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REFLUX CONDENSATION OF PURE VAPORS WITH AND WITHOUT A NONCONDENSABLE GAS INSIDE PLAIN AND ENHANCED TUBES*

A. N. Abdelmessih
Seattle University, Seattle, WA 98122-4460
T. J. Rabas and C. B. Panchal
Argonne National Laboratory, Argonne, IL 60559-4815

ABSTRACT

Estimates of the surface-area and vapor-release reductions are obtained when commercially available enhanced tubes (spirally ribbed) replace plain tubes in a reflux unit condensing pure organic vapors with different concentrations of a noncondensable gas. This investigation was undertaken because there are no existing data and/or prediction methods that are applicable for these shell-and-tube condensers commonly used in the process industries. To obtain these estimates, existing design methods published in the open literature were used. The major findings are that 1) surface-area reductions can almost approach the single-phase heat transfer enhancement level, and 2) vapor-release reductions can approach a factor of four. The important implication is that enhanced tubes appear to be very cost effective for addressing the recovery of volatile organic vapors (VOCs), and for a vast number of different reflux-condenser applications.

A reflux or knockback condenser is a vertical intube type with countercurrent flow of the condensate with respect to the vapor. The vapor enters the bottom of the tubes and flows upward while the condensate, which is condensed on the tube walls, flows by gravity downward and drains from the bottom of the tubes. A sketch of a reflux condenser is shown in Figure 1.

In process industries, reflux condensers are mounted on top of chemical reactors, boilers, or distillation columns and in dephlegmators as hydrocarbon separation equipment. They minimize piping and pressure losses, significantly reduce condensate subcooling, and are very desirable in multicomponent separation processes. Because reflux condensers are commonly required to deal with vapors containing noncondensable gases and vapor mixtures, they therefore are particularly valuable in abiding with the environmental laws, such as the case of separating volatile organic compounds from air and/or nitrogen.

The technology base of reflux condensers has been relatively stagnant. One technology worthy of consideration for reflux condensers is internally enhanced tubes. They are commonly used for intube condensation of refrigerants in residential air-conditioning units and for the shellside condensation of steam in multi-effect desalination units. A comprehensive bibliography of publications related to these successful applications is presented by Bergles et al. [1].

We are not aware of any publication(s) that even address the feasibility of using enhanced tubes in reflux condensers. There is good reason to believe that the enhanced-condenser technology could be a cost-effective solution based on the above-mentioned successful applications. However, none of the successful applications are confronted with the added complication of large concentrations of a noncondensable gas and/or of vapor mixtures and the countercurrent flow issue with the reflux design.

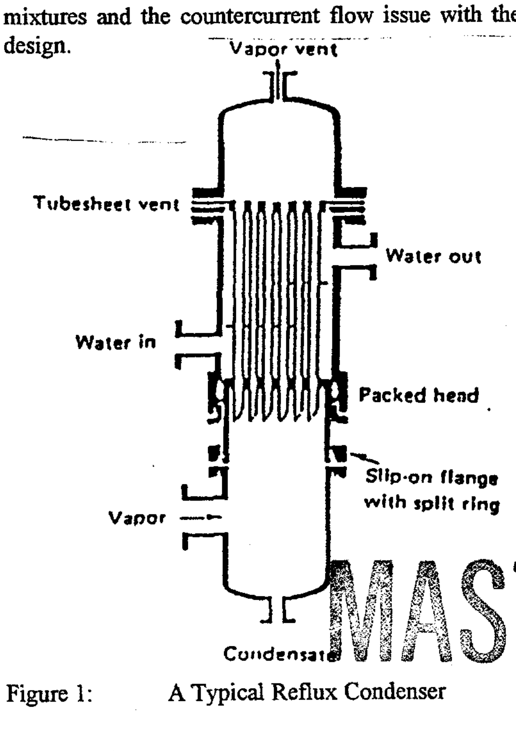

Figure 1: A Typical Reflux Condenser

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Some encouragement for the success of enhanced tubes in reflux condensers is found in the literature. Noteworthy is the statement by Webb [2] that any enhancement technique effective for convective heat transfer to gases should be beneficial for convective condensation with a noncondensable gas. Rabas and Panchal [3] very recently showed that surface-area and vapor-release reduction ratios of two and five, respectively, are possible based on currently accepted design methods for vertical intube downflow condensers. The purpose of this work was to investigate the feasibility of internally enhanced tubes for reflux condensers. The analysis description developed for the feasibility investigation is first presented and then the results from some cases are discussed.

