Internal Dropwise Condensation: Modeling and Experimental Framework for Horizontal Tube Condensers

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Abstract

Many researches have been focused on internal condensation with machined microstructure patterns on internal surfaces that can improve heat transfer performance up to 400% comparing to smooth surfaces. However, microstructure patterns are not effective for all geometries such as micro-channels and other complicated shapes. This study uses a dropwise condensation (DWC) promoter that can be easily applied to different geometries to improve internal condensation heat transfer. The main objectives of this study are to investigate internal condensation heat transfer by the dropwise condensation promoter via i) developing an internal dropwise condensation model and ii) conducting internal condensation experiments with the dropwise condensation promoter. The dropwise condensation model was modified to the available internal condensation model. The experiments for internal steam condensation were conducted for aluminum and copper tubes with external fins and an internal coating for dropwise condensation. The experimental results are compared to the results from several correlations in the literature and the model developed in this study. It is found that most correlations in the literature overestimate internal condensation heat transfer coefficients at low mass fluxes. The dropwise condensation model shows that dropwise condensation promoters can enhance internal condensation heat transfer coefficients. This conclusion is verified by experimental results demonstrating that the hydrophobic dropwise condensation promoter used in this study can enhance internal condensation heat transfer coefficients by 50%.

Keywords: internal condensation, dropwise condensation, dropwise condensation promoter, hydrophobic coatings, and internal dropwise condensation model.

1 Nomenclature

2	A	area, m^2
3	c	factor determined by surface inclination
4	C_p	heat capacity, $kJ/kg\cdot K$
5	D	diameter of tube, m
6	G	mass velocity, $kg/m^2\cdot s$
7	g	gravitational acceleration, m/s^2
8	h	heat transfer coefficient, $W/m^2\cdot K$
9	H	latent heat, kJ/kg
10	J_G	dimensionless vapor velocity, $J_G = xG/[gD\rho_v(\rho_f - \rho_v)]^{0.5}$
11	K	adjustment for degrees of subcooling, flow regime and geometry
12	k	thermal conductivity, $W/m\cdot K$
13	L	length of test tube, m
14	$\dot{m}$	mass flow rate, kg/s
15	$N(r)$	large drop distribution, m^{-2}
16	$n(r)$	small drop distribution, m^{-2}
17	Nu	Nusselt number, $Nu = hD/k_f$
18	N_s	nucleation site density, m^{-2}
19	N_t	number of tubes
20	Pr	Prandtl number, $Pr = C_p\mu_f/k_f$
21	p_{red}	reduced pressure, $p_{red} = p/p_{crit}$
22	Re	Reynolds number, $Re = GD/\mu_f$
23	q	heat transfer rate, W
24	q''	heat flux, W/m^2
25	R	thermal resistance, K/W
26	r	radius, m
27	T	temperature, K
28	x	vapor quality
29	X_{tt}	Martinelli parameter, $X_{tt} = (\mu_f/\mu_v)^{0.1} (\rho_v/\rho_f)^{0.5} ((1-x)/x)^{0.9}$

31 Greek letters

32	δ	thickness of the coating layer, m
33	ϵ	void fraction
34	μ	fluid viscosity, $kg/m\cdot s$
35	η	efficiency
36	θ	contact angle, degree
37	ρ	density of condensate, kg/m^3
38	σ	surface tension, N/m
39	τ	sweeping period, s
40	ϕ	angle in Fig. 1

42 Subscripts

43	a	advancing
44	air	air
45	c	condensate
46	coat	coating

1	e	effective
2	f	fluid
3	fin	fin
4	i	interfacial
5	in	inside
6	LH	latent heat
7	li	liquid in
8	lo	liquid out
9	max	maximum
10	min	minimum
11	out	outside
12	r	receding
13	red	reduced
14	sat	saturation
15	SH	sensible heat
16	St	stratified flow
17	t	tube
18	tot	total
19	v	vapor
20	w	surface

21		
22	<u>Abbreviation</u>	
23	DWC	dropwise condensation
24	FWC	filmwise condensation
25	HTC	heat transfer coefficient
26	ID	inner diameter
27	OD	outer diameter
28		

1. INTRODUCTION

For the internal condensation removing latent heat of a working fluid flowing inside of a tube by a coolant fluid at outside of the tube, extended surface such as fins is typically used at a coolant side to increase heat transfer area, which is resulting in the increase of the condensation heat transfer rate. Most of thermodynamic properties are available from NIST data base [1] and water and ammonia commonly used in heating and cooling applications are studied by many researchers [2]. As working fluids, hydrocarbons such as pentanes, hexane and heptane are often used in geothermal plants due to low toxicity and boiling points, and other non-toxic working fluids and their mixtures are continuously studies [3-5].

