

Boiling film coefficients in evaporators when the refrigerant flows inside plain or finned tubes are influenced by:

1. Tube size (coefficient increases with a decrease in tube diameter).
2. Method of expansion (direct or recirculated).
3. Refrigerant distribution.
4. Presence of oil.
5. Liquid subcooling prior to expansion.
6. Superheat of leaving refrigerant.
7. Evaporator temperature.
8. Refrigerant pressure drop.

Attempts to correlate the film coefficient with the mass velocity of the refrigerant have been generally unsatisfactory. Whether the refrigerant is evaporated on the inside or outside of tubes, the film coefficients increase as the loading (or heat flux) is increased. An approximation of this effect is:

$$h_f = C_1 + C_2 \left(\frac{q}{A} \right) \quad (4)$$

where C_1 and C_2 are constants.

This above straight line relationship appears to be reasonably accurate for halocarbon refrigerants within the range $1000 < q/A < 5000$, in Btu/h/(sq ft).

The nature of the tube surface has a pronounced effect upon the temperature difference required to promote boiling or, conversely, the amount of heat that can be transferred per degree of temperature difference across the refrigerant film. A roughened surface provides more points on which nucleate boiling can form and promotes the formation of smaller gas bubbles which are readily swept away to be replaced by liquid refrigerant.

Refrigerant velocities within the tubes of dry expansion coolers must be kept above certain minimum values to obtain good refrigerant distribution and usually to obtain satisfactory oil return. A minimum pressure drop for good performance is the result.

It is common practice to rate the performance of an evaporator in terms of the saturated refrigerant temperature based upon the exit refrigerant pressure. Pressure drop within an evaporator results in an average refrigerant temperature higher than that of the rating basis. Within the pressure drop ranges given previously, a decrease in the true value of the ΔT_m due to an increase in pressure drop is usually offset by an improved film coefficient or improved refrigerant distribution.

The overall boiling refrigerant film coefficient h , varies in an inverse manner with superheat. A change in superheat from 8 F deg to 14 F deg in a dry expansion cooler can decrease the value of h , by as much as 12 percent.

Dimensionless correlations have been developed¹⁻⁴ whereby the boiling film coefficients can be mathematically determined when the physical properties of the fluid and the surface characteristics are known. Despite this work, experimental determination of the boiling film coefficients for each refrigerant in each style of evaporator and with each method of refrigerant flow control is the safest method.

Chilled Liquid Film Coefficient

The Reynolds number and other dimensionless groups involving physical constants of the liquid and the surface characteristics can often be successfully applied in predicting liquid film coefficients.^{4,10,11,12}

For turbulent flow of liquids or gases inside of straight clean round tubes, Equations 3 or 5 of Table 6 in Chapter 4, Heat Transfer are generally used as the basic equation for the liquid film coefficient.

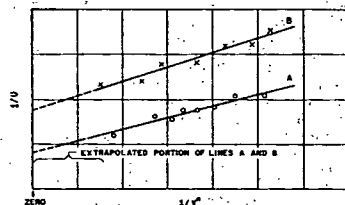


Fig. 7.... Typical Wilson Plot

The term $(\mu/\mu_s)^{0.4}$ in Equation 5 of Table 6 in Chapter 4 can be considered as unity for most applications using water or brine.

For liquid water at temperature up to 220 F, Equation 14 of Table 6 in Chapter 4 is used.

For turbulent flow of liquids or gases across rows of tubes, Equations 25 and 26 of Table 6, Chapter 4, are generally used to determine the film coefficient.

The method formerly used to predict values of the fluid film coefficient h_f when the fluid is on the outside of the tubes was to predict values for cross-flow and apply a factor to account for bypass and leakage around the baffles and tube bundle.¹³

Investigations^{1,2,3,4,14} have produced a method for accurately determining the fluid film coefficient h_f when the fluid is on the outside of the tube surface.

Low-tonnage evaporators frequently employ the multi-tube, counter-flow principle. The fluid being cooled flows on the outside of the tube and parallel to the axis of the tube or tubes. The flow is considered as similar to flow through tubes. It is customary to consider d to be four times the hydraulic radius based upon the total wetted perimeter when pressure drop calculations are concerned, and d to be four times the hydraulic radius based upon the portion of the wetted perimeter through which heat flows for calculations involving the value of h_o . Film coefficients for flow of fluids either inside or outside tubes are greater with turbulent flow, low viscosity, high thermal conductivity, small or closely packed tubes, and minimum bypass and leakage in transverse-baffle heat exchangers.

Separation of Coefficients—Analysis of Test Results

One of the most useful analytical plots at the disposal of the engineer is the Wilson Plot. An outline of a Wilson Plot is shown in Fig. 7. The reciprocal of U is plotted against the reciprocal of the velocity or of the fluid flow rate of the fluid being cooled. The value of the exponent n depends upon whether the fluid being cooled is inside tubes, flowing parallel to the axis of the tube, or is flowing across tubes by means of transverse baffles. For flow through tubes and parallel to the axis of tubes $n = 0.8$. Various investigators have reported values of n from 0.55 to 0.65 for flow of fluids across tubes in transverse baffled heat exchangers. A reasonably safe value is $n = 0.6$.

