Int. J. Heat Mass Transfer. Vol. 30, No. 2, pp. 373–378, 1987
Printed in Great Britain

0017–9310/87 \$3.00+0.00

An improved approach to condenser design using film models

D. R. WEBB and D. PANAGOULIAS
Department of Chemical Engineering, University of Manchester Institute of Science and Technology,
Sackville Street, Manchester M60 1QD, U.K.

(*Received* 10 *April* 1986)

Abstract—Speed, reliability and robustness have led to the widespread use of equilibrium models in industrial design of condensers despite the considerable advantage in physical reality of the film models. A new design method is described based on the film model approach which has proved to be fast and reliable. The film method has been evaluated against data measured in a large shell-and-tube condenser. Systems studied include steam–air, mixtures of methanol, isopropanol and water and a four-component hydrocarbon mixture at atmospheric pressure. The application of the film model to the design of standard shell-and-tube condensers is demonstrated.

1. INTRODUCTION

THE FILM theory method was first applied to condensation by Colburn and co-workers [1, 2], in the mid 1930s but its use in design practice remains limited to the case of condensation of a single vapour from a non-condensing gas despite the advantage of greater physical reality over the more popular 'equilibrium approach', for example as described by Bell and Ghaly [3].

Developments in recent years have been concerned mainly with the generalisation of the method to the case of multicomponent vapour mixtures and the formulation of improved computer based design procedures. Detailed descriptions of this work and its application are to be found in Webb and McNaught [4], and Owen and Lee [5].

Improvements are described which increase the speed of the film methods to the extent that their execution is possible on a desktop computer. Furthermore, the application of the improved method is demonstrated. The method is assessed against data from a number of systems at atmospheric pressure.

2. DESIGN STRATEGY

Film theory methods, as described in the literature, involve a two-stage solution. A set of local rate equations is first solved to predict the rates of mass and energy transfer at a given position and a set of ordinary differential equations must then be integrated to carry the solution forward over an increment of area. In the present approach these steps are combined. The solution of the local equations and the change of conditions over an increment of area are achieved simultaneously. This is possible because it has become apparent that the differential equations of downstream change are well approximated by first-order difference equations [6].

A second significant improvement in efficiency is obtained by ensuring that local equations are solved in a single iterative sequence. A fairly general approach to achieving this has been given by Taylor [7], and in this respect the present method is a special case of that approach.

In the work at the University of Manchester Institute of Science and Technology on multicomponent condensation it has been demonstrated, both theoretically and experimentally—reviewed in Webb [8]—that there is no advantage in using 'interactive' rate equations for calculating mass transfer rates in condensation and the present method is based on use of the effective diffusivity method. Further it has been shown that condensate mass transfer is often unimportant and that the limiting cases of 'mixed' and 'unmixed' condensate give identical predictions. The 'unmixed' case is applied in this paper. The 'unmixed' case corresponds to the limiting behaviour shown when condensate becomes separated from the vapour flow, while the 'mixed' case corresponds to a non-separating flow with fully mixed condensate.

In the condensation of pure vapours or totally condensable vapour mixtures, a new approach is suggested. In practice some non-condensable species will inevitably be present, particularly with condensation under reduced pressure as a result of air leakage into the equipment. It is proposed that a small flow of an appropriate non-condensable is considered to be present in any condenser feed which is totally condensable under process conditions. In practice condensers are overdesigned and having space capacity will operate with an inactive region of non-condensing gas near the condenser vent. This should be well described by the proposed approach. The presence of a non-condensing gas has a beneficial effect. It avoids the difficulty associated with the point of complete condensation where there is a

NOMENCLATURE

A, B, C	Antoine constants
A	heat exchanger area
C_p	specific heat capacity at constant pressure
Δh_v	latent heat of vaporisation
K	'K' value
$\dot{M}$	mass flowrate
$\dot{n}$	molar flux
$\dot{N}$	molar flowrate
P	pressure
S, S_1, S_2	defined in equation (6)
T	temperature
$\tilde{x}, \tilde{y}$	mole fractions.
Greek symbols	
α	heat transfer coefficient
β	mass transfer coefficient
γ	activity coefficient of species in liquid
ε	Ackermann correction factor
ϕ	rate factor, equations (6) and (10).