Many researchers have been studied the effects of geometries on internal condensation heat transfer and its enhancement. Dobson and Chato *et al.* [6] experimentally showed that condensation heat transfer coefficients at a shear-dominated flow regime in smooth horizontal tubes were affected by mass flux and quality. Using a helically coiled, concentric tube-in-tube heat exchanger, heat transfer coefficients of HFC-134a were improved by 33% - 53%, but pressure drops increased by 29% - 49% comparing to straight tubes [7]. Briggs *et al.* [8-9] measured heat transfer and pressure drop of CFC-113 during internal condensation using micro-fin tubes and wire inserts and Bergelin *et al.* [10] tested turbulent flow for air/water mixture and organic vapors in vertical tubes. Eckels and Pate *et al.* [11] tested R12 and R134a in smooth tubes and reported that heat transfer coefficients of R134a was higher than R12 by 25% - 30%. Torikoshi and Ebisu *et al.* [12] found that the condensational heat transfer coefficients for R134a, R32 and its mixture were larger than those for R22.

Micro-and-macro structures on interior wall of condenser tubes are common methods for heat transfer enhancement for cooling and refrigeration applications. Cavallini *et al.* [13] presented a new computational procedure to obtain the heat transfer coefficient and pressure drop during internal condensation with smooth, low-finned, micro-finned, and cross-grooved tubes and reported that micro finned tubes enhanced heat transfer coefficients by 80% - 140 %, but pressure drop was increased by 20% - 80% comparing to smooth tubes. Liebenberg *et al.* [14] presented a review paper for the condensation of refrigerant in horizontal micro-finned tubes. Koyama *et al.* [15] showed that heat transfer coefficients and pressure drop of binary refrigerant mixtures of R22 and R114 in a horizontal tube with internal spiral grooves, increased with an increase of helix angles. Fang *et al.* [16] used a hydrophobic surface as an internal dropwise condensation promoter for silicon micro channels and reported the

improvements of local heat flux and heat transfer coefficients and the increase of pressure over hydrophilic surfaces in annular/stratified region.

The condensing vapor inside a condenser tube is generally divided into several flow regimes; vapor flow, annular flow, stratified flow, plug and slug flow, and finally single phase flow regimes. As the flow starts to cool, a thin film is formed on the wall of the tube as annular flow regime with increasing the thickness of the condensate film and the excess liquid will fall to the bottom forming as a stratified flow regime. Condensates and vapor will fill the whole cross section of the tube and form plug and slug flow. Finally it will collapse into a single phase flow when all vapor condensed. According to Thome [17-18], the pressure drop across the condensation section with the mass velocity less than 400 kg/m²-s are insignificant with respect to the saturation pressure.

Many correlations for internal condensation were established for different flow regimes. In order to determine the flow regime for internal flow, dimensionless vapor velocity, J_G and Martinelli parameter, X_{tt} used by Cavallini [19] are calculated by

$$J_G = \frac{xG}{[gD\rho_v(\rho_f - \rho_v)]^{0.5}} \quad (1)$$

$$X_{tt} = \left(\frac{\mu_f}{\mu_v}\right)^{0.1} \left(\frac{\rho_v}{\rho_f}\right)^{0.5} \left(\frac{1-x}{x}\right)^{0.9} \quad (2)$$

where x and μ_v are the vapor quality and vapor viscosity, respectively. For annular flow, J_G is greater than 2.5. J_G less than 2.5 and X_{tt} less than 1.6 are obtained in annular-stratified flow and J_G less than 2.5 and X_{tt} greater than 1.6 is considered a stratified-slug flow regime.

Many heat transfer correlations are based on latent heat, Reynolds (Re), and Prandtl (Pr) numbers. Calvallini *et al.* [19] developed the model for fully stratified flow for $J_G = 2.5$, and the equation for stratified heat transfer coefficient, h_{st} for organic refrigerants, which is given by,

$$h_{st} = 0.725 \left[1 + 0.82 \left(\frac{1-x}{x} \right)^{0.268} \right]^{-1} \left[\frac{k_f^3 \rho_f (\rho_f - \rho_v) g H_{fg}}{\mu_f D (T_{sat} - T_w)} \right]^{0.25} + \frac{h_f (\arccos(2\epsilon - 1))}{\pi} \quad (3)$$

where $h_f = h_l(1-x)^{0.8}$ and $h_l = \frac{k_f}{D} 0.023 Re_f^{0.8} Pr_f^{0.4}$.

