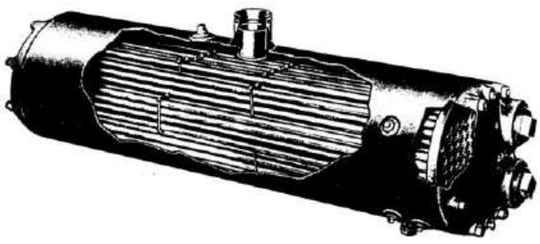
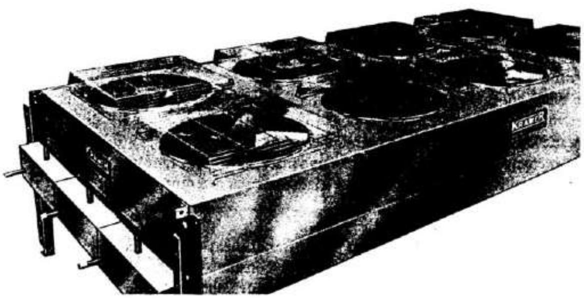
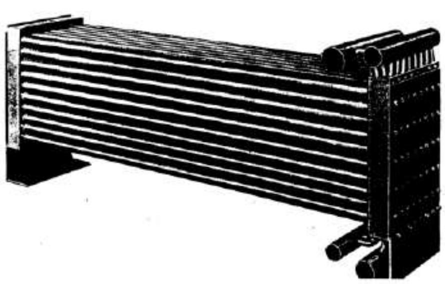
Ammar Abdulkadhim (M.Sc.)
E-mail: AmmarAbdulkadhim@mustaqbal-college.edu.iq

Chapter Two

Condenser: Fluid Flows and Heat Transfer Analysis

1. Introduction

- The previous sections presented tools for computing heat-transfer coefficients and pressure drops of the fluid exchanging heat with the refrigerant in a condenser or evaporator. For the condenser the fluid to which heat is rejected is usually either air or water. Air-cooled condensers, a shell and- tube condenser. Also, another type of water-cooled condenser has cleanable tubes ash shown below.
- When the condenser is water-cooled, the water is sent to a cooling tower for ultimate rejection of the heat to the atmosphere. Some years ago air-cooled condensers were used only in small refrigeration systems (less than 100 kW refrigerating capacity), but now individual air-cooled condensers are manufactured in sizes matching refrigeration capacities of hundreds of kilowatts.
- The water-cooled condenser is favored over the air-cooled condenser where there is a long distance between the compressor and the point where heat is to be rejected. Most designers prefer to convey water rather than refrigerant in long lines. In centrifugal-compressor systems large pipes are needed for the low-density refrigerants (see Sec. 11-25), so that the compressor is close-coupled to the condenser. Water-cooled condensers there for predominate in centrifugal-compressor systems.

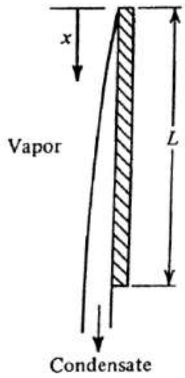
2. Required condensing capacity

- The required rate of heat transfer in the condenser is predominately a function of the refrigerating capacity and the temperatures of evaporation and condensation. The condenser must reject both the energy absorbed by the evaporator and the heat of compression added by the compressor. A term often used to relate the rate of heat flow at the condenser to that of the evaporator is the *heat-rejection ratio*

$$\text{Heat-rejection ratio} = \frac{\text{rate of heat rejected at condenser, kW}}{\text{rate of heat absorbed at evaporator, kW}}$$

3. Condensing coefficient

The basic equation for calculating the local coefficient of heat transfer of vapor condensing on a vertical plate was developed by Nusselt by pure physical analysis. The equation for the local condensing coefficient is



$$\frac{h_{cv} x}{k} = \left(\frac{g\rho^2 h_{fg} x^3}{4\mu k \, \Delta t}\right)^{1/4}$$

where h_{cv} = local condensing coefficient on vertical plate, $W/m^2 \cdot K$
x = vertical distance measured from top of plate, m
g = acceleration due to gravity = 9.81 m/s^2
ρ = density of condensate, kg/m^3
h_{fg} = latent heat of vaporization, J/kg
μ = viscosity of condensate, Pa · s
Δt = temperature difference between vapor and the plate, K