Subscripts	
c	relating to the coolant
dew	relating to the dew point
ex	experimental
g	relating to the gas phase
i, j	indices denoting component number
I	relating to the interface
n	nth component (non-condensing gas)
o	overall (condensate surface to coolant)
T	total (sum over all species present)
th	theoretical.
Superscripts	
	modified transfer coefficient
(1)	relating to beginning of area increment
$\sim$	denotes molar quantity
$-$	denotes mean value in interval.

discontinuity in many of the variables or their derivatives. Furthermore in the general case of condensation of multicomponent vapours this non-condensing gas is the appropriate reference component for binary diffusivities.

The method proposed involves sub-dividing the condenser into increments of area chosen as one half the area of each baffle space. The rating calculations lead to the evaluation of vapour flowrate at the mid point and end of each baffle space and the calculation of pressure drop which is generally based on these flowrates is readily built into the method.

3. THEORY

The film theory approach to condensation has been described elsewhere, for example ref. [4], and only those equations necessary to understand the new model are included in this section.

In the multicomponent condensation of a mixture of n components, local conditions are completely defined if gas temperature, flowrate and coolant temperature are known. These quantities have to be determined at a position, $A + \Delta A$, from their known values at A, where A is exchanger area measured from the vapour inlet. They are determined by ordinary differential equations which in the present approach are approximated as difference equations evaluated at the mean condition within A.

For gas temperature T_g

$$\Delta T_g = \frac{\alpha_g e^{\varepsilon}(T_I - \bar{T}_g)\Delta A}{\dot{N}_g \tilde{C}_{pg}}. \tag{1}$$

For coolant temperature T_c

$$\Delta T_c = \pm \frac{\alpha_0(\bar{T}_c - T_I)\Delta A}{\dot{M}_c C_{pc}} \tag{2}$$

$+$ countercurrent flow

$-$ cocurrent flow.

For vapour flowrate $\dot{N}_{ig}$

$$\Delta \dot{N}_{ig} = \beta_{in} \ln\left[\frac{1 - \bar{\tilde{y}}_{ig}}{1 - \tilde{y}_{iI}}\right]\Delta A. \tag{3}$$

for $i = 1$ to $n - 1$

The key equation in advancing the solution over an increment of area is the condition of continuity of energy flux across the interface and this should be satisfied for the mean conditions, $\bar{T}_g$, $\bar{T}_c$ and $\bar{\tilde{y}}_{ig}$. The energy flux to the coolant is made up of two contributions. The first arises from the transfer of sensible heat while the second is due to latent heat. This can be expressed:

$$\alpha_0(\bar{T}_c - T_I) = \alpha_g \varepsilon (T_I - \bar{T}_g)/(e^{\varepsilon} - 1) + \sum^{n-1} \beta_{in} \Delta \tilde{h}_{vi} \ln\left[\frac{1 - \bar{\tilde{y}}_{ig}}{1 - \tilde{y}_{iI}}\right]. \tag{4}$$

The approach diverges at this point depending on whether the mixture contains a single condensing species or more than one condensing species. It is considered that the former is worth treating separately because of its special importance in condensation of steam–air mixtures. The general case is formulated differently but may be applied to a single condensing species, and has been shown to give identical results.