Nusselt number, Nu , by Jester *et al.* [20] at horizontal stratified flow regime for variety of fluids is given by,

$$Nu = 0.725 \left[\frac{\rho_f (\rho_f - \rho_v) H_{fg} g D^3 \epsilon^3}{k_f \mu_f (T_{sat} - T_w)} \right]^{0.25}, \quad Nu = \frac{hD}{k_f} \quad (4)$$

where the void fraction, ϵ is calculated from Zavi [21],

$$\epsilon = \frac{x}{\left[x + (1-x) \left(\frac{\rho_v}{\rho_f} \right)^{\frac{2}{3}} \right]} \quad (5)$$

Shah *et al.* [22] provided internal correlations for wide range of fluids and flow regimes which depends on Re and Pr numbers with pressure and quality adjustments,

$$Nu = 0.023 Re_f^{0.8} Pr_f^{0.4} \left[1 + \frac{3.8}{p_{red}^{0.38}} \left(\frac{x}{1-x} \right)^{0.76} \right] \quad (6)$$

where Re_f is fluid Reynolds number and p_{red} is reduced pressure which are given by,

$$Re_f = \frac{GD}{\mu_f} \quad (7)$$

$$p_{red} = \frac{p}{p_{crit}} \quad (8)$$

p is saturation pressure and p_{crit} is critical pressure of working fluids.

Bivens and Yokozeki [23] modified the Shah correlation by fitting for wider mass flux ranges and flow regimes with different refrigerants,

$$Nu = Nu_{shah} \left(0.78738 + \frac{6187.89}{G^2} \right) \quad (9)$$

Dobson and Chato *et al.* [6] proposed the correlation for the stratified/annular flow regime for variety of fluids with similar dependency as Shah's correlation, which is given by

$$Nu = 0.023 Re_f^{0.8} Pr_f^{0.4} \left[1 + \frac{2.22}{X_{tt}^{0.89}} \right] \quad (10)$$

Most of the structures used in literature for internal condensation are mechanically machined. However it is a challenge to have surface structures when a small size dimension is required for applications. Therefore, applying coatings can be considered for wide variety of geometries with enhancements of heat transfer, but without higher pressure drop for the condenser unit, which can result in the increase of thermal efficiency and system performance, and reducing the cost of operation. This study is to investigate a feasibility of using a dropwise condensation promoter to improve a heat transfer performance of compact internal air cooled condensers mathematically and experimentally. The internal dropwise condensation model is developed focusing on annular and stratified flow regimes and experimental results are compared to the predictions from the developed model and best available correlations in literature.

2. INTERNAL CONDENSATION HEAT TRANSFER MODELING

Figure 1: Illustrative cross-sectional view of the condensing tube for (a) filmwise condensation and (b) dropwise condensation (assumed)

Figure 1 shows the annular and stratified flow regimes during internal condensation. Filmwise condensation (FWC) is shown in Figure 1a and a dropwise internal condensation is shown in Figure 1b. Figure 1b shows a stratified flow with a hydrophobic coating as a dropwise condensation promoter applied on the internal tube wall. Condensate drops grow either by direct condensation or by coalescence with neighboring drops. When the drops become large enough, they begin to fall and swipe off other drops in the path or directly fall onto the bottom and eventually being accumulated into a stream on the bottom of the tube. With a hydrophobic coating, dropwise condensation will be expected at the tube surface except the surface for the condensate stream on the bottom, which may enhance internal condensation heat transfer.

The internal condensation heat transfer in a horizontal tubular channel is greatly affected by the flow regimes that are determined by hydrodynamic and thermodynamic equilibriums. A low or moderate mass velocity G (low vapor velocity and low interfacial shear forces) conditions is considered in this study, because dropwise condensation with hydrophobic surfaces is most likely occurring at stratified, wavy, slug, or plug flow regimes. For these flow regimes, filmwise condensation at the bottom of a tube can be established relatively well. The heat transfer coefficient, h_{FWC} is obtained by modifying the external filmwise condensation equation on a horizontal tube [24],

$$h_{FWC} = F \left[\frac{\rho_f(\rho_f - \rho_v)gH_f g k_f^3}{D\mu_f(T_{sat} - T_w)} \right]^{0.25} \quad (11)$$

where the factor F depends on the angle 2ϕ in Figure 1a.

For the flow regimes of wavy, slug, and plug, heat transfer coefficients vary considerably from the top to the bottom of a tubular channel depending on the vapor shear forces. The condensation model can be split into top and bottom portions, as seen in Figure 1b. The bottom where condensate stream flows is dealt with differently from the rest of the surface. The heat transfer coefficient, h_{bottom} was obtained by Rosson and Myers et al [25] considering the analogy between heat and momentum transfer,

$$h_{bottom} = \frac{k_f}{D} \left[\frac{\phi \left(8 \left(\frac{GD}{\mu_f} \right)^{0.5} \right)}{5 \left(1 + \left(\frac{1}{Pr_f} \right) \ln(1 + 5Pr_f) \right)} \right] \quad (12)$$

where Pr_f is the Prandtl number of fluid.