The mean condensing coefficient over the total height of the plate L is

$$\overline{h_{cv}} = \frac{\int_0^L h_{cv}\, dx}{L} = 0.943 \left(\frac{g\rho^2 h_{fg} k^3}{\mu \, \Delta t \, L}\right)^{1/4} \quad W/m^2 \cdot K$$

The equation for the mean condensing coefficient for vapor condensing on the outside of horizontal tubes is

$$h_{ct} = 0.725 \left(\frac{g\rho^2 h_{fg} k^3}{\mu \, \Delta t \, ND}\right)^{1/4} \quad W/m^2 \cdot K$$

where N = number of tubes in vertical row
D = OD of tube, m

4. Fouling factor

After a water-cooled condenser has been in service for some time its *U* value usually degrades somewhat because of the increased resistance to heat transfer on the water side due to fouling by the impurities in the water from the cooling tower. The new condenser must therefore have a higher *U* value in anticipation of the reduction that will occur in service. The higher capacity with new equipment is provided by specifying a *fouling factor* l/h_{ff} m2 • K/W. This term expands Eq**. (12-8)** for the *U* value into

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{xA_o}{kA_m} + \frac{A_o}{h_{ff}A_i} + \frac{A_o}{h_i A_i}$$

Several different agencies have established standards for the fouling factor to be used. One trade association[12] specifies 0.000176 $m^2 \cdot K/W$, which means that the condenser should leave the factory with a $1/U_o$ value 0.000176 A_o/A_i less than the minimum required to meet the quoted capacity of the condenser.

5. Desuperheating

Even when the refrigerant condenses at a constant pressure, its temperature is constant only in the condensing portion. Because the vapor coming from the compressor is usually superheated, the distribution of temperature will be as shown in Fig. 12-14. Because of the distortion in the temperature profile caused by the desuperheating process, the temperature difference between the refrigerant and the cooling fluid is no longer correctly re presented by the LMTD

$$\text{LMTD} = \frac{(t_c - t_i) - (t_c - t_o)}{\ln [(t_c - t_i)/(t_c - t_o)]}$$

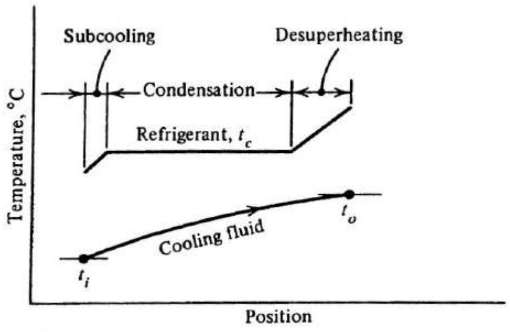


Figure 12-14 Temperature distributions in a condenser.

6. Condenser design

An example will illustrate how some of the principles described in the previous sections are combined in designing a condenser.

Example 12-3 The condensing area is to be specified for a refrigerant 22 condenser of a refrigerating system that provides a capacity of 80 kW for air conditioning. The evaporating temperature is 5°C, and the condensing temperature is 45°C at design conditions. Water from a cooling tower enters the condenser at 30°C and leaves at 35°C.

A two-pass condenser with 42 tubes, arranged as shown in Fig. 12-15, will be used, and the length of tubes is to be specified to provide the necessary area. The tubes are copper and are 14 mm ID and 16 mm OD.

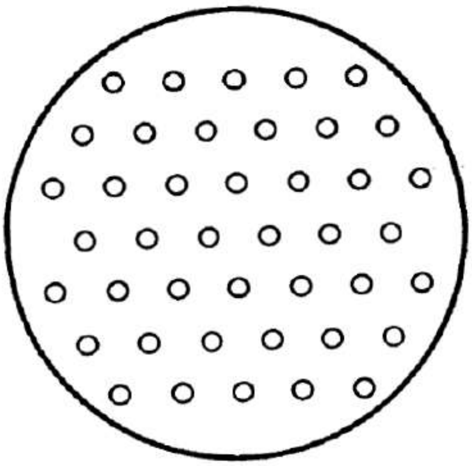


Figure 12-15 Tube arrangement of condenser in Example 12-3.

