THE EFFECT OF CONVECTIVE HEAT TRANSFER COEFFICIENTS ON THE DESIGN OF CONDENSER AND EVAPORATOR OF A REFRIGERATION SYSTEM

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Abstract

In this paper, some possibilities to increase the overall heat transfer coefficients for condenser and evaporator of a refrigeration system are discussed. Some important heat transfer equations have been compiled and are presented for ready reference. Condensing and boiling heat transfer coefficients for R-12, R-22 and R-717 have been calculated for a 1.0 TR cooling capacity system operating on a given refrigeration cycle, and the calculated results are discussed with reference to designs of condenser and evaporator for given heat transfer duties. Some sample calculations of heat transfer coefficients have been included in the Appendix.

Introduction

In a vapour-compression refrigeration system, condenser and evaporator are among the four major components; the other two are compressor and expansion device. The most important item in designs of condenser and evaporator is the rate of heat to be transferred by each. The sizes of these heat exchangers should be minimized for a given heat transfer rate in order to reduce costs and to save space of installation. Calculation methods of the heat transfer characteristics of actual condenser and evaporator are very complex and often impossible. It is, however, possible to analyse the condenser and the evaporator, making necessary assumptions, and the predicted results are useful. Accordingly heat transfer coefficients for R-12, R-22 and R-717 have been calculated and suggestions are given for increasing their overall heat transfer coefficients. These suggestions are expected to be helpful in the designs of condensers and evaporators.

Overall heat transfer coefficients U_c and U_e

Heat transfer rates Q_c and Q_e are directly proportional to U_c and U_e respectively. Calculations of U_c and U_e require calculations of refrigerant side coefficients h_{ic} and h_{ie}, and the air – or water-side coefficients h_{oc} and h_{oe}. The coefficients h_{ic} and h_{ie} cannot be determined accurately primarily because of the fact that the dryness fraction of the refrigerant is not constant along the entire length of the tube, resulting in varying flow velocities inside the tubes. However, working formulae for estimation of the refrigerant side coefficients with acceptable accuracy have been suggested by various investigators. Such formulae may be found in ref (3), and some are listed below for ready reference.

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For condensation inside horizontal tubes, the following equation are recommended for h_{ic}.

$$\frac{h_{ic}\, d}{k_l} = 13.8 \left(\frac{c_{pl}\, \mu_l}{k_l}\right)^{\frac{1}{3}} \left(\frac{L}{c_{pl}\, \Delta t}\right)^{\frac{1}{6}} \left\{\frac{G_v\, d}{\mu_l}\left(\frac{\rho_l}{\rho_v}\right)^{\frac{1}{2}}\right\}^{\frac{1}{5}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

when $\frac{G_l\, d}{\mu_l} < 5{,}000$, and $1{,}000 < \frac{G_v\, d}{\mu_l}\left(\frac{\rho_l}{\rho_v}\right)^{\frac{1}{2}} < 20{,}000$

$$\frac{h_{ic}\, d}{k_l} = 0.1 \left(\frac{c_{pl}\, \mu_l}{k_l}\right)^{\frac{1}{3}} \left(\frac{L}{c_{pl}\, \Delta t}\right)^{\frac{1}{6}} \left\{\frac{G_v\, d}{\mu_l}\left(\frac{\rho_l}{\rho_v}\right)^{\frac{1}{2}}\right\}^{\frac{2}{3}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

when $20.000 < \frac{G_v\, d}{\mu_l}\left(\frac{\rho_l}{\rho_v}\right)^{\frac{1}{2}} < 100{,}000$

$$\frac{h_{ic} d}{k_l} = 0.026 \left(\frac{c_{pl}\, \mu_l}{k_l}\right)^{\frac{1}{3}} \left\{\left[G_v \left(\frac{\rho_l}{\rho_v}\right)^{\frac{1}{2}} + G_l\right]^{\frac{4}{5}} \frac{d}{\mu_l}\right\} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3)$$

when $\frac{G_v\, d}{\mu_l} > 5{,}000$, and $< \frac{G_v\, d}{\mu_l}\left(\frac{\rho_l}{\rho_v}\right)^{\frac{1}{2}} > 20{,}000$

Equation (1), (2) and (3) are applicable to condensation of saturated vapour to saturated liquid. However, little error is introduced when applied to superheated vapour if we assume that the wall temperature is the same as the saturation temperature within the superheat part of the tubes.