Equation (12) is used for bottom portion of the tube covered by the liquid layer shown in Figure 1b.

To get a heat transfer coefficient of filmwise condensation at the top part of the tube, Equation (11) is used with the value of F [25] which is given by,

$$F = 0.31 \left[\frac{GD}{\mu_v} \right]^{0.12} \quad (13)$$

For the dropwise condensation at the top part of the tube, it is assumed that condensate droplets experience a typical dropwise condensation cycle (initial drop generation, drop growth by condensation and/or coalescence, and sweep-off by large falling drops). The dropwise condensation model proposed by Kim and Kim [25] is used to get a heat transfer coefficient of dropwise condensation at the top part of the tube.

Figure 2: Heat transfer model through a droplet with contact angle ϑ on a hydrophobic surface [26]

The dropwise condensation model proposed by Kim and Kim [26] is consisted of the thermal resistance analysis of heat transfer of a single droplet and population density of condensates. The heat

transfer through a sing droplet can be calculated with considering all thermal resistances corresponding to temperature drops between vapor and a condensing surface as the droplet sits/hangs on the hydrophobic surface as shown in Figure 2. When heat is transmitted from vapor to liquid, the interfacial resistance produces the temperature drop, ΔT_i ,

$$\Delta T_i = \frac{q_d}{h_i 2\pi r^2 (1 - \cos \theta)} \quad (14)$$

where q_d , h_i , r , θ are heat transfer through a single drop, interfacial heat transfer coefficient, drop radius, and contact angle of the droplet, respectively.

Temperature drop due to conduction heat transfer through the droplet, ΔT_{drop} is given by,

$$\Delta T_{drop} = \frac{q_d \theta}{4\pi r k_c \sin \theta} \quad (15)$$

where k_c is condensate droplet thermal conductivity.

Hydrophobic promoter coating significantly contributes to the total temperature drop, and the corresponding temperature drop, ΔT_{coat} is,

$$\Delta T_{coat} = \frac{q_d \delta}{k_{coat} \pi r^2 \sin^2 \theta} \quad (16)$$

where δ is hydrophobic coating thickness.

Also, a temperature drop due to the condensate-drop curvature, ΔT_{curv} is [28],

$$\Delta T_{curv} = \frac{2T_{sat}\sigma}{H_{fg}r\rho} \quad (17)$$

where σ and ρ are condensate surface tension and density, respectively.

Minimum viable drop radius, r_{min} , is determined by wall subcooling, $T_{sat} - T_w$, [28] and it is given as,

$$r_{min} = \frac{2T_{sat}\sigma}{H_{fg}\rho(T_{sat} - T_w)} \quad (18)$$

Temperature drop due to drop curvature, ΔT_{curv} , in terms of minimum drop radius, r_{min} , is given as,

$$\Delta T_{curv} = \frac{r_{min}}{r} (T_{sat} - T_w) \quad (19)$$

Overall temperature difference between the vapor and the condensing surface, $T_{sat} - T_w$, equals the sum of the four temperature drops from Equations (14-17, 19),

$$T_{sat} - T_w = \Delta T_i + \Delta T_{drop} + \Delta T_{coat} + \Delta T_{curv} \quad (20)$$

Heat transfer rate, q_d , through a drop of radius r is expressed as,

$$q_d = \frac{(T_{sat} - T_w) \pi r^2 \left(1 - \frac{r_{min}}{r}\right)}{\left[\frac{\delta}{k_{coat} \sin^2 \theta} + \frac{r \theta}{4 k_c \sin \theta} + \frac{1}{2 h_i (1 - \cos \theta)}\right]} \quad (21)$$

In addition to the heat transfer through a single droplet, a drop size distribution should be determined in order to calculate the overall heat transfer rate. For small condensate drops governed by direct condensation, the drop size distribution, $n(r)$ can be obtained from the population density theory as used by Kim and Kim [26],

$$n(r) = \frac{1}{3 \pi r_e^3} \left[\frac{r_{max}}{r_e} \right]^{\frac{2}{3}} \left[\frac{r(r_e - r_{min})}{r - r_{min}} \right] \left[\frac{A_2 r + A_3}{A_2 r_e + A_3} \right] \exp(B_1 + B_2) \quad (22)$$

$$A_1 = \frac{\Delta T}{2 \rho_c H_{fg}}, A_2 = \frac{\theta(1 - \cos \theta)}{4 k_c \sin \theta}, A_3 = \frac{1}{2 h_i} + \frac{\delta(1 - \cos \theta)}{k_{coat} \sin^2 \theta}$$