Lines A and B represent constantly held conditions of operation other than the velocity of the fluid being cooled. For example line B might represent the operation of the evaporator at half of the load represented by line A. Line B might also represent the performance of a brine cooler whose brine viscosity is greater than that of the brine represented

Evaporators for Liquid Cooling

Table 4.... Recommended Fouling Factors for Use in Calculating Heat Transfer Rates*

Application	Non-Ferrous Tubes	Ferrous Tubes
Brine Coolers (Brine Side)		
Inhibited salt brines	0.0005	0.001
Non-inhibited salt brines	0.001	0.002
Solvent brines (methylene chloride, fluorinated hydrocarbon refrigerants)	None	None
Water Coolers (water side)		
Recirculated water, closed systems	0.0005	0.001
Recirculated water, open systems	0.001	0.002
Non-recirculated water	Use 1/4 of that allowed for condenser service but in no case less than 0.0005.	

* Fouling factors are in (F deg) (hr) (sq ft) per Btu.

by line A and which is at a lower evaporator temperature than that used to determine line A. The scale $1/w$ must start at zero to make the plot of value. When $1/w$ equals zero or infinite velocity the resistance on the fluid side theoretically equals zero. The intercepts of lines A and B on the $1/U$ scale, therefore, represent the sum of all other resistances. The fluid side coefficients can be determined from the slope of lines A and B.

Fouling Factors

Resistance due to fouling depends upon the material of which the tube is made, nature of the fluid, and fluid velocity. Copper tubes usually become fouled at a slower rate and to a lesser degree than steel tubes. Fluid velocities less than approximately 3 fps promote additional fouling by permitting static accumulation of deposits. Recommended fluid velocities for copper and steel tubes are from 5 to 10 fps providing pressure drop limitations are not exceeded. Table 4 lists recommended fouling resistances. The values are expressed as $1/h_{fs}$.

Table 5.... Overall Heat Transfer Coefficients for Liquid Coolers

Type of Evaporator	Minimum	Maximum
Flooded shell-and-plain-tube (water to Refrigerants 12, 22, and 717)	130	190
Flooded shell-and-plain-tube (water to Refrigerant 12 or 22)	90	140
Flooded shell-and-plain-tube (brine to Refrigerant 717)	45	100
Flooded shell-and-plain-tube (brine to Refrigerant 12 or 22)	30	90
Dry expansion, shell-and-plain-tube (water to Refrigerant 12, 22, or 717) (Refrigerant in tubes)	80	160
Dry expansion, shell-and-internal-finned-tubes (water to Refrigerant 12 or 22) (Refrigerant in tubes)	160	250
Dry expansion, shell-and-plain-tube (brine to Refrigerant 12, 22, or 717) (Refrigerant in tubes)	60	140
Dry expansion, shell-and-internal-finned-tubes (non-salt brines to Refrigerant 12, or 22)	100	170
Shell-and-plain-tube coil (water in shell) (Refrigerant 12, 22, or 717 in coil)	10	25
Baudelot cooler, flooded (Refrigerant 12 or 22 to water)	100	200
Baudelot cooler, dry expansion (Refrigerant 717 to water)	60	150
Baudelot cooler, dry expansion (Refrigerant 12 or 22 to water)	60	120
Double-pipe cooler (Refrigerant 717 to water)	50	150
Double-pipe cooler (Refrigerant 717 to brine)	50	125
Spray type shell-and-tube water coolers (Refrigerants 11, 12, 22, 113, or 717)	150	250
Tank-and-agitator, coil type water cooler (flooded, Refrigerant 717)	80	125
Tank-and-agitator, coil type water cooler (flooded, Refrigerant 12 or 22)	60	100
Tank, ammonia (Refrigerant 717) to brine cooling, coils between can in ice tank	15	40
Tank, high velocity raceway type (Refrigerant 717 to brine)	80	110

Water and brine coolers operating at normal conditions do not require the addition of a fouling resistance to the refrigerant side. Caution should be exercised when designing evaporators for service below -20 F because the oil that is circulating through the refrigerant circuit may congeal on the tube surface and add an appreciable resistance to heat flow. This is particularly true of the dry expansion type cooler.

Comprehensive tables of fouling resistances are given in Reference 1.

Overall Heat Transfer Rates.

Due to the many variables that enter into the determination of overall heat transfer rates, Table 5 gives only approximate minimum and maximum values of U .

The higher values of U generally apply to loadings, (Q/A) , corresponding to a (ΔT_m) of 12-F deg or more, at the higher velocities. The lower values of U apply to loadings corresponding to (ΔT_m) of 8 F deg and less, and at low fluid velocity.

Water and Brine Properties

The physical properties of a fluid required in order to calculate associated heat transfer film coefficients h and pressure drop are: (1) density or specific gravity, (2) specific heat, (3) thermal conductivity, and (4) viscosity.

The properties of water are given in Reference 12. The properties of calcium and sodium chloride brine are given in Chapter 21. Some properties of ethylene and propylene brine are given in Chapter 21. Additional information may be obtained from Reference 7 of Chapter 21. Properties of the alcohol brines are given in Reference 17.

REFERENCES

1. Standards of Tubular Exchanger Manufacturers (Tubular Exchanger Manufacturers Association, New York, 1952, 3rd ed.).
2. Joseph Fabregas: Design heat exchangers more exactly (Chemical Engineering, April 1956, p. 181).
3. Townsend Tinker: Shell side characteristics of shell and tube heat exchangers (ASME Paper No. 56-A-123, 1956).
4. O. P. Bergelin, K. J. Bell, and M. D. Leighton: Heat Transfer and Fluid Friction During Flow Across Banks of Tubes (University of Delaware Engineering Experiment Station Bulletin No. 4, 1952, Revised April 1958).