3.1. *Condensation of a single component from a non-condensing gas*

Equations (1)–(3) are used to replace the average conditions $\bar{T}_g$, $\bar{T}_c$ and $\bar{\tilde{y}}_{ig}$ in equation (4) by their values at the beginning of the interval. In the case of the composition, this involves the solution of a quadratic (and incidentally it is this term which prevents the same approach for multicomponent mixtures). This gives

$$f(T_I) = \alpha_{\dot{g}}(T_I - T_g^{(1)})/S_1 - \alpha_0(T_c^{(1)} - T_I)/S_2 + \beta_{12}\Delta\tilde{h}_{vl}\ln\left\{\frac{(1-S)}{2} + \frac{1}{2}[(1+S)^2 + 4S(\tilde{y}_{1I} - \tilde{y}_{1g}^{(1)})]^{1/2}\right\} \quad (5)$$

where

$$\begin{aligned} S_1 &= (1 + \alpha_{\dot{g}}\,e^{\varepsilon}\Delta A/2\dot{N}_g\tilde{C}_{pg}) \simeq 1.0 \\ S_2 &= (1 \mp \alpha_0\Delta A/2\dot{M}_cC_{pc}) \simeq 1.0 \\ S &= 2\dot{N}_g^{(1)}/\beta_{\dot{1}2}\Delta A \\ \alpha_{\dot{g}} &= \alpha_g\varepsilon/(e^{\varepsilon} - 1); \varepsilon = \dot{n}_1\tilde{C}_{p1}/\alpha_g \\ \beta_{\dot{1}2} &= \beta_{12}\phi/(e^{\phi} - 1); \phi = \dot{n}_1/\beta_{12}. \end{aligned} \quad (6)$$

The only quantity in equation (5) which is a sensitive function of temperature is $\tilde{y}_{1I}$ and this dependence has been expressed by the Antoine equation

$$\tilde{y}_{1I} = \exp\left[A - \frac{B}{C + T_I}\right]_1 \bigg/ P. \quad (7)$$

Equation (5) has been solved by the Newton–Raphson method for T_I in a single iteration loop which updates all other dependent quantities in equations (5) and (6). Experience with many hundreds of calculations with air–water has shown convergence to within 0.1°C in 2–4 iterations when $T_I \simeq T_{g,dew}$ (condensate-side control) and when $T_I \simeq T_c$ (gas-side control) and 4–6 iterations when both films are controlling and a good initial estimate of T_I is not available.

3.2. *Condensation of multicomponent vapours from a non-condensing gas*

The same basic approach is used as in Section 3.1 except that no method has yet been found to achieve convergence of T_I based on the equation of continuity of energy, equation (5). In its place has been written the condition that liquid composition should sum to unity, that is

$$f(T_I) = 1 - \sum^{n-1} \tilde{x}_{iI} = 0. \quad (8)$$

The liquid composition may be related to the known gas composition at the beginning of the interval by appropriate manipulation of the equation for the transfer flux of the ith constituent of the mixture

$$f(T_I) = 1 - \sum^{n-1} \tilde{y}_{ig}{}^{(1)}/\{1 + e^{\phi_i}(K_i - 1)[1 + \dot{n}_T\Delta A/2\dot{N}_g^{(1)}]\} \quad (9)$$

$$\phi_i = \dot{n}_T/\beta_{in}$$

$$\dot{n}_T = \left[\frac{\alpha_0(T_c^{(1)} - T_I)}{S_2} - \frac{\alpha_{\dot{g}}(T_I - T_g^{(1)})}{S_1}\right]\bigg/\Delta\tilde{h}_v \quad (10)$$

$$K_i = \exp\{A - B/(C + T_I)\}_i\gamma_i/P.$$

All terms in equation (9) have therefore been expressed directly as functions of T_I and it may therefore be solved by an analytical Newton–Raphson method in one independent variable.

The range of convergence of equation (9) is limited. Interfacial temperature must lie between T_c and $T_{g,dew}$. The guess $T_I = T_c$ converges slowly because $f'(T_I)$ is very large near T_c. However an initial guess $T_I = T_{g,dew}$ and using the previous value of T_I in later intervals has given convergence of T_I within 0.1°C in 3–5 iterations in most cases.

