Table 8 . . . . Heat Transfer Coefficients for Film Type Condensation

t_f = *liquid film temperature* = $t_{sat} - 0.75\ \Delta t$

1. Vertical Surfaces, height L, Laminar Condensate Flow, $N_{Re} = 4\Gamma/\mu_f < 1800$	$h = 1.13\ F_1(h_{fg}/L\Delta t)^{1/4}$	(1)
	$h = 1.11\ F_2(b/w_l)^{1/3}$	(2)
Turbulent Flow, $N_{Re} = 4\Gamma/\mu_f > 1800$	$h = 0.003\ (F_1)^2(\Delta t L/\mu_f{}^2 h_{fg})^{1/2}$	(3)
	$h = 0.0077\ F_2(N_{Re})^{0.4}(1/\mu_f)^{1/3}$	(4)
2. Outside Horizontal Tubes, N rows in a vertical plane, Length L, Laminar Flow	$h = 0.79\ F_1(h_{fg}/N\mathrm{d}\Delta t)^{1/4}$	(5)
	$h = 1.05\ F_2(L/w_l)^{1/3}$	(6)
Finned Tubes	$h = 0.689\ F_1(h_{fg}/\Delta t D_{eq})^{1/4}$ Where D_{eq} is determined from	(7)
	$\dfrac{1}{(D_{eq})^{1/4}} = 1.30\ \dfrac{A_s\phi}{A_{ef}(L_{mf})^{1/4}} + \dfrac{A_p}{A_{ef}(\mathrm{d})^{1/4}}$	(7a)
3. Simplified Equations for Steam; Outside Vertical Tubes, $N_{Re} = 4\Gamma/\mu_f < 2100$	$h = 4000/(L)^{1/4}(\Delta t)^{1/3}$	(8)
Outside Horizontal Tubes, $N_{Re} = 4\Gamma/\mu_f < 1800$ Single Tube	$h = 3100/(d')^{1/4}(\Delta t)^{1/3}$	(9)
Multiple Tubes	$h = 5800/(Nd')^{1/4}(\Delta t)^{1/3}$	(9a)
4. Inside Vertical Tubes	$h = 0.065\left(\dfrac{c_{pf}k_f\rho_f f}{2\mu_f\rho_v}\right)^{1/2} G_m$	(10)
	Where $G_m = \left(\dfrac{G_i{}^2 + G_iG_o + G_o{}^2}{3}\right)^{1/2}$	(10a)
5. Inside Horizontal Tubes, $\dfrac{\mathrm{d}G_l}{\mu_l} < 5{,}000$		
$1{,}000 < \dfrac{\mathrm{d}G_v}{\mu_l}\left(\dfrac{\rho_l}{\rho_v}\right)^{1/2} < 20{,}000$	$\dfrac{h\mathrm{d}}{k_l} = 13.8\left(\dfrac{c_p\mu_l}{k_l}\right)^{1/3}\left(\dfrac{h_{fg}}{c_p\Delta t}\right)^{1/6}\left[\dfrac{\mathrm{d}G_v}{\mu_l}\left(\dfrac{\rho_l}{\rho_v}\right)^{1/2}\right]^{0.2}$	(11)
$20{,}000 < \dfrac{\mathrm{d}G_v}{\mu_l}\left(\dfrac{\rho_l}{\rho_v}\right)^{1/2} < 100{,}000$	$\dfrac{h\mathrm{d}}{k_l} = 0.1\left(\dfrac{c_p\mu_l}{k_l}\right)^{1/3}\left(\dfrac{h_{fg}}{c_p\Delta t}\right)^{1/6}\left[\dfrac{\mathrm{d}G_v}{\mu_l}\left(\dfrac{\rho_l}{\rho_v}\right)^{1/2}\right]^{2/3}$	(12)
For $\dfrac{\mathrm{d}G_l}{\mu_l} > 5{,}000\ \dfrac{\mathrm{d}G_v}{\mu_l}\left(\dfrac{\rho_l}{\rho_v}\right)^{1/2} > 20{,}000$	$\dfrac{h\mathrm{d}}{k_l} = 0.026\left(\dfrac{c_p\mu_l}{k_l}\right)^{1/3}\left(\dfrac{\mathrm{d}G_E}{\mu_l}\right)^{0.8}$	(13)
	Where $G_E = G_v\left(\dfrac{\rho_l}{\rho_v}\right)^{1/2} + G_l$	(13a)

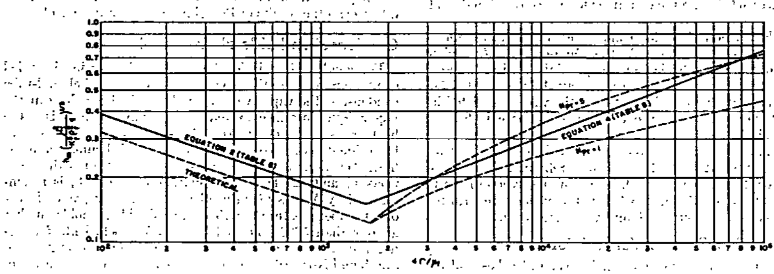


Fig. 13 . . . . Film-Type Condensation

Table 9 . . . . Values of Condensing Coefficient Factors for Different Refrigerants*