For evaporation of R-12 and R-22 inside horizontal copper tubes, the following equation may be used (ref 3),

$$h_{ie} = C_1 \left(\left(\frac{k_l}{d}\right)\left[\left(\frac{G d}{\mu_l}\right)^2 \left(\frac{J\, \Delta x\, L\, g_c}{l\, x\, g}\right)\right]^n \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(4)$$

Where, C_1 = constant = 9×10^{-4} when $x_e \le 0.90$, and
= 8.20×10^{-3} when $x_e > 0.90$ (x_e is the dryness fraction of the refrigerant leaving the tube).

n = constant = 0.50 when $x_e \le 0.90$, and
= 0.40 when $x_e > 0.90$.

and $g_c = 1 \frac{kg\, m}{N\, s^2}$.

The outside (water or air) film heat transfer coefficients h_{oc} and h_{oe} for condenser and evaporator respectively can be calculated using standard equations available in text books.

The overall heat transfer coefficient U_o based on the outside tube surface can then be expressed as

$$\frac{1}{U_0} = [\frac{1}{h_0} + R_{f0} + \frac{r_0 \, l_n \, (r_0)}{k} + (\frac{r_0}{r_i}) R_{fi} + (\frac{r_0}{r_i}) \frac{1}{h_r}] \quad \text{..........................(5)}$$

where R_{fo} and R_{fi} are the fouling factors at outside and inside of the tube respectively.

The heat transfer rate, Q, is then

$Q = F \, U_o \, A_o \, (\Delta t)_{ln}$, $(\Delta t)_{ln}$ is the log-mean temperature difference.

Design of condenser

For the design of condenser, high heat transfer rate per unit area of the condenser tube surface between the condensing refrigerant and the cooling medium must be achieved. This requires that the overall coefficient U_o should be made as high as possible. It can be seen that U_o has always a value lower than the smaller of the two coefficients h_{ic} and h_{oc}. When air is the fluid on the outer side, $h_{oc} << h_{ic}$. U_o is even lower than h_{oc} , however large h_{ic} may be. Therefore, coefficient h_{oc} should be increased in order to increase U_o. This is achieved by providing extended surfaces (or fins) on the air side of the condenser, as in the air-cooled condenser. In shell-and-tube condenser, on the other hand, the water-side heat transfer coefficient may be as high as 5,000 W/m^2K. Condensing coefficients h_{ic} for R-12, R-22 and R-717 are calculated and tabulated below for the following operating conditions of a vapour compression refrigeration system:

Cooling capacity	:	3.517 kW (1 TR)
Evaporating temperature	:	-15^0C
Condensing temperature	:	30^0C
Tube material		Copper. Single-tube condenser and evaporator.
Tube diameter	:	12.7 mm o.d. with tube thickness of 1.016 mm.

Liquid is saturated before expansion, vapour is saturated at outlet of evaporator, and compression process is isentropic.

Using equation (1), (2) or (3), the condensing coefficients are calculated and tabulated below.

Refrigerant	Condensing heat transfer coefficient, w/m^2K
R-12	2,650
R-22	2,400
R-717	11,000

Hence, in ammonia condensers, fins may be added on the water-side; whereas in R-12 and R-22 condensers, fins should be added on the refrigerant side in order to increase the overall heat transfer coefficient.

Design of evaporator

The design of evaporator also depends on high heat transfer rate between the medium to be cooled and the boiling refrigerant. In air conditioning applications, evaporators with extended surfaces on outside of tubes are used since the air side coefficient is much smaller than the boiling refrigerant coefficient. Combined with forced convection of air over the extended surfaces and with few rows deep coil, high rate of heat transfer can be achieved in direct-expansion and flooded-type evaporators. In shell-and-tube evaporators, on the other hand, the

overall coefficient is already high because of boiling refrigerant on one side and a flowing liquid on the other side. For sensible cooling (without water vapour condensation) of air, the air side coefficient for finned-tube surface is given by