$$B_1 = \frac{A_2}{\tau A_1} \left[\frac{r_e^2 - r^2}{2} + r_{min}(r_e - r) - r_{min}^2 \ln \left(\frac{r - r_{min}}{r_e - r_{min}} \right) \right],$$

$$B_2 = \frac{A_3}{\tau A_1} \left[r_e - r - r_{min} \ln \left(\frac{r - r_{min}}{r_e - r_{min}} \right) \right]$$

$$\tau = \frac{3 r_e^2 (A_2 r_e + A_3)^2}{A_1 (11 A_2 r_e^2 - 14 A_2 r_e r_{min} + 8 A_3 r_e - 11 A_3 r_{min})}$$

For large condensate drops, the drop size distribution, $N(r)$ is obtained from the empirically established equation by Fevre and Rose [27],

$$N(r) = \frac{1}{3 \pi r_{max}^2} \left(\frac{r}{r_{max}} \right)^{-\frac{2}{3}} \quad (23)$$

The maximum drop radius (the size of departure drops), r_{max} can be estimated based on the force balance between surface tension and gravity as,

$$r_{max} = \left[\frac{6 c (\cos \theta_r - \cos \theta_a) \sin \theta \sigma}{\pi (2 - 3 \cos \theta + \cos^3 \theta) \rho g} \right]^{0.5} \quad (24)$$

The effective radius, r_e is defined as the boundary between the small drop distribution, $n(r)$ and large drops distribution, $N(r)$ with assuming the nucleation sites forming a square array. r_e can be written as,

$$r_e = (4 N_s)^{-\frac{1}{2}} \quad (25)$$

where N_s is the number of nucleation sites on unit area of condensing surface.

Steady-state dropwise condensation heat transfer rate per unit area of condensing surface can be obtained by multiplying heat transfer rate through a single drop with the population density,

$$q''_{top} = \int_{r_{min}}^{r_e} q_d(r)n(r)dr + \int_{r_e}^{r_{max}} q_d(r)N(r)dr \quad (26)$$

Once total heat flux obtained from the dropwise condensation model, a heat transfer coefficient, h_{top} of dropwise condensation could be obtain as,

$$h_{top} = K \frac{q''}{T_{sat} - T_w}, \quad K = \frac{A_{out}}{A_t} \quad (27)$$

where K is the ratio of cross section areas of outlet and total tube to account for the effects of tube geometry, flow conditions, and subcooling temperatures at the top portion of the tube.

In order to calculate the condensation heat transfer coefficient for whole stratified flow regime for the annulus test section of a tube, heat transfer from the top and bottom is adjusted as given by,

$$q_{top} = h_{top}A_{top}(T_{sat} - T_w), \quad A_{top} = DL\phi \quad (28)$$

$$q_{bottom} = h_{bottom}A_{bottom}(T_{sat} - T_w), \quad A_{bottom} = DL(\pi - \phi) \quad (29)$$

where L , ϕ , A_{top} and A_{bottom} are test section length, angle represented in **Figure 1**, and the top and bottom heat transfer areas, respectively. h_{top} and h_{bottom} are heat transfer coefficients from the top and bottom portions.

Total heat flux, q''_{tot} and heat transfer coefficients for dropwise condensation, h_{DWC} can be obtained as following,

$$q''_{tot} = \frac{q_{top} + q_{bottom}}{A_{top} + A_{bottom}} = (T_{sat} - T_w) \left[h_{top} \left(\frac{\phi}{\pi} \right) + h_{bottom} \left(1 - \frac{\phi}{\pi} \right) \right] \quad (30)$$

$$h_{DWC} = \frac{q''_{tot}}{T_{sat} - T_w} = h_{top} \left(\frac{\phi}{\pi} \right) + h_{bottom} \left(1 - \frac{\phi}{\pi} \right)$$

Calvallini [19] provided the relations between void fraction, ϵ and angle, ϕ and it is given by,

$$2\phi = \pi - \cos^{-1}(2\epsilon - 1) \quad (31)$$

Equations (28 - 31) are also used to modify filmwise condensation equations from Rosson and Myers [25] to include combined effects of test section geometries, phase change and fluid flow.

2.3 Model results and discussion

Heat flux and internal heat transfer coefficients results with varying parameters (mass flux, exit quality, subcooling temperature, and contact angle) from the developed dropwise condensation model

are shown in Figures 3 to 6. Figures 3 to 5 contain the results of the dropwise condensation and the filmwise condensation (modified from Rosson and Myers [25]) and Figure 6 shows heat flux and internal heat transfer coefficients versus contact angle.

