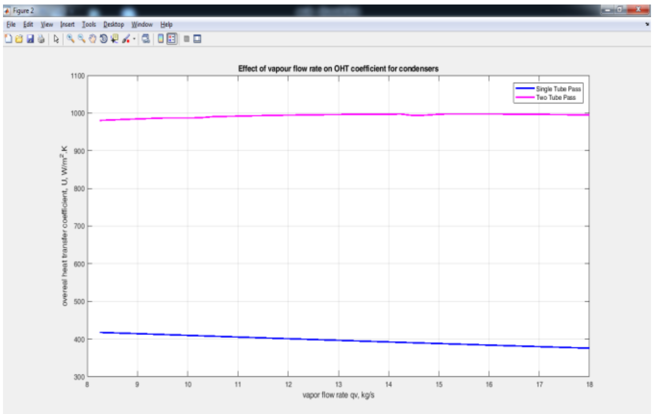
Figure 4: Simulation Results of Double Tube with all Parameters

- **Results for Single Tube and double Tube comparison**

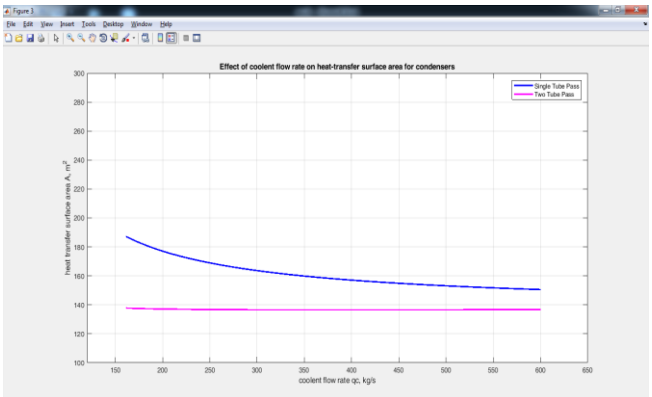
Simulation results of Single Tube and double Tube comparison with all parameters show in figure 5.



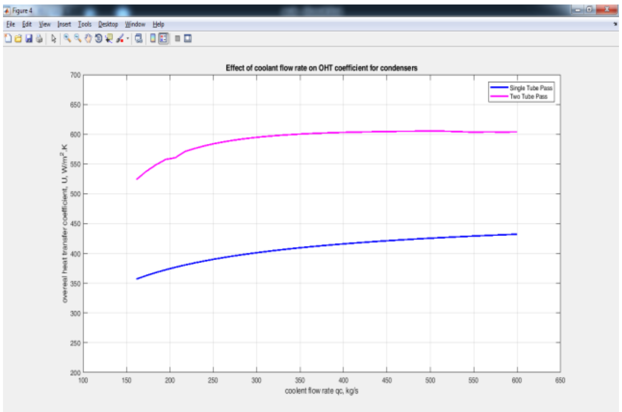
(a)



(b)



(c)



(d)

Figure 5: Simulation Results of Single Tube and Double Tube comparison with all Parameters

VII. Conclusion

The mathematical design models have been developed based on the equilibrium method, and thereby a computer aided design algorithm for the design of a multi-component condenser has been developed based on Bell and Ghaly's method. Based on the algorithm, a computer program has also been developed. This program was tested for a wide range of operating variables and found satisfactory in giving expected results. The design algorithm based on Bell and Ghaly's method was compared with Kern's method. It has been found that Kern's method undersizes the condenser because it omits the mass transfer resistance and does not take enough care to handle sensible heat transfer during condensation. This fact has been incorporated into Bell and Ghaly's method by taking the vapour phase heat-transfer coefficient during vapour sensible heat transfer in the hope that it will approximately overestimate the mass transfer resistance. The undersize of the condenser may be severe by Kern's method if the condensing range becomes large.

The effects of operating variables viz. vapour flow rate, vapour inlet temperature, coolant flow rate and its inlet temperature on the output variables viz. overall heat transfer coefficient, heat transfer surface area were studied for both single and two-tube pass condensers. As a result, 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. Further, 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 other variables have nominal effects.

References

[1] Kern, D. Q., AIChE J. 17(5) (1971) 1037.
[2] Bell, K. J., Ghaly, M. A., AIChE Symp. Ser. 131(69) (1973) 72.
[3] Colburn, A. P., Drew, T. B., Trans. Am. Inst. Chem. Engnrs. 33 (1937) 197.
[4] Colburn, A. P., Hougen, O. A., Ind. Eng. Chem. 26 (1934) 1178.
[5] Colburn, A. P., Edison, A. G., Hydrocarbon Processing 33 (1941) 457.
[6] Krishna, R., Panchal, C. B., Chem. Eng. Sci. 32 (1977) 742.
[7] Heat Exchanger Design Handbook, Hemisphere. 1983 New York
[8] Lockhart, Q. W., Martinelli, R. C., Chem. Eng. Prog. 48 (1949) 39.
[9] Kutaleladze, S. S., Borishanskii, V. M., Concise Encyclopedia of Heat Transfer, Pergamon Press, Oxford, 1966.
[10] Sinnott, R. K., Coulson & Richardson's Chemical Engineering Vol. 6,Butterworth-Heinemann, London, 1996.
[11] S. BANDYOPADHYAY et al., Computer Aided Design (CAD) of a Multi-component, Chem. Biochem. Eng. Q. 21 (2) 97–103 (2007) 103
[12] G. Ackermann, Combined heat and mass transfer in the same field at high temperature and partial pressure differences, Forsch. Ing. Wes. VDI 8 (382) (1937) 1-16.

Prakhar Mishra, Mechanical Engineering, M.Tech Student, Department of Mechanical Engineering, Faculty of Engineering & Technology, Rama University, Kanpur, India.

Mr. Ravendra Singh, Assistant Professor, Department of Mechanical Engineering, Faculty of Engineering & Technology, Rama University, Kanpur, India