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

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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 containing propane, butane, hexane, heptane and octane in the mole composition of 0.15, 0.25, 0.05, 0.30 and 0.25, respectively. 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.

Key words:
CAD, multi-component condensation, design

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]

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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.

Kern[1] 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. This was proposed by *Krishna* and *Panchal*.[6]

In this paper, 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.

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 paper, 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.

Assumptions

The design method is based on the following assumptions:

1. The liquid and vapour compositions are in equilibrium at the vapour bulk temperature.

2. Liquid and vapour enthalpies are those of the equilibrium phases at the vapour bulk temperature.

3. 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.

4. The total latent heat of condensation and sensible heat of the cooling condensate are transferred through the entire thickness of the liquid film.

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

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

At bubble point

$$\sum_{i=1}^{n} y_i = 1 \tag{2}$$

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

$$Z_{\mathrm{F},i} = y_i + (1-\theta)x_i \tag{3}$$

where $Z_{\mathrm{F,I}}$ is the vapour feed composition.

From eq. (3) the following equations are obtained for calculating the vapour and liquid mixture's composition.

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

and

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

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

$$\Phi_{\mathrm{T}} = U_{\mathrm{m}} A_{\mathrm{o}} \Delta T_{\mathrm{m}} \tag{6}$$

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.

$$\frac{\mathrm{d}\Phi_{\mathrm{T}}}{\mathrm{d}A_{\mathrm{o}}} = \frac{\mathrm{d}\Phi_{\mathrm{sv}} + \mathrm{d}\Phi_{\mathrm{p}}}{\mathrm{d}A_{\mathrm{o}}} = U_{\mathrm{m}}(T_{\mathrm{p}} - T_{\mathrm{c}}) \tag{7}$$

where,

$$U_{\mathrm{m}} = \frac{1}{\dfrac{\mathrm{d}A_{\mathrm{o}}}{h_{\mathrm{i}}\mathrm{d}A_{\mathrm{i}}} + R_{\mathrm{di}}\dfrac{\mathrm{d}A_{\mathrm{o}}}{\mathrm{d}A_{i}} + \dfrac{x_{\mathrm{w}}}{k_{\mathrm{w}}\mathrm{d}A_{\mathrm{i}}} R_{\mathrm{do}} + \dfrac{1}{h_{\mathrm{o}}}}$$

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:

$$\frac{\mathrm{d}\Phi_{\mathrm{sw}}}{\mathrm{d}A_{\mathrm{o}}} = h_{\mathrm{sw}}(T_{\mathrm{v}} - T_{\mathrm{l}}) \tag{8}$$

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

$$\frac{\mathrm{d}\Phi_{\mathrm{T}}}{U_{\mathrm{m}}(T_{\mathrm{v}} - T_{\mathrm{c}})} = \frac{\mathrm{d}A_{\mathrm{o}}}{1 + \dfrac{Z U_{\mathrm{m}}}{h_{\mathrm{m}}}} \tag{9}$$

where, $Z = \dfrac{\mathrm{d}\Phi_{\mathrm{sv}}}{\mathrm{d}\Phi_{\mathrm{T}}}$

Integration yields the following equation:

$$\int_0^{A_{\mathrm{o}}} \mathrm{d}A_{\mathrm{o}} = \int_0^{\Phi_{\mathrm{T}}} \left(\frac{1 + \dfrac{Z U_{\mathrm{m}}}{h_{\mathrm{sv}}}}{U_{\mathrm{m}}(T_{\mathrm{v}} - T)} \right) \mathrm{d}\Phi_{\mathrm{T}} \tag{10}$$

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

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

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

$$A_{\mathrm{o}} = \int_0^{\Phi_{\mathrm{T}}} \left(\frac{1 + \dfrac{Z U_{\mathrm{m}}}{h_{\mathrm{sv}}}}{U_{\mathrm{m}}(T_{\mathrm{v}} - T)} \right) \frac{\mathrm{d}\Phi}{(T - T_{\mathrm{m}})} \tag{12}$$

where, $T_{\mathrm{m}} = \sum_{k=1}^{N_{\mathrm{tp}}} \dfrac{T_{\mathrm{c}}^{\mathrm{k}}}{N_{\mathrm{tp}}}$

The term $(T - T_m)$ within 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.

