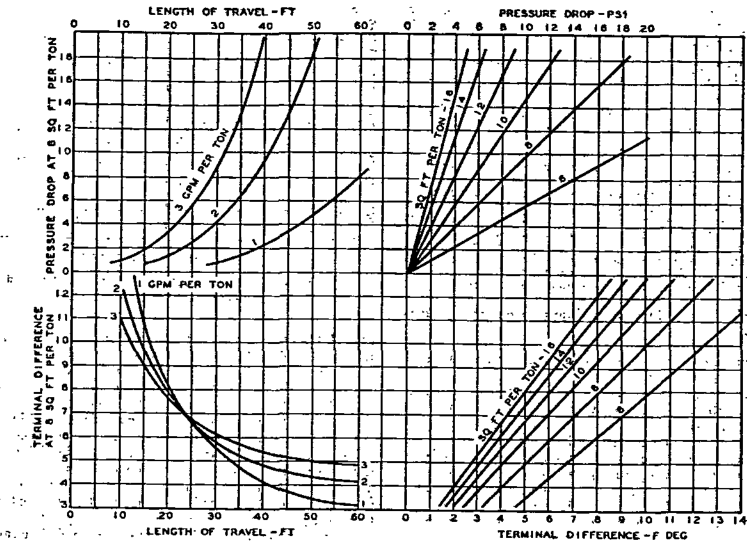


* ⅝ in. OD tubes, 19 fins per inch, 1/16 in. high.

Fig. 12 .... Rating Diagram for Horizontal Refrigerant 12 Condenser*

factor from 30 to 28 on vertical open shell-and-tube condensers.

*Example 4:* Select a condenser for 60 tons of refrigeration with 240 gpm water available for condensing at an entering temperature of 80 F.

*Solution:* The water flow rate is 4 gpm per ton (240 ÷ 60), thus fixing the temperature rise at 7 F deg (28 gal deg per (min) (ton) ÷ 4 gpm per ton), and the leaving water temperature at 87 F. From Fig. 11, at a loading of 10 sq ft per ton and a flow rate of 4 gpm per ton, the leaving terminal difference is approximately 5 F deg, and the corresponding water flow rate is 2.94 gpm per tube. The resulting condensing temperature would be 87 + 5 + 1.5 (for noncondensables) or 93.5 F, corresponding to a condensing pressure of 176.4 psig. A lower terminal difference probably would not be justified due to the required increase in surface.

## Horizontal Closed Shell-and-Tube Condenser for Halocarbon Refrigerants

Horizontal closed shell-and-tube condensers for halocarbon refrigerants are built in sizes from about five tons up to those required on the largest installations. In the large installations, however, greater operating economy is obtained by a multiple condenser installation.

Halocarbon refrigerant condensers have been made with many types of material specifications, including all prime surface ferrous tubes and all prime surface or finned, nonferrous tubes.

The most common tube used is the nominal ⅝-in. OD copper tube with integral fins on the outside, 19 per inch and 1/16 in. high. These tubes are either silver brazed into thin copper or steel tube sheets, or rolled into heavier nonferrous or steel tube sheets.

With the exception of the tube material and the use of the tube fins on the refrigerant side, the horizontal shell-and-finned-tube condenser for Refrigerant 12 is very similar in construction to, and has the same advantages and disadvantages as, the ammonia horizontal shell-and-tube condenser. Fig. 12 is a typical set of rating curves developed for the purpose of economical condenser selection. Example 5 illustrates the use of this diagram.

*Example 5:* Determine the length of water travel and pressure drop required for a horizontal Refrigerant 12 condenser with a leaving terminal difference of 5 F deg, a loading of 12 sq ft per ton, and a water flow rate of 3 gpm per ton.

*Solution:* Enter the lower right quadrant of Fig. 12 at a terminal difference of 5 F deg, proceed vertically to the 12 sq ft per ton line, then horizontally to the 3 gpm per ton curve (lower left quadrant), reading 24 ft travel at the bottom of the lower left quadrant. Proceed vertically into the upper left quadrant to the 3 gpm per ton curve, then horizontally to the 12 sq ft per ton line in upper right quadrant. From this point read the pressure drop of 2.5 psi at the top of the upper right quadrant.

## Other Types of Water-Cooled Condensers

A *shell-and-U-tube condenser* is similar to a shell-and-tube condenser, except that the cooling water is circulated through a bundle of U-tubes terminating in a single tube sheet.

A *shell-and-coil condenser* is one wherein the cooling water is circulated through one or more continuous or assembled coils contained within the shell.

A *double-pipe* or *tube-in-tube condenser* consists of one or more assemblies of two tubes, one within the other, in which the refrigerant vapor is condensed either in the annular space or in the inner tube.



An *atmospheric condenser* consists of a number of horizontal tubes arranged one above the other. The refrigerant is condensed within the tubes, and the cooling water is distributed along the top tube, flowing by gravity over the tubes directly below. Sufficient water must be circulated to obtain uniform and complete wetting of the tubes.