ANALYSIS DESCRIPTION

Some important considerations and selections needed for this feasibility investigation are the following:

1) enhancement type and performance level,
2) vapor and noncondensable gas,
3) noncondensable-gas prediction method,
4) flooding criterion,
5) pressure drop prediction method, and
6) heat transfer prediction method.

Enhancement Type and Performance Level

An internally finned or ribbed tube geometry was selected as shown in Figure 2. Enhanced tubes of this type are used in all residential air-conditioning condensers (Webb, [4]). This preferred enhancement geometry, which consists of low, multistart ribs with a low spiral angle (almost axial), was selected because of the good performance regardless of the flow regime with the vertical orientation. Further discussion supporting this selection is presented by Rabas and Panchal [3]. Also they showed that single-phase heat transfer enhancement levels between 1.5 to 2.5 are possible with enhanced tubes of this type. The assumption will be made in the analysis that the mass transfer enhancement levels are the same with this enhanced tube type.

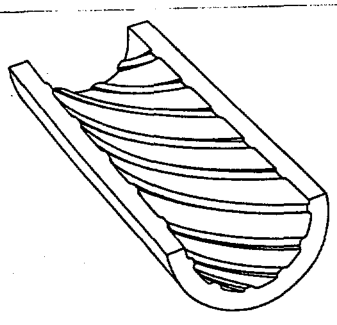

Figure 2: Typical Spirally Ribbed Enhanced Tube

Vapors and Noncondensable Gas

Para-xylene was the selected organic vapor. It is number two on the list of common VOC emissions compiled by Ruddy and Carroll [5] that need to be restricted. Steam was the second vapor selected, mostly for comparison purposes because of its low molecular-weight. Nitrogen, was the noncondensable gas selected for the investigation.

Noncondensable Gas Prediction Method

The algorithm based on the pointwise trial and error analysis of Colburn-Hougen [6] was used for the analysis. The details of the analysis are contained in publications by Rabas and Panchal [3] and Rabas and Mueller [7]. As a result, no further discussion is presented here.

Flooding Criterion

Flooding occurs when the inlet total flow rate exceeds a threshold value that depends on the fluid properties, tube inside diameter, and inlet configuration. With a flooding condition, some or all of the condensate is entrained with the exiting vapor/noncondensable gas mixture and a substantial increase of the pressure drop occurs. Rabas and Arman [8] identified two different flooding mechanisms: (1) condensate drops entrained or "sucked" back into the tube, and (2) shearing of liquid from the condensate film on the tube wall. The first to occur is the drop-entrainment mechanism, if design precautions are not implemented. The most common prevention methods are tapered tube ends as in Fig. 1 (English et al. [9]), a secondary tubesheet (Rabas and Arman [8]) or an increased tube diameters (Diel and Koppany [10]). However, the second or the shear-type mechanism can not ignored.

Most of the flooding investigations did focus on the shearing mechanism because various types of inlet configurations were usually employed to bypass the drop-entrainment mechanism. McQuillan and Whalley [11] compared 22 flooding correlations with an experimental flooding data bank for gas-liquid flow in vertical, circular plain tubes. They concluded that the best empirical correlation is

$$K_g = 0.286\,Bo^{0.26}\,Fr^{-0.22}\,\{1 + \mu_l/\mu_w\}^{-0.18} \qquad (1)$$

Comparisons of the flooding mass velocities predicted with this correlation with results from the correlations of Chen et al. [12], Diehl and Koppany [10], and English et al. [9], and are presented in Table 1. No end taper was used in the correlation of English et al. in order to obtain the lowest prediction of flooding mass velocities. With para-xylene, Table 1 shows that the most conservative values were usually obtained with Equation (1) and followed by results from the correlation of Diehl and Koppany [10]. Because the predictions with Equation (1) are the most conservative for para-xylene and most likely other organic fluids, we decided to use it to predict flooding conditions.

