Design Techniques for R-717, R-22, and R-12 Shell-and-Coil Subcoolers and Intercoolers

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ABSTRACT

Subcoolers and Intercoolers are becoming essential part of any refrigeration system. The most common type found in industrial refrigeration is the Shell-and-Coil configuration. This paper presents a design technique for sizing of coils for Ammonia, R-22, and R-12 systems. The evaluation of inside and outside heat transfer coefficients is based on well established correlations and computer programs. Correlations are also presented for pressure drop evaluation. The derived correlations are valid for temperature range 0 - 120 °F.

TECHNIQUES DE CONCEPTION DES SOUS-REFROIDISSEURS ET REFROIDISSEURS INTERMEDIAIRES MULTITUBULAIRES POUR R717, R22 et R12.

Résumé : Les sous-refroidisseurs et les refroidisseurs intermédiaires sont devenus une partie essentielle de tout système frigorifique. Le type le plus courant dans le froid industriel est le type multitubulaire. Cette communication présente une technique de conception pour le dimensionnement des serpentins des systèmes à ammoniac, R22 et R12. L'évaluation des coefficients de transfert de chaleur interne et externe se fonde sur des corrélations bien établies et des programmes d'ordinateur. On présente aussi des corrélations pour l'évaluation de la chute de pression. Les corrélations établies sont valables pour des températures de - 18 à 50°C.

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NOMENCLATURE

Ai = inside surface area, ft^2
Am = log mean area, ft^2
Ao = outside surface area, ft^2
Ch = heat transfer factor, $Btu/hr^{.2}-ft^{2.6}-°F$
COP = coefficient of performance
Cp = pressure drop factor, $lb-s^{1.8}/in^{2.6}$
Di = inside diameter, in.
Do = outside diameter, in.
f = Darcy's friction factor
Fh = constant for heat transfer, $Btu/hr^{.2}-ft^{2.6}-°F$
Fp = constant for pressure drop, $lb/hr^{.2}-ft^{2.6}$
g = gravitational acceleration, ft/s^2
GTD = greatest temperature difference, °F
h = enthalpy, Btu/lb
hi = inside heat transfer coefficient, $Btu/hr-ft^2-°F$
ho = outside heat transfer coefficient, $Btu/hr-ft^2-°F$
k = refrigerant thermal conductivity, Btu/hr-ft-°F
K = coil thermal conductivity, Btu/hr-ft-°F
L = coil straight length, ft
LMTD = log mean temperature difference, °F
LTD = least temperature difference, °F
p = pressure drop, psi
Pr = Prandtl number
q_{in} = heat input, Btu/hr
R = wall thermal resistance, $hr-ft^2-°F/Btu$
Re = Reynold's number
Rffi = inside fouling factor, $hr-ft^2-°F/Btu$
Rffo = outside fouling factor, $hr-ft^2-°F/Btu$
Rw = net wall resistance, $hr-ft^2-°F/Btu$
s = entropy, Btu/lb-°F
T = temperature, °F
Ts = saturation temperature, °F
T = temperature difference (range), °F
Uo = overall heat transfer coefficient, $Btu/hr-ft^2-°F$
V = velocity, ft/s
w_{net} = mechanical work, Btu/hr
ρ = refrigerant density, lb/ft^3
μ = bulk viscosity, lb/ft-hr
μ_{avg} = average viscosity for 0-120 °F range, lb/ft-hr
μ_o = viscosity at 0 °F, lb/ft-hr
μ_w = viscosity at wall temperature, lb/ft-hr

INTRODUCTION

In any refrigeration system there are various pressure vessels, heat exchangers, and other auxiliary components that are interlinked via complex network of piping. The basic aim of this conglomeration is to transfer heat from one section of the system to another at a most economical cost.

It is important to note that the final system cost should be evaluated on the basis of the equipment cost, the installation and maintenance cost minus the energy savings, i.e., C - ES. The term, ES, will be zero for an ordinary design, but will be greater than zero for an energy intensive modified system. Any modification will of course result in a rise in 'C', so that the actual indicator of net benefit is Z = C/(C - ES).

There are numerous ways to increase this factor in a refrigeration system. The most common method is to feed a subcooled refrigerant to the expansion valve. One way to attain subcooled refrigerant is to over design the condenser so that besides condensing, the refrigerant is also subcooled. The problem with this method is that the condenser cannot achieve high level of subcooling. The tower water (in case of water cooled condensers) is usually available at 70°F to 80°F with a temperature range of about 10 degrees. The refrigerant condenses at 95°F to 105°F. Hence, there is a limit on the approach temperature. The close approach temperature results in smaller LMTD which in turn results in larger surface area. Large surface area means larger shell and more tubes, which means extra material and labor cost. It is important to note that drilling holes in a tubesheet is a major component in determining the final cost of a heat exchanger.