4. FEATURES OF COMPUTER PROGRAM

4.1. *Physical properties*

Physical properties are evaluated within the iteration loop from appropriate fitting equations at the temperatures (of condensate, interface, gas film, bulk gas or coolant) demanded by their application. In the case of the multicomponent hydrocarbon mixtures only latent heat, vapour pressure and diffusivities are estimated for each constituent. Other properties are evaluated for one representative constituent. This was acceptable in the present study because in the hydrocarbon mixture the constituents were of very similar physical properties.

4.2. *Heat and mass transfer coefficients*

Coolant, condensate and gas film coefficients are evaluated by standard methods within the loop of the iterative procedure with properties at the appropriate temperature:

Coolant	Dittus–Boelter
Condensate	Nusselt modified
Gas film	Taborek [11].

With the condensate, the approach recommended by Owen [5] has been adopted to include the effects of inundation, as given by Short and Brown [9], and vapour shear, as developed from the work of Shekriladze and Gomelauri [10]. The gas film has been treated by the J factor method described by Taborek [11], and the Chilton–Colburn analogy ($J_H = J_D$) applied to obtain mass transfer coefficients.

4.3. *Pressure drop calculations*

Pressure drop is estimated by the methods given by Taborek [11], based on the average loadings in an increment of area. A two-phase enhancement is calculated by the method described by Grant [12]. Pressure drop is calculated after the thermal rating has been completed for that interval. The physical characteristics of each baffle space are separately specified and nozzle pressure drop is calculated for all common nozzle positions.

4.4. *Multiple tube passes*

The program has been written for the particular case of an E-shell and designed to accommodate all tube pass arrangements up to four in which pass partition lanes are horizontal or vertical diameters. The exchanger may contain an odd or even number of baffles. It should be noticed that with this limitation heat transfer takes place to two passes, at most, in any half baffle space, even with exchangers having four coolant passes.

4.5. *Other features*

(a) Condensate is assumed to be sub-cooled to the film average temperature appropriate for laminar condensate flow. The sensible heat of sub-cooling is added to the heat load to be conducted across the condensate.

(b) The modifications described by Taborek [11] for laminar flow are included both with respect to heat and mass transfer and become effective at shell-side Reynolds numbers below 100.

(c) In multipass units it is possible for the vapour to be cooled below the temperature of some subsequent pass. In this event the condensation rate is set to zero and application of the standard film model equations gives rise to vapour reheating.

4.6. *Performance of the program*

As a general guide the following performances have been achieved on a widely available microcomputer. Figures are for condensation of one, two or four condensable vapours from a non-condensing gas in a rating calculation involving 16 increments of area and with non-compiled execution.

		Time
1 component	1 pass	4 min
1 component	2 passes	8 min
1 component	4 passes	8 min
2 components	1 pass	4.5 min
4 components	1 pass	6 min.

With compiled execution the program has been shown to be about 20 times faster. Memory requirement is about 30–40K with liberal allowance for input–output. The program has been extensively tested over the following ranges of variables.

Gas/vapour ratio	0.001–1000
Pressure	1–0.05 bar
Total flowrates	10 fold
Degree of superheat	100°C

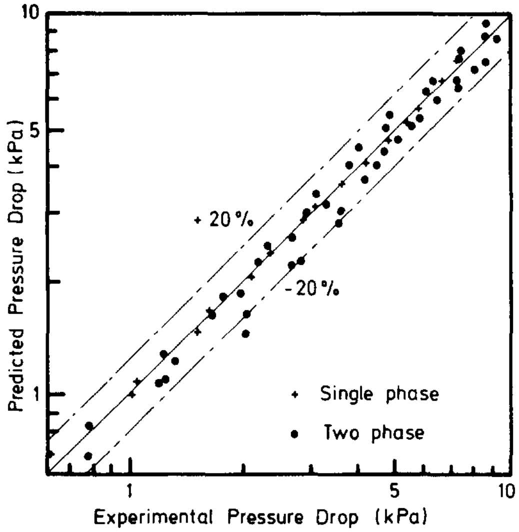

FIG. 1. Pressure drop predictions. Pressure drop in simulated conditions of condensation. Water was sprayed over a tube bundle at various air loadings.