Mass flux, exit quality, subcooling temperature, and contact angles vary from 2 - 20 kg/m²-s, 0 - 1, 12 - 30 K and 90° - 150°, respectively. If parameters are not used as variables, constant values of 10 kg/m²-s, 0.5, 30K and 90° are used. In addition, constant dimensions and conditions are assumed: $D = 4$ mm, $\delta = 10$ μm, $h_i = 15.7 \times 10^6$ W/m²-K, $k_{\text{coat}} = 0.2$ W/m-K, $T_{\text{sat}} = 368$ K, and $N_s = 2.5 \times 10^{11}$ m⁻².

As shown in **Figure 3** for the results of the dropwise and filmwise condensation models, heat flux and internal heat transfer coefficient increase as mass flux increases. It is expected that more condensation heat is transferred with a higher mass flux and internal heat transfer coefficient increase as the mass flux increases.

Figure 3: Heat flux and internal HTC vs. mass flux of water condensation

Figure 4: Heat flux and internal HTC vs. exit quality of water condensation

Figure 4 shows that heat fluxes and heat transfer coefficients by the dropwise condensation are higher than those of filmwise condensation and the heat fluxes. The heat fluxes and heat transfer coefficients are higher at higher quality and decrease as the quality decrease, which indicates that those are decreasing as the increase of the amount of condensate in the tube. Heat flux and heat transfer coefficient decrease sharply as the exit quality decreases from 0.1 to 0. At low exit quality, heat transfer from the bottom portion of tube where condensate accumulates becomes dominant. It is noted that dropwise condensation will make the starting length of plug/slug flow in the test section shorter than that with filmwise condensation due to higher heat flux transferred by dropwise condensation [16].

As shown in Figure 5, heat flux increases as the subcool temperature increases. Heat transfer coefficient of dropwise condensation remains relatively constant while that of filmwise condensation decreases. Note that for the filmwise condensation, steam condenses quickly and the amount of condensate increases as the subcooling temperature increases. This can increase the liquid layer thickness and decrease the heat transfer coefficient of the filmwise condensation. Thus, dropwise condensation model provides higher heat transfer coefficients than those of filmwise condensation model.

Figure 5: Heat flux and internal HTC vs. subcooling temperature of water condensation

Figure 6 shows heat flux vs. contact angle of water condensation. As shown in Figure 6, the heat flux and internal heat transfer coefficient of dropwise condensation decreases as contact angle increases, because thermal resistance of the droplets increases as the increase of contact angle on the surface.

Figure 6: Heat flux vs. contact angle of water condensation

3. CONDENSATION EXPERIMENTS

3.1 Experimental approach

The effects of dropwise condensation in air cooled condensers were investigated. A condensation test setup was constructed and water was used as the working fluid. The tests sections are consisting of finned circular tubes and the hydrophobic coating provided by NEI Corporation (Somerset, NJ). Results are compared with various correlations from literature. Experiments were performed at saturation pressure of 88 kPa. Table 1 shows thermal properties of water taken from Engineering Equation Solver (EES) [30] at the saturation pressure.

Table 1: Testing fluid properties

Test fluid	Saturation temperature	Thermal conductivity	Viscosity (liquid)	Heat capacity	Latent heat
Units	[K]	[W/m-K]	[kg/m-s]	[kJ/kg-K]	[kJ/kg]
Water	369	0.66	0.000294	4.21	2267

3.2 Experiment setup

Three different types of air cooled condensers were used for the experiment: aluminum and copper tubes without a coating, and a copper tube with a hydrophobic coating provided by NEI Corporation.

Each test tube was 24 in. (609.6 mm) in length with 20 in. (508 mm) of the finned section. The tube OD and ID were 0.25 in. (6.35 mm) and 0.15 in. (3.81 mm), respectively. Individual fin OD was 0.62 in. (15.7 mm), thickness was 0.03 in. (0.76 mm), and fin to fin spacing was 0.09 in. (2.29 mm) as shown in Figure 7.

Figure 8 shows the experiment apparatus consisted of a boiling chamber and the finned air cooled condenser tubes. The condenser tube was mounted horizontally. The end of the condenser tube was open to the atmosphere. All condensation experiments were done in near atmospheric pressure (about 88 kPa). External convection heat transfer by air flow around the finned condenser tubes was used for the condensation. The steam inlet temperature and air temperature around the tube were about 95 °C (368 K) and 25 °C (298 K), respectively. Outlet temperatures were also taken and the pressure drop across all test sections were less than 3 kPa. All temperature measurements were taken by T-type thermal couples with uncertainty of $\pm 0.5K$. Pressure measurements are taken by absolute pressure gages.