SHELL-AND-COIL SUBCOOLER

The most common and widely used method is to install a subcooler before an expansion device as shown in figure 1a. A subcooler could either be a shell-and-tube type or a shell-and-coil configuration. The latter type is common in the industry because of its simplicity and lower cost. A typical configuration is shown in figure 2. The coil is submerged in a saturated pool of refrigerant at a saturation temperature corresponding to the low side pressure.

To examine the net effect of a subcooler it is important to study the T-s chart as shown in figure 1b. The refrigeration effect is defined as:

$$q_{in} = \Delta h\big|_{T_s} \qquad (1)$$

A system with no subcooling follows the broken line path between Point 2 and Point 4'. The refrigeration effect is:

$$q_{in} = h_5 - h_{4'} \qquad (2)$$

On the other hand a system with a subcooler follows the solid line path between Point 2 and Point 4. The refrigeration effect for this case is:

$$q_{in} = h_5 - h_4 \qquad (3)$$

Hence, the refrigeration effect is higher with the subcooler. The magnitude of the enhancement depends upon the amount of subcooling achieved.

DESIGNING A SHELL-AND-COIL SUBCOOLER

Designing a correct coil could be cumbersome, involving single and two-phase flow analysis. The essence of this paper is to formulate a simple method that would enable a designer who has the basic knowledge of heat transfer and fluid flow to size an appropriate coil.

The equations and curves were developed on the basis of well known empirical correlation, a feedback from the previous jobs, and the computer program developed for flooded evaporators.

The design technique for three most common and widely used refrigerants, R-717 (ammonia), R-22, and R-12 is presented. The range of temperature common in practice is between 0 - 120°F, therefore, the analysis is valid for this range. The transport properties for these refrigerants are shown in Table 1.

The overall resistance to heat transfer based on the outside surface of the coil is given as,

$$1/U_oA_o = 1/h_oA_o + 1/h_iA_i + R_{ffo}/A_o + R_{ffi}/A_i + R_w \qquad (4)$$

$$1/U_o = 1/h_o + 1/h_i\,(A_o/A_i) + R_{ffo} + R_{ffi}\,(A_o/A_i) + R\,(A_o/A_m)$$

$$1/U_o = 1/h_o + 1/h_i\,(D_o/D_i) + R_{ffo} + R_{ffi}\,(D_o/D_i) + (D_o/24K)\ln(D_o/D_i) \qquad (5)$$

where, $A_m = (D_o-D_i)/\ln(D_o/D_i)$; $R = (D_o-D_i)/24K$. The fouling resistances are a matter of choice. They depend upon the level of impurities in the system. They could also be recommended by the specifier. A common value for each side is 0.0005 hr-ft^2°F/Btu. To facilitate calculations, the last term in equation

2 was evaluated for different low carbon steel pipe sizes. The tabulated values are given in Table 2.

In order to evaluate the second term, $1/hi$, a Sieder-Tate [1] correlation for turbulent in-tube flow is applied, i.e.,

$$hi = 0.027\ k/Di\ Re^{0.8}\ Pr^{1/3}\ (\mu/\mu w)^{0.14} \qquad (6)$$

Substituting the transport properties from Table 1 and rewriting equation 6 in different format results in,

$$hi = 0.027\ (V^{0.8}/Di^{.2})\ Fh\ (\mu/\mu w)^{0.14} \qquad (7)$$

where, $Fh = (k^2 cp\, \rho^{2.4}/\mu^{1.4})^{1/3}$, $(Btu/hr^{.2}\text{-}ft^{2.6}\text{-}°F)$

Table 1 indicates that the variation in quantities of the term, Fh, is not large for the range 0 - 120 °F. Hence, it is acceptable to substitute an average value for this term. The term, $\mu/\mu w$, is replaced by, $\mu avg/\mu o$, which is a safe assumption. The final equation for hi, is,

$$hi = Ch\ V^{.8}/Di^{.2}, \quad (Btu/hr\text{-}ft^2\text{-}°F) \qquad (8)$$

where, Ch is:
388.7 for R-717
123.1 for R-22
98.6 for R-12

Equation 8 shows that in order to evaluate hi, the only parameters a designer has to know are the coil size and the refrigerant velocity.

To evaluate the first term, ho, the outside heat transfer coefficient, consult the curves in Figures 3 - 5 for the appropriate refrigerant. Ordinarily this would be a challenging part of the design as this coefficient depends on various physical and flow parameters [2].