5. EXPERIMENTAL RESULTS

Extensive experimental work has been carried out in a horizontally mounted E-shell condenser of standard design at atmospheric pressure. The shell is of 0.438 m diameter with a tube length of 2.438 m between tube plates. The unit has seven vertically cut segmental baffles with a cut of 35.5%. Vapour enters through a nozzle of 0.203 m diameter while the vent is of 0.152 m diameter. The unit has 196 tubes (14 SWG and 19.05 mm o.d.) arranged in two passes on a triangular pitch (angle of 60°). Full design details are given by Panagoulias [6]. The unit can be fed with a range of vapours for condensation studies and can be operated with a water spray over the tubes to simulate condensation. The experimental results appear in full in Panagoulias [6] and copies of principal data are available on request. Only selected results are presented in the limited space available in this paper.

5.1. *Pressure drop in single-phase flow and simulated condensation*

Pressure drop over the condenser as a whole (inlet to outlet) has been measured for single-phase air flowrates from 0.2 to 0.6 $kg\,s^{-1}$ giving pressure drops between 0.5 and 7 kPa. The Bell method as described by Taborek [11] proved to give excellent prediction of single-phase pressure drop as shown in Fig. 1.

Pressure drop over a single baffle space has been measured at air flowrates from 0.15 to 0.6 $kg\,s^{-1}$ with water sprayed over the bundle at rates from 0.15 to 1.0 $kg\,s^{-1}$ giving pressure drops from 0.1 to 9.0 kPa. The water was sprayed from 16 individual locations over the top of the shell, spaced to match the tube area immediately below. These data were compared with the predictions of the Bell method modified by

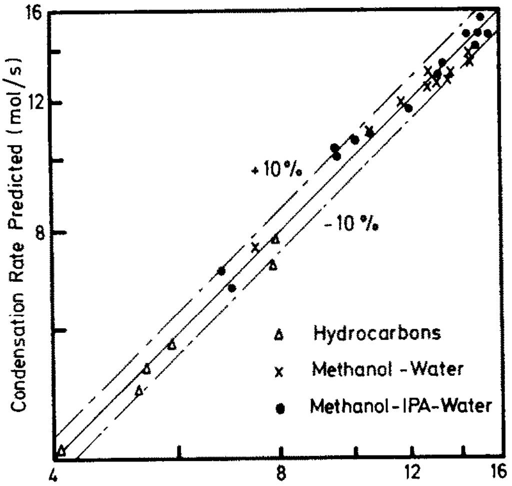

FIG. 2. Comparison of condensation rates. Predictions of condensation rate at atmospheric pressure by the film model for a range of vapours forming miscible condensates.

the method of Grant [12], to account separately for the two phase enhancement of crossflow and window flow zones. The comparison is shown in Fig. 1.

5.2. *Condensation studies*

The following table summarises the experimental work on which this evaluation is based.

System	No. of runs
1 Methanol–water	13
2 Methanol–isopropanol–water	13
3 Commercial hexane (four constituents)	6
4 Steam–air	24

The mass flowrate of vapours to be condensed ranged from 0.05 to 0.8 kg s^{-1} with heat loads from about 0.2 to 1 MW. In all cases studied the condenser was operated at capacity by increasing the coolant temperature to the point where a substantial part of the vapours remained uncondensed. Heat loads based on cooling water temperature rise matched the enthalpy change of the species within the condenser within ±10%.