$$h_0 = \frac{A_0}{A_i}\,\eta_{fin}\,h_0' \qquad \text{..........(6)}$$

Where, (A_0 / A_i) is the ratio of the outside area to inside area of the finned-tube, η_{fin} is the fin efficiency, and h_0 ' is the heat transfer coefficient on the air side without fins. In fin designs, (A_0 / A_i) is made such that the product ($\frac{A_0}{A_i}\eta_{fin}$) is much higher than unity. For cooling and dehumidifying coils, simultaneous heat and mass transfer on the air side must be considered for designing such a coil, because the coil is wet on the outside. The refrigerant side coefficient h_{ie} is much higher than the air side coefficient h_{oe}. Therefore, attention must be given to increase h_{oe} rather than increasing h_{ie}. On the other hand, in the case of evaporators for chilling liquids such as brine-coolers, water-chillers, and for ice plants, the liquid side coefficient is generally higher than the refrigerant side coefficient. For a 1.0 TR cooling capacity system with operating conditions stated earlier, the boiling heat transfer coefficients h_{ie} have been calculated for R-12 and R-22; and that for R-717 has been taken from ref(3). These values are tabulated below.

Refrigerant	Condensing heat transfer coefficient, h_{ie}, W/m^{2}K
R-12	2,510
R-22	3,070
R-717	2,280

The water-side coefficient may be as high as 5,000 W/m^{2}K, which is considerably higher than the boiling heat transfer coefficients. Therefore, h_{ie} should be improved to obtain higher overall coefficient. This may be achieved by intensification of the boiling process using small hydraulic diameter tubes with rough surfaces.

Overall heat transfer coefficient for evaporator

Using equation (5), the overall coefficients based on the outside surface of the evaporator tubes have been calculated for R-12, R-22 and R717, and are tabulated below.

Refrigerant	h_{oe}, W/m^{2}K	h_{ie}, W/m^{2}K	U_o, W/m^{2}K
R-12	4,950	2,510	1,300
R-22	4,960	3,070	1,470
R-717	4,960	2,280	1,225

By increasing h_{ie}, say, upto the value of the water-side coefficient (i.e. $h_{ie} = h_{io}$ = 4,960 W/m^{2}K) for R-12, R-22 and R-717, we calculate U_o, = 1,870 W/m^{2}K which shows an increase of 44% for R-12, 27% for R-22 and 53% for R-717. Therefore, for design of evaporators, the refrigerant side heat transfer coefficient should be increased by suitable augmentation process.

In the case of condensers, similar conclusions can be made for R-12 and R-22. For condensers with R-717, the water-side coefficient should be increased as far as practicable.

The above calculations have been made for simple heat exchangers. Although this does not provide precise prediction of performance of an actual condenser or and evaporator under real operating conditions, it gives an insight in their designs. Constant heat transfer coefficients over the entire heat exchanger length has been assumed which may introduce some error in the calculations. If the heat transfer coefficients are not assumed constant, the computations become complex and requires computer solutions. In such case heat transfer coefficients are to be computed at each end of the heat exchanger using the inlet and outlet cooling fluid temperatures. Then the following equation should be used.

$$Q = F\,A \left[\frac{U_2\,\Delta t_1 - U_1\,\Delta t_2}{\ln \dfrac{U_2\,\Delta t_1}{U_1\,\Delta t_2}} \right] \quad \text{instead of the equation} \quad Q = F\,U\,A \left[\frac{\Delta t_1 - \Delta t_2}{\ln \dfrac{\Delta t_1}{\Delta t_2}} \right]$$

where subscript 1 refers to one end and subscript 2 refers to the other end.

Conclusions

From the present study, the following conclusions may be drawn.

The overall coefficient of heat transfer is the most important single factor in the designs of condenser and evaporator of a refrigeration system. In air-cooled condensers, fins should be added on the air side to increase the overall coefficient. For water-cooled ammonia condenser, extended surface should be added on the water side, whereas for water-cooled R-12 and R-22 condensers it should be added on the refrigerant side.

In evaporators for cooling and dehumidifying moist air, fins are to be used on the air side, whereas in evaporators used for liquid chilling, the boiling process should be intensified by some
suitable means. Increasing the refrigerant-side coefficient of heat transfer upto the water-side coefficient, it is possible to increase the overall heat transfer coefficient by about 44% for R-12, 27% for R-22 and 53% for R-717.