The principles of calculation and rating given for shell-and-tube condensers apply equally for the other condensers.

## Noncondensable Gases

Noncondensable gases circulate with the refrigerant, but unlike the refrigerant, do not liquefy in the condenser. They may be one or a mixture of two or more of the following: nitrogen, hydrogen, oxygen, chlorine, oil vapors, water vapors, etc.

Noncondensable gases come (a) from outside the system by being drawn into the crankcase of the compressor along with the oil and through leaks in piping, stuffing boxes, etc., when the system is being operated under a vacuum, and (b) from inside the system by improper evacuation of a new system before charging, by decomposition of the lubricant under the heat of compression, and (c) by miscellaneous chemical reactions within the system.

When noncondensable gases are present, they collect on the high side of the system, resulting in a condensing pressure above that corresponding to the temperature at which the refrigerant is actually condensing. The excess pressure is caused in part by the partial pressure effect of the gases and in part by the effect of these gases in forming a resistance film over some of the condensing surface, thus lowering the heat transfer coefficient.

Due to the continuous flow of vapor and gases into the condenser, the noncondensable gases will tend to accumulate in the coldest and least agitated part of the condenser, or in the receiver. The purge connection for the elimination of these noncondensable gases should, therefore, be made to that part of the high-pressure side of the system.

The presence of noncondensable gases in the condenser not only causes higher head pressures, but also increases brake horsepower, reduces capacity, and promotes oxidation of the oil, especially at the valves, due to the presence of oxygen at this point of high discharge temperatures. This has been confirmed by operating experience with all refrigerating systems. For steam and air, the film resistance coefficient between steam and a metal wall is up to six times as great with 4.5 percent air as it is with no air present.[8]

According to Dalton's law of partial pressures, the presence of these gases, raises the condensing pressure $P$ by the amount of the partial pressure, $P'$. The increase of the work of compression is proportional to the ratio $P + P'/P$ and is greater with higher partial pressures and lower condensing pressures.

A test for noncondensable gases is to shut down the refrigeration system while allowing the condenser water to flow for a sufficient length of time so that the refrigerant is at the same temperature as the water. If the condenser pressure is higher than the pressure corresponding to the refrigerant temperature, there are noncondensable gases present. Experimental data have shown that, with these gases present in the condenser under the above conditions, with a 10 F deg difference between refrigerant temperature and the temperature corresponding to the condensing pressure, there will be as much as 20 to 25 F deg increase in condensing temperature during operation.

Noncondensable gases can be eliminated to a great degree by purging. This may be accomplished by shutting down the system as mentioned above, and purging by blowoff directly

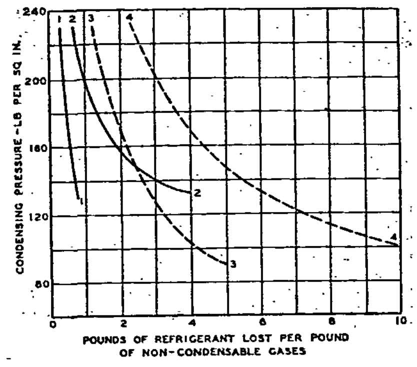


Fig. 13 .... Loss of Refrigerant During Purging at Various Gas Temperatures and Pressures

into the atmosphere or by using devices for this purpose which will operate while the system is in operation. Purging connections should be made where noncondensable gases will accumulate in the system.

Purging devices include a refrigerated surface within a container into which is introduced a mixture of refrigerant gas and noncondensable gases. The low temperature surface condenses a large amount of the refrigerant, and the remaining mixture is allowed to escape to the atmosphere. Fig. 13 shows the refrigerant loss when purging ammonia and Refrigerant 12 at various temperatures and pressures, and indicates the desirability of purging devices. Even though Refrigerant 12 losses are considerably greater than ammonia, it is economical to purge these systems as thoroughly as ammonia systems. This can be determined by balancing the savings in compressor power requirements and increased wear on machinery against the loss of refrigerant.

## Construction Codes

Where state or municipal codes do not take precedence, design *pressures, materials, welding, tests,* and *relief devices* should be in accordance with the *ASME Unfired Pressure Vessel Code*, Section 8, and ASHRAE *Standard 15–63* which is identical with the *American Standard Safety Code for Mechanical Refrigeration*, B9.1–1964.

# PART II: AIR-COOLED CONDENSERS

The heat transfer process in an air-cooled condenser can be subdivided into three main phases: (1) desuperheating, (2) condensing, and (3) subcooling. Fig. 14 shows the changes of state of Refrigerant 12 passing through the condenser coil and the corresponding temperature change of the cooling air in contact with the outside of the condenser. Depending on the refrigerant, the relative limits of desuperheating, condensing, and subcooling zones will vary slightly, but Fig. 14 is typical for most of the commonly used refrigerants.

With Refrigerant 12, the desuperheating zone occupies only 5.2 percent of the condenser coil, due to the large tem-