The flooding mass velocities most likely are larger for enhanced tubes. Surface-tension forces pull the condensate into the valleys between the ribs and thin the condensate film at the fin tips. Disrupting the film and entraining the condensate should therefore occur at larger inlet mass velocities. Because we are not aware of any flooding data obtained with enhanced tubes, Equation 1 was used and should yield a conservative estimate.

TABLE 1: Comparison of Flooding Gt Values (kg/m^2s)

Fluid Type	Pres kPa	I.D. mm	Eq. (1)	Chen	Diehl	English
para-	110	25.4	8.16	11.47	8.94	8.97
xylene		38.1	9.02	12.72	8.94	10.13
		50.8	9.68	13.58	8.94	11.04
para-	55	25.4	6.52	8.60	6.97	6.56
xylene		38.1	7.21	9.55	6.97	7.41
		50.8	7.73	10.21	6.97	8.07
steam	110	25.4	5.48	6.28	7.41	4.75
		38.1	6.06	7.08	7.41	5.37
		50.8	6.50	7.65	7.41	5.85
steam	55	25.4	4.28	4.68	5.54	3.39
		38.1	4.72	5.19	5.54	3.83
		50.8	5.07	5.61	5.54	4.17

Pressure Drop Prediction Method

A recent procedure for calculating the countercurrent pressure drop in vertical tubes was developed by Chunangad [13]. Unfortunately when the suggested procedure was used, the predicted values of the frictional, gravitational, and momentum pressure drop components were similar to that predicted for cocurrent flows. As a result, a new method was needed to compute the plain-tube, single-component pressure drop.

The first quantity needed for the calculation is the vapor void fraction. We used the film-thickness predictions based on the Nusselt and Bekin correlations, as recommended by Bankoff and Lee [14] and reported by McQuillan and Whalley [11]. For laminar flow, the film thickness based on the Nusselt condensing theory is used;

$$\delta^* = 0.908\, Re_l^{\,0.333} \qquad (2)$$

and for turbulent flow, the film-thickness correlation of Bekin is used

$$\delta^* = 0.135\, Re_l^{\,0.583} \qquad (3)$$

In Equations (2) and (3), δ^* is defined as

$$\delta^* = \delta\, [\, g\, \rho_l\, (\rho_l - \rho_g\,) / \mu_l^{\,2}\,]^{\,0.333} \qquad (4)$$

The vapor void fraction can now be determined knowing the film thickness. The void fraction, ε, expressed as the ratio of the cross-sectional area of the vapor to the total cross-sectional area is then

$$\varepsilon = [\, (D_i - 2\,\delta) / D_i\,]^{\,2} \qquad (5)$$

Bell [15] investigated the use of the cocurrent, pressure drop correlations for the reflux or countercurrent condition. The gravity and momentum-change pressure drop expressions, when modified for countercurrent flow, yielded reasonable results; however, no modification for the frictional pressure drop contribution was possible. As stated by Chen et al, [12], that Martinelli-based correlations do not work for reflux condensation. We therefore used the correlation for the interfacial shear stress between the condensate and vapor introduced by Chen et al, [12] for calculating the interfacial shear:

$$\tau_i = C\, \rho_l\, (\, g\, v_l\,)^{\,0.667} Re_l^{\,1.8} \qquad (6)$$

where

$$C = 0.023[\mu_l^{\,1.133} \mu_g^{\,0.2} / (D_i^{\,2} g^{\,0.667} \rho_g \rho_l^{\,0.333})] \qquad (7)$$

The total pressure drop gradient is then obtained with the expression

$$dP/dL = \rho_l(1-\varepsilon)g - \{4\tau_i/D_i + xGt[(xU_l)_{in} - (xU_l)_{out}]/dL\} \qquad (8)$$