References

[1] McAdams W.H. Heat Transmission. McGraw-Hill Book Co. Inc., 1954

[2] McQuiston F.C. and Parker J.D. Heating, Ventilating and Air Conditioning – Analysis and Design. John Wiley and Sons 1982.

[3] Handbook of Fundamentals. American Society of Heating, Refrigeration and Air Conditioning Engineers. 1967.

Nomenclature

Symbol	Meaning	Symbol	Meaning
A	Surface area	l	Length
c_p	Specific heat at constant pressure	m	Mass flow rate
d	Tube diameter	Q	Heat transfer rate
F	Log-mean temperature difference correction factor	r	Tube radius
G	Mass velocity	U	Overall heat transfer coefficient
g	Acceleration due to gravity, 9.81 m/s^2	x	Dryness fraction
h	Heat transfer coefficient	ρ	Density
J	Joule's equivalent, 1.0 in S.I. Units	μ	Dynamic viscosity
k	Thermal conductivity	η	Efficiency
L	Latent heat		

Subscripts

b	Bulk fluid	i	Tube inside
c	Condenser or cold	l	Liquid
e	Evaporator	o	Tube outside
f	Fouling	v	Vapour
h	Hot	w	Water

APPENDIX

Sample calculations for that transfer coefficients are given below. A refrigeration cycle with the following operating conditions is assumed.

Refrigerant = R-12
Cooling capacity = 1 TR = 3.517 kW
Condensing temperature = 30^0C
Evaporating temperature = -15^0C
Liquid entering expansion device is saturated
Vapour entering compressor is dry and saturated
Compression is isentropic

From tables for properties of R-12
h_1 = 344.92 kJ/kg.
$h_3 = h_4$ = 228.54 kJ/kg.

Dryness fraction of vapour leaving evaporator, $x_1 = 1.0$
Dryness fraction of liquid-vapour mixture entering evaporator, $x_4 = 0.27$

Mass flow rate of refrigerant, m_r is calculated as 108.8 kg/h.

Condensing heat transfer coefficient, h_{ic}

Inner diameter, d_i = 12.70 – 2 x 1.016 = 10.70 mm = 0.0107 m.

Then, cross-sectional area, $A_i = \frac{\pi}{4} d^2 = \frac{\pi}{4}(0.0107)^2 = 8.99 \times 10^{-5}\ m^2$

Then, $G_1 = G_v = \frac{108.8}{0.0000899} = 1.21 x 16^2 \frac{kg}{h\,m^2}$

For R-12 at 30^0C,
L=135.04 kJ/kg, μ_1 = 0.744 kg/m h, μ_v = 0.0462 kg/m h
c_{p1} = 0.985 kJ/kgK, ρ_1 = 1292.0 kg/m^3, ρ_v = 42.5 kg/m^3,
k_1 = 0.068 W/mK.

Then, $\frac{d_i\,G_l}{\mu_1} = \frac{0.0107\,x\,(1.21\,x\,10^6)}{0.744} = 17{,}400$, which is greater than 5,000.

and

$$\frac{d_i\,G_v}{\mu_1}\,(\rho_1/\rho_v)^{\frac{1}{2}} = \frac{0.0107x(1.21x10^6)}{0.744}\,x\left(\frac{1292.0}{42.5}\right)^{\frac{1}{2}} = 95{,}940 \text{ which is greater than } 20{,}000.$$

Hence, using eq (3) for condensation inside horizontal tubes,

$$\frac{h_{ic}\ x\, 0.0107}{0.068} = 0.026\left\{\frac{0.985\, x\, 0.744}{0.068}\right\}^{\frac{1}{3}}\left\{\frac{0.0107}{0.744}\left[(1.21\, x10^{6})\, x\left(\frac{1292.0}{42.5}\right)^{\frac{1}{2}} + (1.21\, x10^{6})\right]\right\}^{0.80}$$

Or, h_{ic} = 2,650 W/m^{2}K.