Results and discussion

The upper and lower limits of the operating variables used are given in Tab. 1. The value of a set of operating and geometrical variables and output variables are given in Tab. 2. A logical structure and flow chart based on the algorithm used for the design of the condenser are shown in Fig. 1(a) and 1(b).

The surface area of the condenser calculated for various cumulative heat load for condensation of hydrocarbon vapours using Kern's and Bell and Ghay's

Table 1 – *Range of variables*

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_c/K
range	8 – 18	430 – 480	150 – 600	287 – 308

Table 2 – *Set values of operating, geometrical, and output variables for the condensation of hydrocarbon vapour mixture*

Variable	Value
Operating variables:	
coolant flow rate, q_{ma}/kg s^{-1}	12.00
vapour inlet temperature, $T_{i,v}$/K	430.00
vapour outlet temperature, $T_{o,v}$/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
Geometrical variables:	
tube outer diameter, d_o/mm	25.4
tube inside diameter, d_i/mm	19.86
length of tube, l/m	2.348
pitch arrangement	Equilateral triangular pitch
pitch	1.25d_o
number of tubes, N	2
baffle type	25% cut Segmental baffle
baffle spacing, m	D_s/5
tube sheet thickness, δ/mm	54
tube roughness, mm	0.0016
thermal conductivity of tube wall material, h/W m^{-2} K^{-1}	16.30
fouling resistance, R/m^2 K W^{-1}	0.0004
Output variables:	
dew point of the vapour mixture, T_d/K	429.72
bubble point of the vapour mixture, k	330.28
total heat load, Φ_l/kW	5885.05
weighted mean temperature difference, ΔT/K	77.0
overall heat transfer coefficient, U/W m^{-2} K^{-1}	546.10
heat transfer surface area, A/m^2	139.82
total number of tubes, N_t	752
tube side pressure drop, Δp/Pa	0.1677 × 10^5
shell side pressure drop, Δp/Pa	0.1123 × 10^4

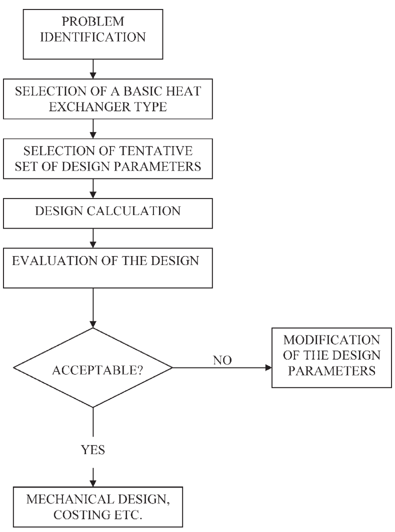

Fig. 1a – *Basic logical structure for process design of condenser*

methods are shown in Fig. 2. Fig. 2 reveals that Kern's method under design the multi-component condenser. Both the methods give the same area for superheating and subcooling zone but differ during condensation. This is because during multi-component condensation not only is there heat transfer