Table 2 shows the variations of the different components of the pressure gradient through the reflux condenser starting from the bottom. For the particular case, the tube inside diameter is 34.8 mm, the pressure level is 110 kPa, and the vapor is para-xylene. The inlet vapor flow rate is 27.2 kg/hr and the heat-flux range is 95 to 151 kW/m^2. The findings for this case are typical to those obtained for all other runs conducted. The condensate pressure gradient due to gravity continuously decreases up the tube and reaches a value that is about one third of the inlet value because of the reduced liquid void fraction or condensate film thickness with elevation. The pressure gradient due to the interfacial shear decreases at a much faster rate and becomes about 1,000 times less at the tube top because the vapor velocity and condensate film thickness decrease in the upward direction. Finally, the fraction of the pressure gradient due to momentum change is reduced by almost 10,000 times from the bottom to the top of the tube. Again, the reasons for the reductions are the reduced quality and liquid velocity typical of reflux condensers.

TABLE 2: Pressure Gradient Variations

Tube Length m	Condensate Gravity	Interfacial Shear	Condensate Momentum Change	Total Gradient
inlet	0.0530	0.00191	–	-
0.046	0.0492	0.00127	0.00451	0.0434
0.091	0.0446	0.00075	0.00209	0.0418
0.137	0.0389	0.00036	0.00070	0.0379
0.183	0.0303	0.00009	0.00011	0.0301
exit	0.0154	0.00000	0.00000	0.0154

Heat Transfer Prediction Method

Cocurrent, single-component condensation inside downflow vertical or horizontal plain tubes has been intensively investigated for the last 25 years. Publications by Bell et al. [16], Breber [17], and Palen et. al. [18] are excellent overviews of these developments. However, very little effort was devoted to the countercurrent or reflux condensing arrangement. Only two prediction methods were found in the open literature: the methods of Chen et al. [12] and Chunangad [13].

The correlation of Chen et al. [12] for countercurrent condensation for a single-component vapor is

$$Nu_{cg}=\{[(0.31/Re_l^{1.32})+(Re_l^{2.4}Pr_l^{3.9}/2.37\text{x}\,10^{14})]^{0.333} - C\,Re_l^{1.8}\,Pr_l^{1.3}/771.6\}^{1/2} \quad (9)$$

where

$$h = Nu_{cg}\,k_l\,(g/\nu_l^2)^{0.333} \quad (10)$$

The procedure suggested by Chunangad [13] is as follows. When the superficial liquid Reynolds numbers is less than 30, the Nusselt number is calculated as

$$Nu = 1.1\,Re_l^{-0.333} \quad (11)$$

For liquid Reynolds numbers between 30 and 1600, Kutateladze correlation is used

$$Nu = 0.756\,Re_l^{-0.22} \quad (12)$$

Finally, for Reynolds numbers above 1600, correlation of Labuntsov is used

$$Nu = 0.023\,Re_l^{0.25}\,Pr_l^{0.5} \quad (13)$$

The heat transfer coefficient is calculated from Equations (11), (12), and (13) in the following manner

$$h = Nu\,k_l\,[\rho_l\,(\rho_l - \rho_g)\,g/\nu_l^2]^{0.333} \quad (14)$$

Table 3 presents a comparison of the predicted thermal performance for these two inside condensing prediction methods. The same design conditions used to obtain Tables 2 were again used. This mass flow rate is below the flooding mass flow rate predicted with Equation (1). The variations in the condensate Reynolds number were between 64 and 1200. Note that the shorter exchanger is obtained with the prediction method of Chen et al. [12]. The inside heat transfer coefficients are somewhat larger for most of the tube length except near the top where the condensate film thickness is very small. There is a continuous increase in the heat transfer coefficient from the bottom to the top of the tube because of (1) the reduction of the condensate film thickness and (2) the film is mostly in laminar flow. This comparison shows that there is reasonable agreement between these prediction methods. We decided to use the method of Chen et al. [12] simply because the prediction method is readily available in the open literature.

