Parts C and D of Fig. 6 show values of pressure as a function of length of travel, water quantity, and loading. Part C is based on a loading of 8 sq ft per ton, while Part D provides corrections for other loadings. The values of pressure drop used on Fig. 6 are based on a nominal tube length of 14 ft. Pressure loss in the water heads is based on Fig. C-5 of Reference 5.

Example 3: Select a water-cooled condenser for 75 tons of refrigeration with an entering water temperature of 80 F, reasonable water cost and average power cost.

Solution: At a flow rate of 5 gpm per ton, the leaving terminal difference (Part A of Fig. 6) is reasonable at a length of travel of 56 ft (4 pass, 14 ft condenser). The terminal difference would be 6.4 F at a loading of 8 sq ft per ton and 5.4 F for a loading of 10 sq ft per ton (Part B of Fig. 6).

For a 56 ft travel and 5 gpm per ton, the resulting pressure drop would be 4.9 psi at a loading of 8 sq ft per ton (Part C, Fig. 6) and 3.3 psi at 10 sq ft per ton (Part D, Fig. 6). If the lower pressure drop were selected, the condenser would require about 750 sq ft of external tube surface and would have a leaving terminal difference of approximately 5.4 F deg plus 1.5 F deg for noncondensable gases, or 6.9 F deg. The condensing temperature would be the sum of (1) the entering water temperature, (2) the temperature rise of 30 gal deg per (min) (ton) ÷ gpm per ton, and (3) the leaving terminal difference, or 80 + 6.0 + 6.9 = 92.9 F. This corresponds to a condensing pressure of 174.6 psig.

Although the optimum condenser selection depends upon the conditions of each installation, the prevailing power and water rates, and the quantity of water available, the above selection would be satisfactory for average water and operating conditions. For use with water which will cause heavy scaling, the condenser should be selected for low values of terminal difference and on the flatter portion of the performance curves, where a change in heat transfer will result in a smaller change in condensing pressure.

Fig. 6 shows that, at low water quantities, short lengths of travel, and low values of square feet per ton, the leaving terminal difference becomes excessive. At such conditions, a small change in heat transfer would mean a large change in condensing pressure. Conversely, the curves show that, at high water quantities for long lengths of travel and relatively small values of square feet per ton, the pressure drop becomes excessive. The curves therefore provide a direct comparison and a balance of conditions to give a practical and complete analysis of condenser performance.

In Fig. 7 the test results of other investigations on ammonia condensers have been condensed to express the heat transfer as a function of the water velocity. Curve A is for clean tubes. Curve B refers to data taken in the same manner as for Curve A, after the tubes had been in service 16 hr. Curve C refers to dirty tubes, although the length of time the condenser had been in service was not stated.

The variation in heat transfer as a function of the length of time the condenser has been in service is clearly indicated by the progressively lower values as indicated by Curves A, B, and C of Fig. 7. Ferrous tubes seem to have an increasing scale resistance, even with normal water, due to the continuous oxidation of inside tube surfaces. High rates of heat transfer would usually indicate that condensers having new clean tubes were tested.

Curve D of Fig. 7 shows the coefficient of heat transfer that may be expected in practice after the condenser has been in service a reasonable length of time. The curves of Fig. 7 are based on external tube surface actual temperature differences measured at various points and averaged. The product of the logarithmic mean temperature difference and the coefficient of heat transfer from Curve D will yield the heat transfer with sufficient precision for application purpose. Values so obtained will be in agreement with data from Reference 6.

The water friction pressure loss, important on horizontal condensers, consists of the loss through the tubes plus the loss resulting from entry conditions, return bends and changes in cross-section area. Fig. 8 shows expected pressure loss through tube circuits and in heads forming the multipass arrangement.

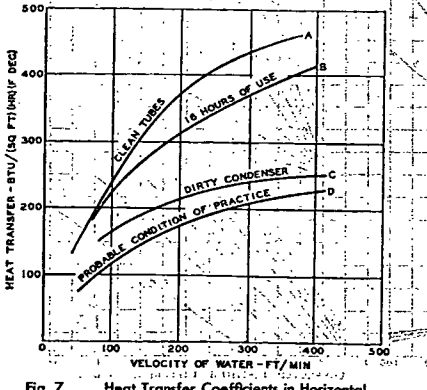


Fig. 7 .... Heat Transfer Coefficients in Horizontal Shell-and-tube Ammonia Condensers

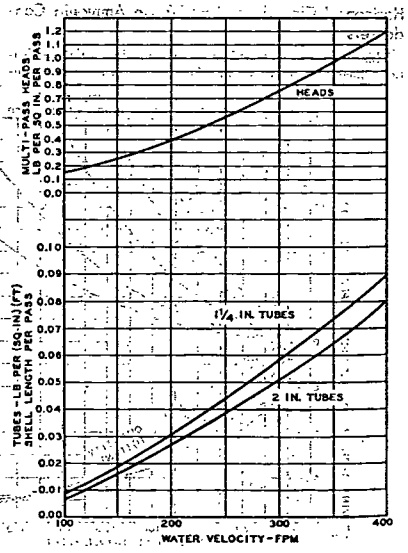


Fig. 8 .... Water Friction Pressure Loss in Horizontal Multipass Condensers

Tube pressure loss is given per foot of tube. Pressure drop through the heads includes loss in change of direction and all other losses for one pass.