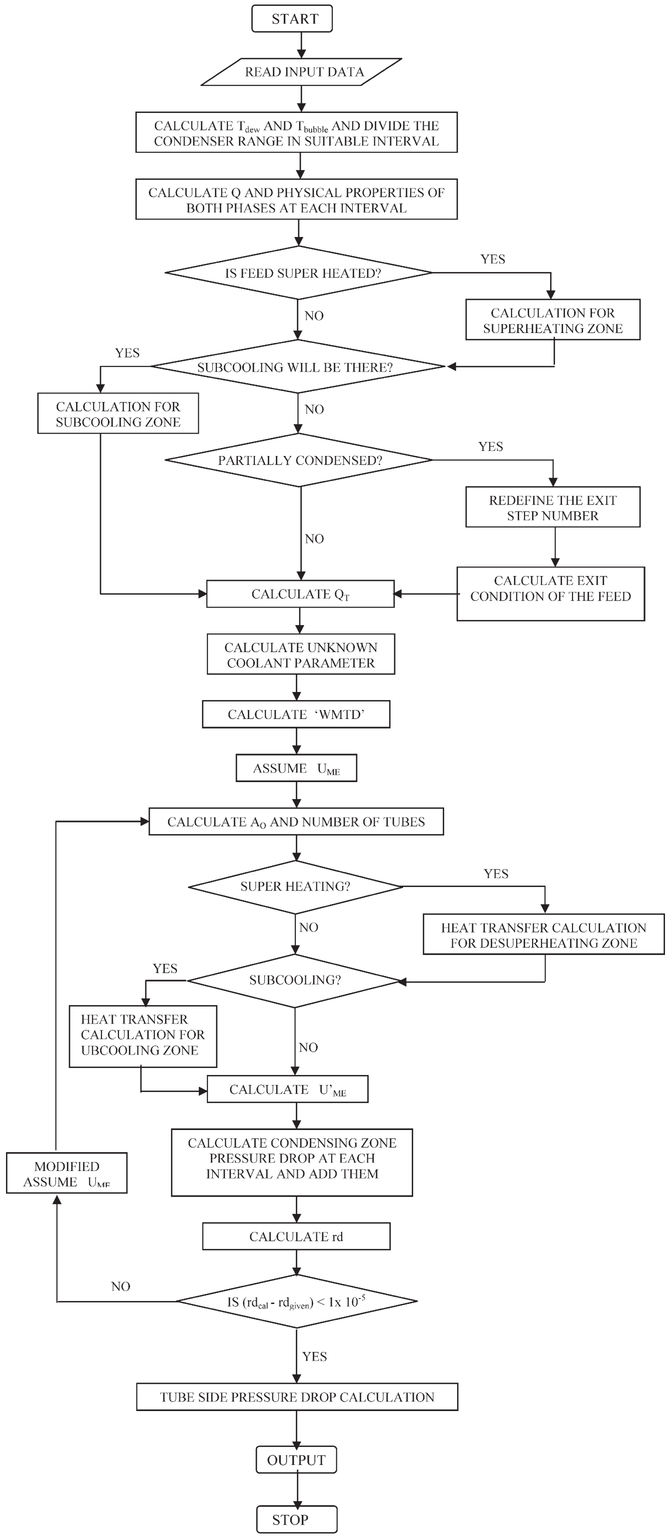

Fig. 1b – *Flow chart for the design of multi-component condenser*

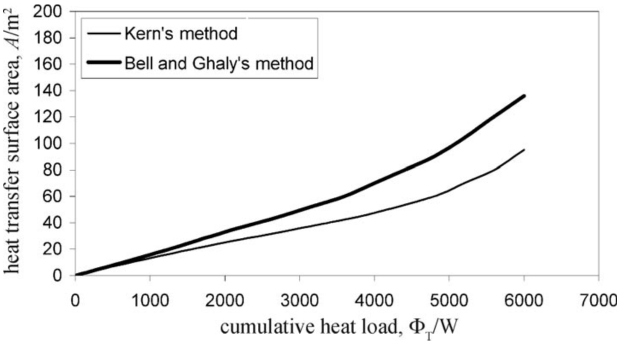

Fig. 2 – *Heat-transfer surface area for various cumulative head loads for condensation of hydrocarbon vapours*

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

Fig. 3 shows 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 Fig. 3, 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.