TABLE 3: Comparison of Heat Transfer Prediction Methods

Quantity	Chen	Chunangad
L, m	0.213	0.26
Inside h, W/m²C	1.02 to 1.99	0.80 to 2.10
Overall h, W/m²C	0.88 to 1.51	0.72 to 1.58
Heat flux, kW/m²	94 to 151	79 to 142

ENHANCED REFLUX CONDENSER RESULTS

The purpose of this section is to quantify the surface-area and vapor-release reduction ratios for reflux condensation based on design methods described above. The presentation of the results is divided into three parts:

1) pure vapors (no noncondensable gas),
2) vapor to gas weight ratios of 100 and 10 (rich mixtures), and
3) equal weight flows of the vapor and noncondensable gas (lean mixtures).

For all the results to be presented, the inlet mixture flow rate was selected that was just below the flooding limit predicted by Equation (1).

A reflux condenser computer program was written based on a one-dimensional pointwise analysis. To represent the shellside performance, the cooling water is assumed to flow in an outside annular passage in a countercurrent manner or entering from the top of the unit. A typical inlet summer temperature of 29.4 °C (85.0 °F) was selected for the cooling water. This program was used to obtain all the following results.

Pure Vapors

Table 4 shows the tube length for complete condensation of both para-xylene and steam at two pressure levels, 110 and 55 kPa. Values were obtained for three cases: with a plain tube, with a frictional pressure drop increase of a factor of about two, and with a condensation enhancement of two. Table 4 shows that the tube length is reduced by a factor of two for para-xylene and by one-third for steam. Also note that doubling the pressure drop has almost no impact on the required tube length. Therefore the predicted performance of the enhanced tube is essentially the same as obtained with the doubling of the heat transfer. The reason for the greater length reduction with papa-xylene is that the condensing heat transfer coefficient is less than that of steam.

TABLE 4: Tube Length for Complete Condensation

Fluid Type	para-xylene		steam	
Pressure, kPa	110	55	110	55
Vapor Flow Rate, kg/hr	30.4	23.6	20.4	15.9
Plain Tube, m	0.25	0.26	0.41	0.43
Pressure Drop Doubled, m	0.26	0.26	0.41	0.44
Heat Transfer Doubled, m	0.13	0.13	0.27	0.29

Rich Mixtures

For the rich-vapor case, the exit vapor flow rates or qualities are calculated with different enhancement scenarios to determine the possible emission reductions for a fixed-length tube. The tube length is selected to obtain about a 10 percent exit quality with a plain-tube unit. Again, the inlet vapor flow rate is selected by trial-and-error to be just below the flooding limit. Tables 5 and 6 show the results for inlet mass-flow ratios of 100 and 10, respectively. The purpose of the investigation is to determine the significance of (1) the mass transfer improvement, (2) the pressure drop increase, and (3) the condensing enhancement. The enhanced tube results are characterized by all enhancement options being doubled.

The most important result consistently displayed in Tables 5 and 6 is that the exit vapor quality can be significantly reduced with the use of enhanced tubes, the maximum reduction reaching a factor of four. This magnitude of emission reduction is very significant. As expected, the reductions are greater for para-xylene because it is the "poorer" heat transfer fluid. Retubing reflux condensers with enhanced tubes can significantly reduce the emissions of volatile organic compounds when mixed with small quantities of a noncondensable gas.

TABLE 5: Exit Qualities with Vapor to Gas Weight Ratio Equal to 100

Fluid Type	para-xylene		steam	
Pressure, kPa	110	55	110	55
Vapor Flow Rate, kg/hr	29.0	22.2	22.7	20.4
Tube Length, m	1.13	1.13	0.98	0.89
Plain Tube	0.102	0.104	0.097	0.095
Mass Transfer Doubled	0.016*	0.029*	0.038*	0.075*
Pressure Drop Doubled	0.103	0.105	0.097	0.096
Heat Transfer Doubled	0.097	0.100	0.090	0.086
Enhanced Tube	0.016*	0.029*	0.038*	0.075*