The comparison between theory and experiment is shown in Fig. 2 for systems 1–3 of the above table. The method described in the theory section has been applied to systems 3 and 4. All the data have been analysed by the film method as described in Webb and McNaught [4]. The new method was accurate within the precision expected on the basis of approximations in the estimation of physical properties. The film theory approach has matched the observed thermal performance of the condenser within ±10%. In the case of system 1 it has been demonstrated that the performance of the equilibrium approach is inferior giving predictions within a wider band of ±25%. Separate comparisons were made with the predictions of the differential and integral forms of the equilibrium approach. Furthermore, it has been demonstrated, with system 2 that there is no advantage in using the more sophisticated methods which incorporate an improved description of the diffusional behaviour of the vapour phase.

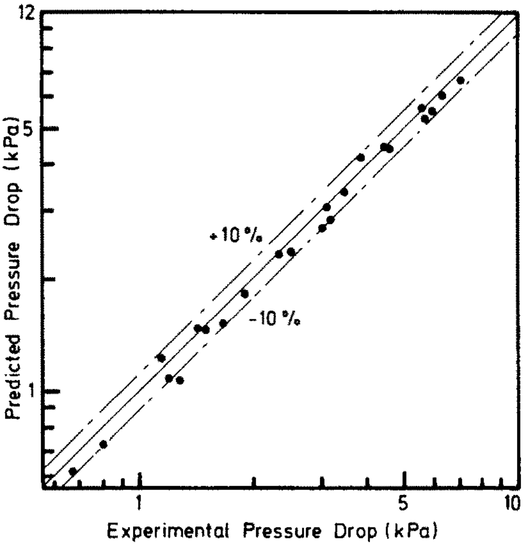

FIG. 3. Pressure drops in steam condensation. Condensation of steam with appreciable air concentrations at the atmospheric pressure with flowrate distribution predicted by the film model.

In the case of the condensation of steam from air the comparison is shown in Fig. 3. Here the condensation process was taken almost to completion and comparison of condensation rate is not meaningful. However, the comparison between predicted and measured pressure drops provides a fairly sensitive test of predictions of rates of condensation. This is due to the fact that the pressure drop is proportional to the square of the velocity. At high air flowrates of course this effect is less pronounced.

6. CONCLUSIONS

(1) A new procedure is suggested for condenser design by the 'film theory' method.

(2) The film theory approach has been applied successfully to condenser rating for a range of systems at atmospheric pressure including steam–air mixtures, methanol–isopropanol–steam mixtures and a mixture of light hydrocarbons.

(3) The new procedure has been demonstrated to work well for the steam–air and mixed hydrocarbon systems.

REFERENCES

1. A. P. Colburn and T. B. Drew, The condensation of mixed vapours, *Trans. Am. Instn chem. Engrs* **33**, 197 (1937).
2. A. P. Colburn and O. A. Hougen, Design of cooler condensers for mixtures of vapours with non-condensing gases, *Ind. Engng Chem.* **26**, 1178 (1934).
3. K. J. Bell and M. A. Ghaly, An approximate generalised design method for multicomponent partial condensers, *A.I.Ch.E. Symp. Ser.* **69**(131), 72–79 (1972).
4. D. R. Webb and J. M. McNaught, Condensers in developments. In *Heat Exchanger Technology* (Edited by D. Chisholm), 1st edn, pp. 71–126. Applied Science, Barking, U.K. (1980).
5. R. G. Owen and W. C. Lee, A review of some recent developments in condensation theory. In *Condensers—Theory and Practice, I. Chem. Engrs Symp. Ser.* No. 75, pp. 261–308 (1983).
6. D. Panagoulias, Condensation of multicomponent vapors in a condenser of semi-industrial scale. Ph.D. thesis, University of Manchester (1984).
7. R. Taylor, A. Lucia and R. Krishnamurthy, A Newton-like algorithm for the efficient estimation of rates of multi-component condensation by a film model, *I. Chem. Engrs Symp.* No. 75, 261–308 (1983).
8. D. R. Webb, Heat and mass transfer in condensation of multicomponent vapours, *Proc. 7th Int. Heat Transfer Conference*, Munich, Paper Cs 27 (1982).
9. B. E. Short and H. E. Brown, Condensation of vapours on vertical banks of horizontal tubes, *Proc. Instn mech. Engrs, Gen. Discuss. Heat Transfer* 27–32 (1951).
10. I. G. Shekriladze and V. I. Gomelauri, Theoretical study of laminar film condensation of flowing vapour, *Int. J. Heat Mass Transfer* **15**, 235–246 (1966).
11. J. Taborek, *Heat Exchanger Design Handbook*, Chap. 3. Hemisphere, New York (1983).
12. I. D. R. Grant, Flow and pressure drop with single- and two-phase flow on the shellside of segmentally baffled shell and tube heat exchangers. In *Advances in Thermal and Mechanical Design of Shell and Tube Heat Exchangers*, pp. 1–22, NEL Report 590. East Kilbride, Glasgow (1975).