Refrigerant	Film Temp., Fahr. $t_f = t_{sat} - 0.75\ (\Delta t)$	F_1	F_2
Refrigerant 11	75	154	822
	100	153	815
	125	151	803
Refrigerant 12	75	133	672
	100	122	608
	125	112	538
Refrigerant 22	75	153	822
	100	144	755
	125	132	675
Sulfur Dioxide	75	290	1920
	100	299	2000
	125	318	2170
Ammonia	75	409	3040
	100	408	3035
	125	408	3030
Propane	75	159	850
	100	157	845
	125	154	836
Butane	75	156	840
	100	156	843
	125	157	845

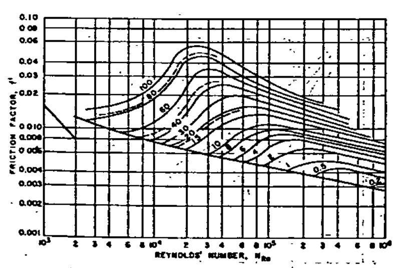
* Property values from Chapter 19.

$$F_1 = \left(\frac{k_f{}^3\rho_f{}^2 g}{\mu_f}\right)^{1/4} \qquad \text{Units} = \left(\frac{(\mathrm{Btu})^3(\mathrm{lb_m})}{(\mathrm{hr})^4(\mathrm{ft})^7(\mathrm{F})^3}\right)^{1/4}$$

$$F_2 = \left(\frac{k_f{}^3\rho_f{}^2 g}{\mu_f}\right)^{1/3} \qquad \text{Units} = \left(\frac{(\mathrm{Btu})^4(\mathrm{lb_m})}{(\mathrm{hr})^4(\mathrm{ft})^7(\mathrm{F})^3}\right)^{1/3}$$

Impurities and Noncondensable Gases

The vapor entering the condenser often contains a small percentage of impurities such as oil. The oil forms a film on the condensing surfaces which offers additional resistance to the heat transfer. Some allowance should be made for this, especially when no oil separator is used and when the discharge lines from the compressor to the condenser are short. Experiments show that the presence of noncondensable gases in the condenser adversely affects the heat transfer. The decrease in the heat transfer coefficient is reported by Chapman[70] to be approximately linear with the weight fraction of the noncondensable gas present.



Parameter: Γ/ρs, where Γ = liquid flow rate, pounds per (hour) (foot); ρ = liquid density, pounds per cubic foot; and s = surface tension of liquid relative to water. The values of gas velocity utilized in calculating f' and N_{Re} are calculated as though no liquid was present.

Fig. 14 . . . . Friction Factors for Gas Flow Inside Pipes with Wetted Walls[67]

Pressure Drop

For a general discussion of pressure drop with two-phase flow, refer to Chapter 6, Fluid Flow. Pressure drop data for two-phase flow in a horizontal tube with condensing Refrigerant 22 are reported by Altman, Staub and Norris.[71]

EXTENDED SURFACE

Heat transfer from a prime surface can be increased by attaching *fins* or *extended surfaces*, which increase the area available for heat transfer. In general, the advantage of fins is that they provide a more compact heat exchanger with lower material costs for a given performance. In order to achieve an optimum design, fins are generally provided on the side of the heat exchanger where the heat transfer coefficients are low (such as the air side of an air-to-water coil). Examples of equipment employing extended surface are natural and forced convection coils and shell-and-tube evaporators and condensers. Fins are also used inside tubes in condensers and dry expansion evaporators.

Fin Efficiency

As heat flows from the root of a fin to its tip, there is a temperature drop due to the thermal resistance of the fin material. Therefore, the temperature difference between the fin and the surrounding fluid varies, being greater at the root than at the tip, and causes a corresponding variation in the heat flux. For this reason, increases in the length of a fin result in proportionately less additional heat transfer. To account for this effect, a factor is introduced called the *fin efficiency*. The fin efficiency is defined by Equation 18 as the ratio of the actual heat transferred from the fin to the heat which would be transferred if the entire fin were at its root or base temperature.

$$\phi = \frac{\int h(t - t_e)dA}{\int h(t_r - t_e)dA} \qquad (18)$$

where ϕ is the fin efficiency, t_e is the temperature of the surrounding environment, and t_r is the temperature at the fin root. The fin efficiency will be low for long fins, thin fins, or fins of low thermal conductivity material. Furthermore, the efficiency will decrease as the heat transfer coefficient increases because of the increased heat flow through the fin. For natural convection in air-cooled condensers and evaporators, where h for the air side is low, fins can be made fairly large and of low conductivity materials, such as steel, instead of copper or aluminum. On the other hand, for condensing and boiling, where large heat transfer coefficients are involved, fins must be very short to achieve optimum usage of material.

The heat transfer from a finned surface, such as a tube, which includes both finned or secondary area, A_s, and unfinned or prime area, A_p, is given by Equation 19.

$$q = (h_p A_p + \phi h_s A_s)(t_r - t_e) \qquad (19)$$