TABLE 6: Exit Qualities with Vapor to Gas Weight Ratio Equal to 10

Fluid Type	para-xylene		steam	
Pressure , kPa	110	55	110	55
Flow Rate, kg/hr	23.6	18.1	18.1	14.1
Tube Length, m	3.05	3.35	1.77	1.83
Plain Tube	0.103	0.103	0.102	0.103
Mass Transfer Doubled	0.023*	0.039*	0.040*	0.079*
Pressure Drop Doubled	0.104	0.103	0.102	0.103
Heat Transfer Doubled	0.093	0.098	0.101	0.097
Enhanced Tube	0.023*	0.039*	0.040*	0.079*

* Inlet coolant temperature reaches the local coolant temperature and condensation is terminated

Dilute Mixtures

For a dilute-mixture case, equal vapor and gas mass flow rates, the exit quality is usually limited by the coolant temperature. For this case, we determined the tube length required to reach this limiting exit quality for both a plain and an enhanced tube as well as for the fictitious conditions of only the mass transfer, pressure drop, or the condensing heat transfer doubled.

Table 7 shows that length reductions of about a factor of two are obtained with both vapors. These results show that the potential exists for significant surface-area and cost reductions through the use of enhanced tubes in reflux condensers for the control of VOC emissions.

TABLE 7: Tube Length (m) Required to Obtain the Minimum Exit Quality With Equal Inlet Vapor and Nitrogen Flow Rates

Fluid Type	para-xylene		steam	
Pressure, kPa	110	55	110	55
Vapor Flow Rate, kg/hr	7.71	6.35	9.52	7.26
Min. Quality	0.066	0.0136	0.061	0.123
Plain Tube	10.4	9.15	5.49	3.72
Mass Transfer Doubled	5.33	4.88	2.44	1.83
Pressure Drop Doubled	10.7	9.45	5.49	3.72
Heat Transfer Doubled	10.4	9.45	5.03	3.63
Enhanced Tube	5.33	4.88	2.28	1.78

CONCLUSION

The reduction and control of volatile organic compounds (VOCs) contained in noncondensable gas streams is a very serious problem confronting the chemical, petrochemical, and petroleum processing industries. A feasibility analysis clearly showed that the use of enhanced condenser surfaces can significantly reduce VOC emissions by as much as a factor of four or reduce the required surface area as much as a factor of two. The analysis used the best-available published design methods to predict (1) the limiting flooding condition, and (2) the local plain-tube heat transfer and pressure drop performance.

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NOMENCLATURE

Bo	Bond no, $Bo = Di^2 g(\rho_l - \rho_g)/\sigma$
C	constant defined by Equation (7)
c_p	specific heat
D_i	inside tube diameter
Fr	Froude no., $Fr = Q_l\,[g(\rho_l - \rho_g)^3/\sigma^3]^{1/4}$

g	acceleration due to gravity
Gt	inlet superficial mass velocity of vapor
h	heat transfer coefficient
k	thermal conductivity
K_g	Kutateladze no., $K_g = V_g \rho_g^{0.5} [g\sigma(\rho_l - \rho_g)]^{0.25}$
L	tube length
Nu_{cg}	Nusselt no. defined by Equation (10)
Nu_c	Nusselt no. defined by Equation (14)
P	pressure
Pr	Prandtl no., $Pr = c_p \mu / k$
Q	volume flow rate per unit wetted perimeter
Re	superficial Reynolds no.
U_l	actual liquid velocity, $= xGt/\rho_l (1-\varepsilon)$
V_g	superficial gas velocity, $= x\, Gt / \rho_g$
x	quality

Greek Symbols

δ	condensate film thickness
δ^*	dimensionless condensate film thickness defined by Equation (4)
ε	vapor void fraction
μ	dynamic viscosity
ν	kinematic viscosity
σ	surface tension of liquid
ρ	density
τ_i	interfacial shear stress

Subscripts

g	gas phase
in	inlet
l	liquid phase
metal	tube metal
out	outlet
w	water

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