Once the length of the coil is known, it is important to evaluate the pressure drop. In case the pressure drop is high, the designer has to either select a larger size coil or select multiple circuit coil connected via a common header. The equation used for the pressure drop is,

$$\Delta p = f(L/Di)\ (\rho V^2/2g)\ (\mu w/\mu)^{.14}\ (1/144), \quad (psi) \qquad (9)$$

where, $f = 0.2/Re^{0.2}$ [3]

or, $$\Delta p = 4.193*10^{-6}\ Fp\ (LV^{1.8}/Di^{1.2}) \qquad (10)$$

where, $Fp = \mu^{.6}\ \rho^{.8}\ \mu w^{.14}$, $(lb/hr^{.2}\text{-}ft^{2.6})$

The factor Fp is evaluated using properties in Table 1 with μw taken for the worst condition, i.e., at 0 °F.

or $$\Delta p = Cp\ (LV^{1.8}/Di^{1.2}), \quad (psi) \qquad (11)$$

where Cp is:
$6.762*10^{-5}$ for R-717
$1.220*10^{-4}$ for R-22
$1.349*10^{-4}$ for R-12

CONCLUSION

A simplified design procedure is presented for sizing subcooler and intercooler coils with R-717 (ammonia), R-22, and R-12 for a temperature range of 0 - 120 °F. This procedure will prove to be a helpful tool for practicing engineers that do not want to be intensively involved in the complexities of two-phase flow and boiling heat transfer.

REFERENCES

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[3] Fraas, A. P., and Ozisik, M. N., "Heat Exchanger Design," John Wiley and Sons, New York (1965).

[4] ASHRAE Handbook, "1981 Fundamentals," Atlanta (1982).

Table 1. Factors Fh, Fp. Properties taken from [4]

T	μ	ρ	k	cp	Fh	Fp
			R-717			
0	0.558	41.34	0.335	1.083	12.7714	17.4752
20	0.494	40.43	0.321	1.092	12.9430	17.0418
40	0.437	39.49	0.306	1.103	13.0709	16.6015
60	0.386	38.50	0.291	1.118	13.1837	16.1470
80	0.341	37.48	0.276	1.135	13.2645	15.6868
100	0.301	36.40	0.261	1.158	13.3213	15.2098
120	0.268	35.25	0.246	1.187	13.2853	14.7212
			R-22			
0	0.654	83.825	0.0630	0.271	4.3183	31.7552
20	0.599	81.597	0.0598	0.276	4.2787	30.9148
40	0.553	79.255	0.0566	0.283	4.2178	30.0584
60	0.513	76.773	0.0534	0.291	4.1346	29.1712
80	0.480	74.116	0.0502	0.300	4.0197	28.2478
100	0.449	71.236	0.0471	0.313	3.9052	27.2568
120	0.427	68.054	0.0439	0.332	3.7506	26.1993
			R-12			
0	0.767	90.659	0.0490	0.217	3.3520	34.9050
20	0.687	88.529	0.0467	0.220	3.3685	34.0218
40	0.620	86.296	0.0443	0.224	3.3628	33.1289
60	0.564	83.944	0.0420	0.229	3.3423	32.2211
80	0.517	81.450	0.0397	0.234	3.2962	31.2891
100	0.477	78.785	0.0373	0.240	3.2240	30.3206
120	0.441	75.906	0.0350	0.251	3.1581	29.2926

Table 2. Pipe size [3] and wall thermal resistance for carbon steel

Nom. pipe size (in)	Do (in)	Di (in)	Sch. #	Do/Di	Rw hr-ft²-F/Btu
3/4	1.050	0.824	40	1.2743	0.00038
		0.742	80	1.4151	0.00055
1	1.315	1.049	40	1.2536	0.00045
		0.957	80	1.3741	0.00063
1-1/4	1.660	1.380	40	1.2029	0.00046
		1.278	80	1.2989	0.00065
1-1/2	1.900	1.610	40	1.1801	0.00047
		1.500	80	1.2667	0.00067
2	2.375	2.067	40	1.1490	0.00049
		1.939	80	1.2249	0.00072
2-1/2	2.875	2.469	40	1.1644	0.00066
		2.323	80	1.2376	0.00092
3	3.500	3.068	40	1.1408	0.00069
		2.900	80	1.2069	0.00098
3-1/2	4.000	3.548	40	1.1274	0.00072
		3.364	80	1.1891	0.00103
4	4.500	4.026	40	1.1177	0.00075
		3.826	80	1.1762	0.00109