Solution;
Heat-rejection ratio at a condensing temperature of 45°C and an evaporating temperature of 5°C is 1.27

Ammar Abdulkadhim (M.Sc.)
E-mail: AmmarAbdulkadhim@mustaqbal-college.edu.iq

$$q = (80\ \text{kW})\,(1.27) = 101.6\ \text{kW}$$

Condensing coefficient From Eq. (12-24)

$$h_{\text{cond}} = 0.725\left(\frac{g\rho^2 h_{fg} k^3}{\mu\,\Delta t\,ND}\right)^{1/4}$$

The density ρ and latent heat of vaporization h_{fg} at 45°C are available from Table A-6

$$\rho = \frac{1}{0.90203\ \text{L/kg}} = 1.109\ \text{kg/L} = 1109\ \text{kg/m}^3$$

$$h_{fg} = 160{,}900\ \text{J/kg}$$

The conductivity k and viscosity μ of the liquid refrigerant at 45°C are available from Table 15-5

$$k = 0.0779\ \text{W/m}\cdot\text{K} \qquad \mu = 0.000180\ \text{Pa}\cdot\text{s}$$

The average number of tubes in a vertical row N is

$$N = \frac{2+3+4+3+4+3+4+3+4+3+4+3+2}{13} = 3.23$$

The temperature difference between the vapor and the tube is unknown at this point; therefore Δt will be assumed to be 5 K and the value adjusted later if necessary.

$$h_{\text{cond}} = 0.725\left[\frac{9.81(1109^2)\,(160{,}900)\,(0.0779^3)}{0.000180(5)\,(3.23)\,(0.016)}\right]^{1/4}$$

$$= 1528\ \text{W/m}^2\cdot\text{K}$$

Resistance of metal The conductivity of copper is 390 W/m · K, and the resistance of the tube is

$$\frac{xA_o}{kA_m} = \frac{(0.016-0.014)/2}{390}\,\frac{16}{(14+16)/2} = 0.000002735\ \text{m}^2\cdot\text{K/W}$$

a value that will prove to be negligible in comparison to the other resistances.

Fouling factor From Sec. 12-10

$$\frac{1}{h_{ff}} = 0.000176\ \text{m}^2\cdot\text{K/W}$$

Water-side coefficient The flow rate of water needed to carry the heat away from the condenser with a temperature rise from 30 to 35°C is

$$\frac{101.6 \text{ kW}}{(4.19 \text{ kJ/kg} \cdot \text{K})(35.0 - 30.0)} = 4.85 \text{ kg/s}$$

and the volume flow rate is

$$\frac{4.85 \text{ kg/s}}{1000 \text{ kg/m}^3} = 0.00485 \text{ m}^3/\text{s}$$

The water velocity through the tubes V is

$$V = \frac{0.00485 \text{ m}^3/\text{s}}{(21 \text{ tubes per pass})(\pi/4)(0.014^2)} = 1.5 \text{ m/s}$$

Equation (12-9) can be used to calculate the water-side heat-transfer coefficient h_w using the water properties at 32°C

$$\rho = 995 \text{ kg/m}^3 \qquad \mu = 0.000773 \text{ Pa} \cdot \text{s}$$

Table 7-3

$$c_p = 4190 \text{ J/kg} \cdot \text{K} \qquad k = 0.617 \text{ W/m} \cdot \text{K}$$

$$h_w = \frac{0.617(0.023)}{0.014}\left[\frac{1.5(0.014)(995)}{0.000773}\right]^{0.8}\left[\frac{4190(0.00773)}{0.617}\right]^{0.4}$$

$$h_w = 1.014(27030^{0.8})(5.25^{0.4}) = 6910 \text{ W/m}^2 \cdot \text{K}$$

Required area

$$\frac{1}{U_o} = \frac{1}{1528} + 0.000002735 + \frac{0.016}{0.014}(0.000176) + \frac{0.016}{0.014}\frac{1}{6910} = 0.001023$$

$$U_o = 977 \text{ W/m}^2 \cdot \text{K}$$

The LMTD is

$$\text{LMTD} = \frac{(45 - 30) - (45 - 35)}{\ln \dfrac{(45 - 30)}{(45 - 35)}} = 12.33°\text{C}$$

$$A_o = \frac{101{,}600 \text{ W}}{977(12.33)} = 8.43 \text{ m}^2$$

Length of tubes

$$\text{Length} = \frac{8.43 \text{ m}^2}{(42 \text{ tubes})(0.016\pi)} = 4.0 \text{ m}$$