Steam mass flow was controlled by a high pressure needle valve. Once steady state conditions reached, a mass flow rate of condensate was obtained by collecting only liquid at the end of the condenser tube for 3 minutes and repeated three times to get the average flow rate. The total mass flow rates were similarly obtained using an ice bath. The ice bath condensed all steam at the end of the tube and the weight change of the ice bath was measured for 3 minutes and repeated three times to get an average mass flow rate. Once the total mass flow rate and condensate flow rate were obtained at the outlet of tube, the exit quality was calculated by,

$$x = \frac{\dot{m}_{tot} - \dot{m}_c}{\dot{m}_{tot}} \quad (32)$$

where x is a quality calculated with a condensate flow rate $\dot{m}_c$ and a total mass flow rate $\dot{m}_{tot}$.

Figure 7: Dimensions of air cooled condenser tube and a test section, labels (1) finned air cooled condenser tube (2) outlet pressure and temperature sensors

Figure 8: Boiling chamber of test setup, labels: (1) pressure relief valve (2) pressure sensor (3) temperature sensor, (4) needle valve (5) vapor generation chamber (6) condenser inlet pressure and temperature sensors

3.3 Data reduction

Heat transfer from the internal condenser tube is given in Equation (33),

$$q = \dot{m}(H_{in} - H_{out}) \quad (33)$$

q (W) is heat transfer from the condenser tube; $\dot{m}$ is the mass flow rate; H_{in} and H_{out} are the enthalpy values at the inlet and outlet of the tube

The enthalpies are obtained using Engineering Equation Solver (EES) [30]. From the equation (33), q of the test section is obtained and used for thermal resistance analysis to obtain internal condensation heat transfer coefficients,

$$q = \Delta T / R_{tot} \quad (34)$$

$$R_{tot} = R_{in} + R_{coat} + R_t + R_{out}, R_{tot} = \frac{1}{UA_{in}} \quad (35)$$

R_{tot} is a total thermal resistance of the test section. ΔT is the temperature difference between a tube interior and surrounding air temperatures. U is an overall heat transfer coefficient.

R_{tot} includes the sum of the thermal resistances of the internal condensation (R_{in}), hydrophobic coating (R_{coat}), tube wall (R_t) and external heat transfer (R_{out}) including the effect of fins, and these are obtained as followings, respectively ,

$$R_{in} = \frac{1}{h_{in}A_{in}} \quad (36)$$

A_{in} and h_{in} are a total internal surface area and an internal condensation heat transfer coefficient.

The thermal resistance of hydrophobic coating is calculated by,

$$R_{coat} = \frac{\ln\left(\frac{D_{in}}{D_{in}-2\delta}\right)}{2\pi k_{coat}LN_t} \quad (37)$$

N_t , L , D_{in} , δ , and k_{coat} are numbers of tubes, length of each tube, inner diameter of the tube coating thickness, and coating thermal conductivity, respectively.

The tube thermal resistance is obtained by,

$$R_t = \frac{\ln\left(\frac{D_{out}}{D_{in}}\right)}{2\pi k_tLN_t} \quad (38)$$

D_{out} and k_t are outer diameter and thermal conductivity of condenser tube.

The external thermal resistance is found by,

$$R_{out} = \frac{1}{h_{out}A_{fin}\eta_{fin}} \quad (39)$$

A_{fin} (m^2), h_{out} ($W/m^2\cdot K$), η_{fin} is the total fin surface area, external heat transfer coefficient, and fin efficiency of annular tubes for the condenser tubes.

Fin efficiency is given by,

$$\eta_{fin} = \frac{C_2(K_1(mr_1)I_1(mr_{2c}) - I_1(mr_1)K_1(mr_{2c}))}{I_0(mr_1)K_1(mr_{2c}) - K_0(mr_1)I_1(mr_{2c})} \quad (40)$$

$$r_{2c} = r_2 + \left(\frac{t}{2}\right), C_2 = \frac{2r_1}{r_{2c}^2 - r_1^2}, m = \sqrt{\frac{2h_o}{k_t t}}$$

$$r_1 = \frac{d_i}{2}, r_2 = \frac{d_o}{2}$$

I_0 , I_1 , K_0 , and K_1 are the modified Bessel functions of the First and Second kinds; and t is thickness of the fins.

The external heat transfer coefficient with assuming air velocity of 0.1 m/s is calculated using the correlation of Hilpert [31] by,

$$h_{out} = \frac{Nu_{out}k_{air}}{D_{out}} \quad (41)$$

$$Nu_{out} = 0.989 Re_{air}^{0.33} Pr_{air}^{1/3}, 0.4 < Re_{out} < 4 \quad (42)$$

$$Re_{out} = \frac{\rho_{air} V_{air} D_{out}}{\mu_{air}} \quad (43)$$

3.4 Experimental Results

Figure 9 shows the mass flux versus the quality plotted on a flow regime map. As shown in figure, the corresponding flow regimes of the experiments for water are mostly in the annular and the stratified/annular flow regions. The copper tube with a hydrophobic coating shows the lowest exit quality, which indicates that the hydrophobic coating on a copper tube effectively condenses steam comparing to the other tested tubes.