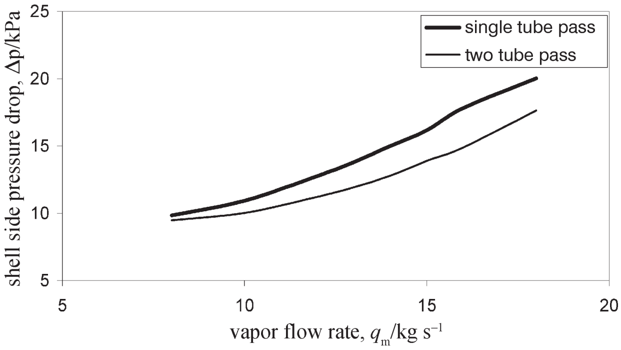

Fig. 4 – *Effect of vapour flow rate on shell-side pressure drop for single and two-tube pass condensers*

Fig. 4 represents 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 Fig. 4 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

Fig. 5 shows 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

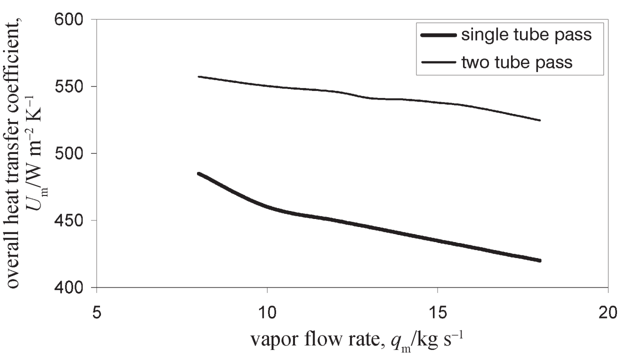

Fig. 3 – *Effect of vapour flow rate on overall heat transfer coefficient for single and two-tube pass condensers*

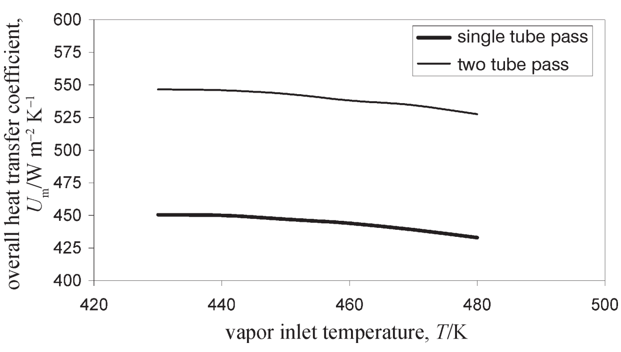

Fig. 5 – *Effect of vapour inlet temperature on overall heat transfer coefficient for single and two-tube pass condensers*

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.

Fig. 6 depicts 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.

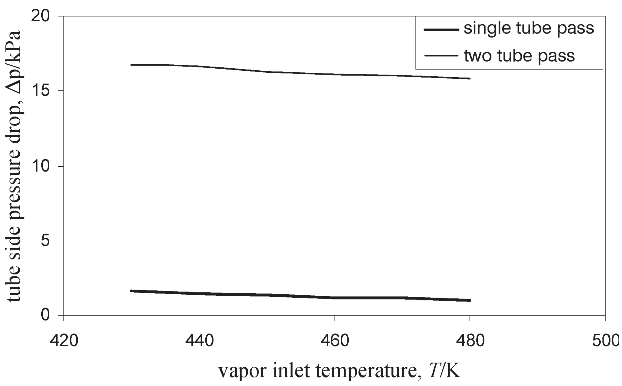

Fig. 6 – *Effect of vapour inlet temperature on tube-side pressure drop for single and two-tube pass condensers*

Effect of the coolant flow rate

Fig. 7 shows the variation of 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