Research on performance of ammonia condensers of various types by Linge[7] has brought out some important conclusions which follow.

Condensation takes place in the superheated part of the condenser as long as the walls are cooled below the condensing temperature.

The product of the coefficient of heat transfer and the mean temperature difference between the vapor and the wall is practically the same for superheated vapor and saturated vapor.

When the superheat increases, the transmission coefficient decreases to about the same extent as the temperature difference increases; thus the quantity of heat transmitted remains about the same. Therefore, an average overall heat transfer coefficient and the mean temperature difference calculated by using the condensing temperature corresponding to the saturated condensing pressure and the entering and leaving water temperatures, may be used to give reasonably accurate condenser ratings.

Vertical Open Shell-and-Tube Ammonia Condenser

The characteristics of this type of condenser are: (a) large condenser capacity may be installed in small floor space, (b) complete installation cost is low, (c) water distribution is simplified, (d) large space is available in the condenser for storing liquid ammonia, (e) it is adaptable to easy purging of contaminant gases, (f) tubes may be easily cleaned without shutting down, (g) the condenser can be made to carry large overloads by increasing the quantity of water circulated through the tubes while, at the same time, avoiding a heavy increase in friction head, and (h) it can be used effectively with condensing water from a cooling tower or spray pond. The construction of a typical vertical ammonia shell-and-tube condenser is shown in Fig. 9. It is common in sizes varying from 16 to 60 in. in diameter, with a range of 20 to 400 tubes, respectively, and a range in heights from 10 to 16 ft, with 14 ft a common height. As a rule, the seamless steel tubes are 2-in. OD of 0.120 in. average wall thickness.

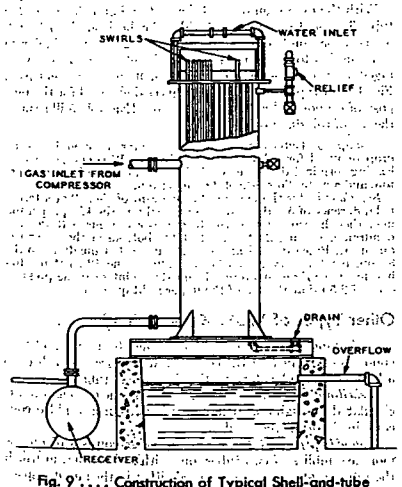


Fig. 9 .... Construction of Typical Shell-and-tube Condenser

The top of each tube is fitted with a water distributor made of cast iron, brass, plastic, glass, or porcelain, to give the water a swirling motion as it enters the tube.

Fig. 10 shows overall heat transfer coefficients for 2 in. and 2½ in. OD tubes at various rates of water flow per tube.

Fig. 11 shows typical values of leaving terminal difference (temperature corresponding to ammonia pressure minus leaving water temperature) as a function of water flow in gallons per minute per tube for loadings from 5 to 18 sq ft of external tube surface per ton and for condensing water quantities from 1 to 5 gpm per ton. The values are for commercially clean 2-in. OD steel tubes with a nominal length of 14 ft, and are based on a condenser free of noncondensable gases. The rise in the condensing water temperature is normally based on a factor of 30 gal deg per (min) (ton), and the heat rejection from the ammonia vapor is 15,000 Btuh per ton. Note that evaporative cooling effect of air within the tubes reduces the

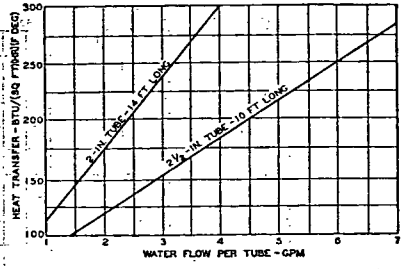
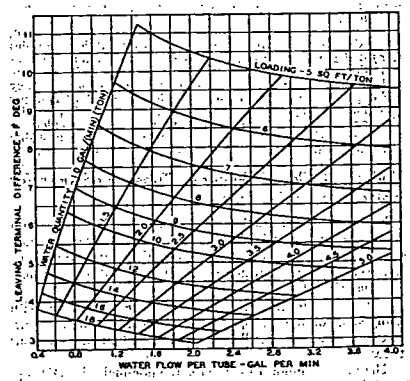


Fig. 10 .... Heat Transfer Coefficients for Vertical Shell-and-tube Ammonia Condensers



* For commercially clean 2 in. steel tubes, 14 ft long, with condenser free of noncondensable gases.

Fig. 11 .... Performance of Vertical Open Shell-and-tube Ammonia Condensers*