consumption increases. To obtain this head pressure, the temperature of the circulating water leaving the condenser must always be less than 96 F by an amount depending upon the size and design of the condenser, the quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot water temperature within 3 or 4 F of the ammonia temperature corresponding to the head pressure, while a small condenser might require a 10 F difference.

Table 2 lists several gases with data as to the temperatures and pressures for which commercial condensers are designed. Internal combustion engines have limiting hot water temperatures of 125 F to 140 F for closed systems, and 110 F to 120 F for open systems, depending upon the quality of the cooling water. The cooling of such fluids as milk or wort has variable requirements and is usually done in counter-flow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

The temperature range, once the hot water temperature is approximately known, depends upon:

1. **Maximum wet-bulb temperature at which the full quantity of heat must be dissipated.**
2. **Efficiency of the atmospheric cooling equipment considered.**

Design Wet-Bulb Temperatures

The maximum wet-bulb temperature at which the full quantity of water must be cooled through the entire range is never, in commercial design, the *maximum* wet-bulb temperature ever known to exist at the location nor the *average* wet-bulb temperature over any period. The former basis would require atmospheric cooling equipment several times greater than normal size, and the latter would result during a large part of the time, in higher condenser water temperatures than those for which the plant was designed. For instance, the maximum wet-bulb temperature recorded in New York City is 88 F, and the July noon average for 64 years is close to 68 F. Yet in the years 1925 to 1934, inclusive, there were but 8 hours per year when the wet-bulb temperature reached 80 F or more, and there were 975 hours in the average summer (June to September inclusive) when the wet-bulb temperature was 68 F or above. As these 975 hours represent a third of the summer period, cooling equipment based upon the noon average July wet-bulb of 68 F would be inadequate. Commercial practice is to choose a wet-bulb temperature for air conditioning design purposes which is not exceeded during more than 5 to 8 per cent of the summer hours (75 F for New York City) with somewhat lower requirements for steam turbines and internal combustion engines. This difference is made because the heaviest load on an air conditioning plant is coincident with high wet-bulb temperatures, whereas the heaviest electric power demand occurs either in the winter or after nightfall in summer, when the wet-bulb temperature is low. Table 1, Chapter 7, shows design wet-bulb temperatures which will not be exceeded more than 8 per cent of the time in an average summer.

Knowing the hot water temperature and the wet-bulb temperature for

which the equipment must be designed, the cold water temperature must be chosen to place the requirement within the effectiveness range of the type of atmospheric water cooling apparatus to be used. This effectiveness is expressed as the percentage ratio of the actual cooling effect to the maximum possible cooling effect. Since the wet-bulb temperature of the entering air is the lowest adiabatic equilibrium temperature to which the water could be cooled, the effectiveness based upon the quantity of water and upon this assumed limiting process is equal to:

$$\frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}.$$

Magnitudes of this effectiveness ratio will vary through wide limits in accordance with the many possible types of construction and conditions

Table 3. Effectiveness of Atmospheric Water Cooling Equipment

Equipment	Cooling Effectiveness—Per Cent		
	Minimum	Usual	Maximum
Spray Ponds	30	40 to 50	60
Spray Towers	40	45 to 55	60
Natural Draft Deck or Atmospheric Towers	35	50 to 70	90
Mechanical Draft	35	55 to 75	90

of operation. Values indicative of the commercial range of the effectiveness ratio are given in Table 3, although unusual designs may operate outside these ranges.

From consideration of the factors which include the cooling range and design wet-bulb temperature, the quantity of water required can be calculated from the amount of heat to be dissipated. The normal amounts of heat to be removed from various processes of the cooling equipment are:

Compressor Refrigeration: 220 to 270 Btu per minute per ton. Usual practice is to assume: 250 Btu per minute per ton which is equivalent to 30 gal per degree F per minute per ton.

Steam Turbine Condensers: 950 to 980 Btu per pound of steam. Usual practice is to assume 970 Btu per pound of steam.

Steam Jet Refrigerating Condensers: 1030 to 1150 Btu per pound of steam. Exact value depends upon initial steam conditions.

Diesel Engine Jackets: 2500 to 4000 Btu per BHP per hour. Usual practice is to assume 3500 Btu per BHP per hour.

Natural Gas or Gasoline Engines: 4500 to 6000 Btu per BHP per hour. Usual practice is to assume 5000 Btu per BHP per hour.

Cooling Ponds

A natural pond is often used as a source of condensing water. The hot water should be discharged close to the surface at the shore line. Natural air movement over the surface of the water will cause evaporation and carry away heat. Because increased density due to the loss of heat causes the cooled water to sink to the bottom of the pond, the suction connection for intake water should be placed as far below the surface as possible, and at as great a distance from the discharge as practicable.