Boiling heat transfer coefficient, h_{ie}

For boiling of R-12 inside horizontal copper tubes, from Eq (4),

$$h_{ie} = 0.0082\ (k_1/d_i)\left[\left(\frac{G_1\ d_i}{\mu_1}\right)^2\left(\frac{(J\,(\Delta x)\,L\,g_c}{l\ x\ g}\right)\right]^{0.40} \qquad \text{for } x > 0.90.$$

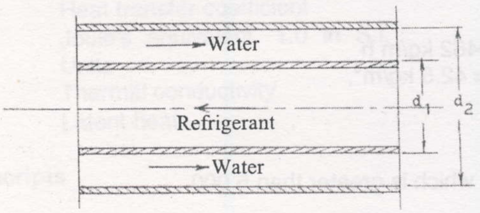
$\frac{G_1 d_i}{\mu_1}$ = 17,400 (as calculated earlier). Assuming a tube length of 6.0 m,

$$\frac{J\,(\Delta\, x)\, L\, g_c}{l\ x\ g} = \frac{1.0\, x\,(1-0.27)\, x\,(135.04\, x\,1000)\, x1}{6.0\ x\ 9.81} = 1675.0$$

Then, $h_{ie} = 0.0082 \times \frac{0.068}{0.0107}\ [\,(17{,}400)^2 \times 1675.0]^{0.40} = 2{,}510$ W/m^{2}K.

Water-side heat transfer coefficient, h_0

We have d_1 = 12.7 mm = 0.0127 m.
Let us assume d_2 = 25.4 mm = 0.0254 m.
$d_a = d_2 - d_1$. See diagram below.



For flow through an annular space, the following equation can be used.

$$h_o = \frac{0.023}{\left(\frac{d_a G}{\mu_b}\right)^{0.20}} x \left(\frac{\mu_b}{\mu_w}\right)^{0.14} x \left(\frac{k_b}{c_{pb}\mu_b}\right)^{2/3} x (c_{pb}\ x G)$$

Taking a bulk temperature of water as 20^0C, the following property values of water are noted from tables.

$k_b = 6.04 \times 10^{-4}$ kW/mK, $c_{pb} = 4.18$ kJ/kgK, $\mu_b = 9.80 \times 10^{-4}$ kg/m s, $\rho_b = 998.0$ kg/m^3.

Then, $\frac{k_b}{c_{pb}\mu_b} = \frac{6.04\ x\ 10^{-4}}{4.18\ x\ (9.8\ x 10^{-4})} = 0.148$

A reasonable water velocity through a condenser or an evaporator is 1.20 m/s. Then,

$G = (1.20)\ (998.0)\ (3600) = 4.38 \times 10^6$ kg/m^{2}h.

$d_a = 0.0254 - 0.0127 = 0.0127$ m.

Assuming an average wall temperature of 27^0C, $\mu_w = 8.60 \times 10^{-4}$ kg/m s.

Then, $(\mu_b / \mu_w) = \frac{9.80 x 10^{-4}}{8.60 x 10^{-4}} = 1.14$

$$\frac{d_a G}{\mu_b} = \frac{0.0127\ x\ (4.38 x 10^6)}{(9.80\ x\ 10^{-4})\ x\ 3600} = 1.58\ x\ 10^4$$

$$h_0 = \{\frac{0.023}{(1.58 x 10^4)^{0.20}}\} \times (1.14)^{0.14} \times (0.148)^{2/3} \times \{4.18 \times (4.38 \times 10^6)\}$$

$$= 17{,}807\ \frac{kJ}{hm^2 K} = 4950.0\ \text{W/m}^2\text{K}$$

<u>Overall heat transfer coefficient, U_o</u>

Now U_0 will be calculated for the evaporator. Similar calculations can be made for the condenser.

We have, $h_{ie} = 2510.0$ W/m^{2}K

$h_{io} = 4950.0$ W/m^{2}K

Also, fouling facors $R_{fo} = 8.80 \times 10^{-5}$ m^{2}K/W, for ordinary water,

$R_{fi} = 0$, assuming refrigerant side as clean.

k = 386.0 W/mK for copper.

$r_o = 0.00635$ m, $r_i = 0.00535$ m.

Substituting the above values in eq (5),

$$\frac{1}{U_{oe}} = 7.69 \times 10^{-4}\ \text{m}^2\text{K/W}$$

or $U_{oe} = 1300$ W/m^{2}K.

This value of U_{oe} has been shown in the table for R-12 earlier.