UNE APPROCHE AMELIOREE DE LA CONCEPTION DES CONDENSEURS, UTILISANT LES MODELES DE FILM

Résumé—La vitesse, la souplesse et la robustesse ont conduit à l'utilisation étendue des modèles d'équilibre dans la conception des condenseurs dans l'industrie, malgré l'avantage considérable de la réalité physique des modèles de film. Une nouvelle méthode est décrite, basée sur le modèle de film, rapide et souple. Elle a été évaluée en comparaison avec les données expérimentales dans un grand condenseur à tube et enveloppe. Les systèmes étudiés incluent air–vapeur d'eau, mélanges de méthanol, isopropanol et eau et un mélange à quatre composants d'hydrocarbures. L'application du modèle de film est faite pour des condenseurs standardisés calandre-tubes.

EIN VERBESSERTES VERFAHREN DER KONDENSATOR-AUSLEGUNG UNTER VERWENDUNG VON FILM-MODELLEN

Zusammenfassung—Schnelligkeit, Zuverlässigkeit und Robustheit haben den Gleichgewichts-Modellen zu einer weitverbreiteten Anwendung bei der industriellen Auslegung von Kondensatoren verholfen—trotz der beträchtlichen Vorteile der Film-Modelle, was die physikalische Wirklichkeit anlangt. Ein neues Auslegungsverfahren wird beschrieben, welches auf einem Film-Modell beruht und sich als schnell und zuverlässig erwiesen hat. Das Film-Verfahren ist in Anlehnung an Versuchsdaten aus einem großen Rohrbündel-Kondensator entwickelt worden. Verschiedene Stoffsysteme wurden untersucht: Wasserdampf–Luft, Gemische aus Methanol, Isopropanol und Wasser und ein Vier-Komponenten-Gemisch aus Kohlenwasserstoffen bei Atmosphärendruck. Die Anwendung des Film-Modells bei der Auslegung eines üblichen Rohrbündel-Kondensators wird gezeigt.

УСОВЕРШЕНСТВОВАННЫЙ МЕТОД РАСЧЕТА КОНДЕНСАТОРА С ПОМОЩЬЮ ПЛЕНОЧНЫХ МОДЕЛЕЙ

Аннотация—Для промышленных расчетов конденсаторов широко используются равновесные модели благодаря их надежности и небольшим затратам времени, несмотря на то, что значительно более целесообразными с физической точки зрения являются пленочные модели. В работе предлагается новый, основанный на использовании пленочной модели метод расчета, с помощью которого можно быстро получить надежные результаты. Применимость этого метода оценивается исходя из результатов измерений для трубного конденсатора больших размеров. При атмосферном давлении исследовались системы: водяной пар–воздух, смеси метанола и изопропанола с водой и четырех-компонентная смесь углеводородов. Дан пример использования пленочной модели для расчета обычных трубных конденсаторов.