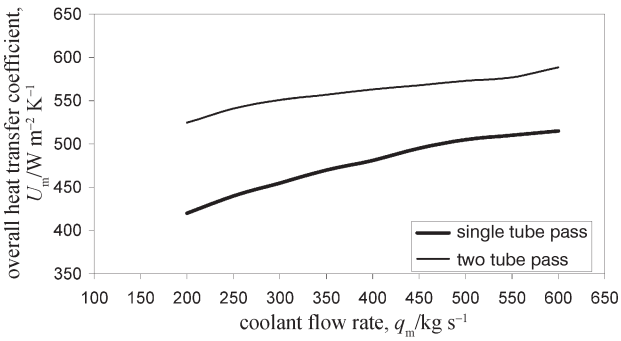

Fig. 7 – *Effect of coolant flow rate on overall heat transfer coefficient for single and two-tube pass condensers*

rate decreases the heat transfer area and thereby increases the overall heat transfer coefficient.

Fig. 8 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.

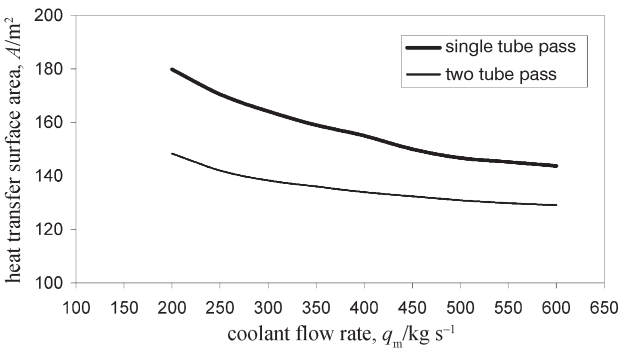

Fig. 8 – *Effect of coolant flow rate on heat-transfer surface area for single and two-tube pass condensers*

Effect of coolant inlet temperature

Fig. 9 shows 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

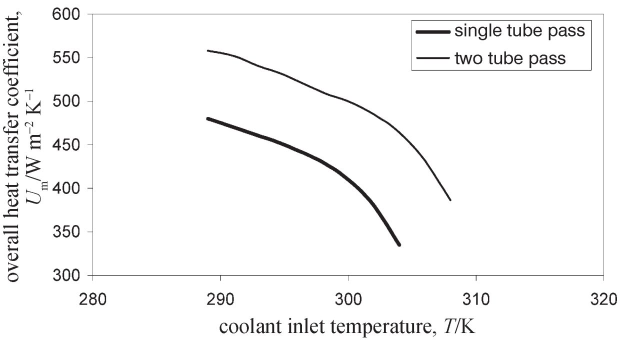

Fig. 9 – *Effect of coolant inlet temperature on overall heat-transfer coefficient for single and two-tube pass condensers*

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.

Conclusions

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.

Nomenclature

A_o – heat transfer area based on outside tube area, m^2
A_i – heat transfer area based on inside tube area, m^2
h_i – heat transfer coefficient inside the tube, W m^{-2} K^{-1}
h_o – heat transfer coefficient outside the tube, W m^{-2} K^{-1}
h_{sv} – sensible heat transfer coefficient, W m^{-2} K^{-1}
K – vapour – liquid equilibrium constant
λ_W – thermal conductivity of the wall, W m^{-1} K^{-1}
N – number of components
q_m – vapor mass flow rate, kg s^{-1}
Φ – heat load, kW
Φ_l – liquid phase heat load, kW
Φ_{sv} – sensible heat load for vapour, kW
Φ_T – total heat load, kW
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, K
T_v – vapour phase temperature, K
T_c – coolant temperature, K
U_m – mean overall design heat-transfer coefficient, W m^{-2} K^{-1}
U_{mc} – overall heat-transfer coefficient for multi-component condenser, W m^{-2} K^{-1}
x – mole fraction in liquid phase, 1
δ_W – tube wall thickness, m
y – mole fraction in vapour phase, 1
Z_F – feed molar composition
θ – vapour fraction molar basis

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