Figure 9: Mass flux vs. quality

The measured heat flux versus the mass flux is showed in Figure 10. The highest heat flux is obtained with the hydrophobic coating as shown in the figure. In general, heat flux increases as mass flux increases except the mass flux range of 10 - 13 kg/m²-s. The heat fluxes with the hydrophobic coated tube are improved by about 20 % over the non-coated tubes.

The calculated thermal resistance versus mass flux is shown in Figure 11 with the individual thermal resistances. To obtain the thermal resistance of the hydrophobic coating by NEI, a thermal conductivity of 0.2 W/m-K, contact angle of 90°, and thickness of 10 µm are assumed, respectively. As shown in the figure, the internal thermal resistances are dominant and the internal thermal resistances of the

1 hydrophobic coating on a copper tube are lower than those of other tubes. Note that the external
2 thermal resistances are kept less than 50% of the total thermal resistances by the increase of heat
3 transfer area with the fins adding on the external surface of the tube. The thermal resistances of tubes
4 and the coating are negligible comparing to other thermal resistances.

5

Figure 10: Heat flux vs. mass flux

Figure 11: Thermal resistance vs. mass flux

Figures 12 and 13 show overall and condensation heat transfer coefficients obtained from experiments. In general, heat transfer coefficients increase as the mass flux increase except the mass flux range of 10 - 13 kg/m²-s. The overall and internal condensation heat transfer coefficients of the hydrophobic coating are improved by about 20% and 50%, respectively comparing to those of the non-coated tube. This result indicates that there is large potential to improve the heat transfer of internal condensation by the coatings that internally promote a dropwise condensation.

The heat transfer coefficients from the experiments and correlations are compared in Figure 14. The correlations developed by Rosson and Myers [25], Dobson [6], Jester [20], and Shah [22] of for internal condensation heat transfer coefficients are used for the comparison. The experimental results are also compared to the predictions of the heat transfer coefficient for the hydrophobic coating by the developed dropwise condensation model in this study. As shown in the figure, the internal condensation heat transfer coefficients obtained by the correlations generally over-predicted condensation heat transfer coefficients more than 50% and the predictions by the dropwise condensation model agree with the experimental results within 50%. Note that the subcooling temperatures used in the experiments are in the range of 15K - 35K; however, many of these correlations are not appropriate for the range of a subcooling.

Figure 12: Overall heat transfer coefficients vs. mass flux

Figure 13: Condensation heat transfer coefficients vs. mass flux

Figure 14: Correlation condensation HTC vs. experimental condensation HTCs

3.5 Uncertainty analysis

Uncertainties were calculated based on the NIST technical note 1297 [32] for temperature, pressure, flow rate measurements, which are given by

$$U_Y = \sqrt{\sum_i \left(\frac{\partial Y}{\partial X_i} \right)^2 U_X^2} \quad (47)$$

Where X_i and Y represent measured and calculated variables, respectively. U_X and U_Y are the measured and calculated uncertainties, respectively.

It is assumed that the uncertainties of $\pm 0.5K$ for Inlet, outlet, and air temperature, 5% for exit qualities, and ± 1 g/min for a mass flow rate, respectively. The maximum uncertainties of heat flux, overall heat transfer coefficients, and internal HTCs are about 30 %, 50 % and 80 %, respectively.

5. CONCLUSION AND SUGGESTED FUTURE WORKS

This study developed an internal dropwise condensation model and conducted internal condensation experiments with a dropwise condensation promoter to investigate the enhancement of the internal condensation heat transfer by the dropwise condensation.

The internal condensation heat transfer model was developed based on separating heat transfer coefficients for the top and the bottom part of a tube. The internal condensation performance with the dropwise condensation promoter was better than those without promoters for water. The heat fluxes and overall heat transfer coefficients with the mass flux range up to $20 \text{ kg/m}^2\text{-s}$ increased by 20% comparing to non-coated copper and aluminum tubes. The internal heat transfer coefficients of the coated tube were improved by 50%.

The heat transfer coefficients obtained in the literature generally over predicted the experimental results by more than 50%; however the dropwise condensation model developed in this study predicted the experimental results within 50%.

For the further studies, the integrated modeling and experimental approaches should be considered for several other working fluids with the consideration of the effects of geometries, operating conditions, and internal flow regimes.

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