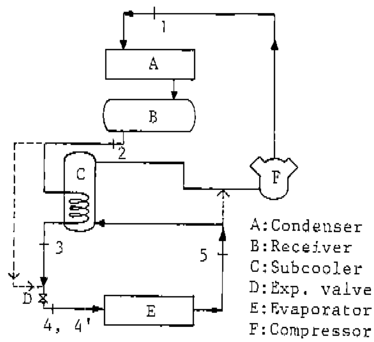

Fig. 1a Vapor compression loop with and without a subcooler

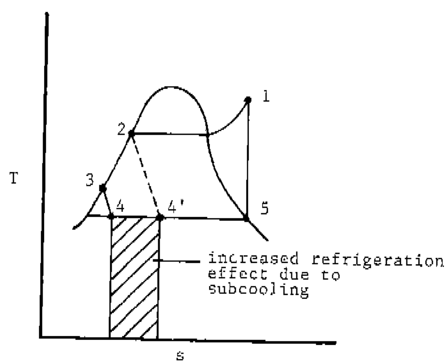

Fig. 1b T-s diagram for refrigeration cycle in Figure 1a

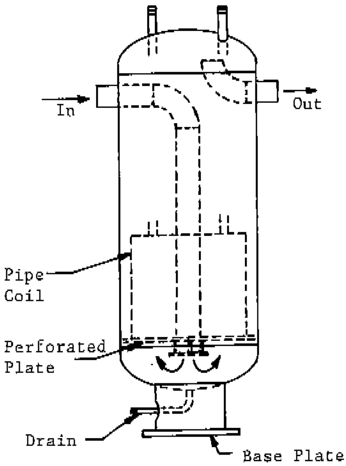

Fig. 2 Typical coil subcooler

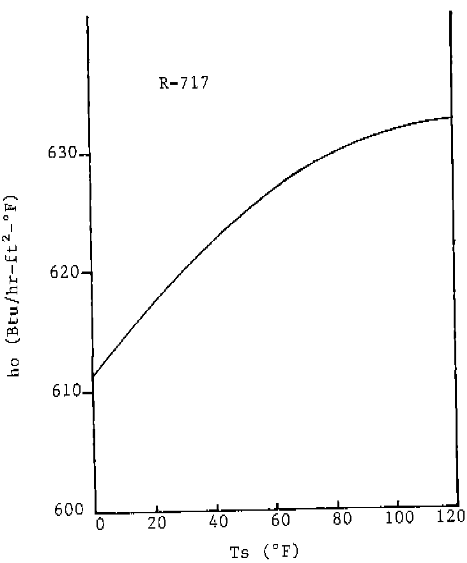

Fig. 3 Outside heat transfer coefficient for Ammonia

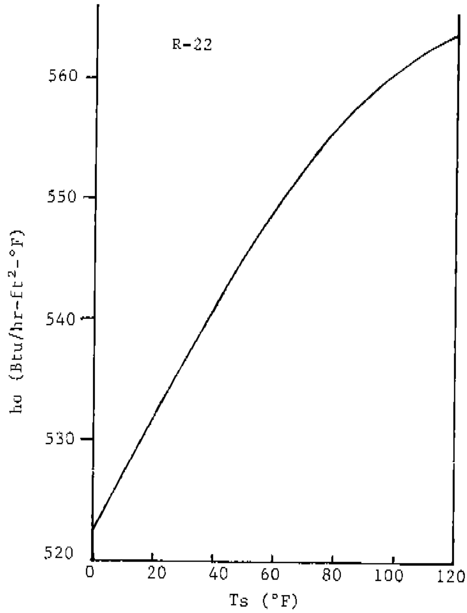

Fig. 4 Outside heat transfer coefficient for Freon 22

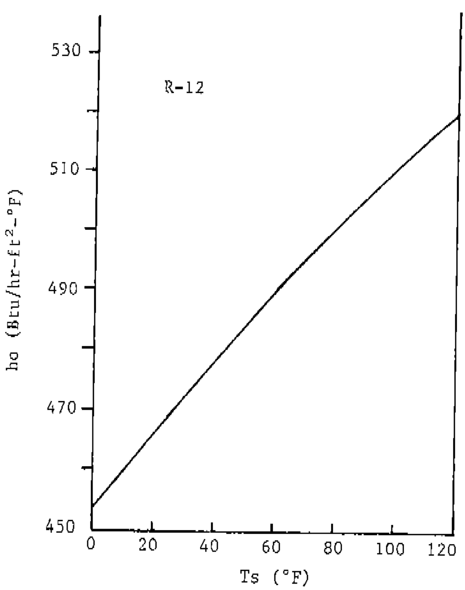

Fig. 5 Outside heat transfer coefficient for Freon 12