

TABLE 2. CONDENSER DESIGN DATA

Gas	MAXIMUM PRESSURE DESIGNED IN CONDENSER	GAS TEMPERATURE IN CONDENSER DEG F	LEAVING HOT WATER TEMPERATURE DEG F	
			Best Condenser Design	Average Condenser Design
Steam.....	28 in. vacuum.....	101.2	97	93
Steam.....	27 in. vacuum.....	115.1	110	105
Steam.....	26 in. vacuum.....	125.9	120	114
Ammonia.....	185 lb gage head pressure.....	96.0	92	88
Carbon dioxide.....	1030 lb gage head pressure.....	86.0	83	81
Methyl chloride.....	102 lb gage head pressure.....	100.0	96	92
Dichlorodi- fluoromethane	117 lb gage head pressure.....	100.0	96	93

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. 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 refrigeration 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 a refrigerating 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 8, 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 efficiency range of the type of atmospheric water cooling apparatus to be used. Efficiency of atmospheric water cooling apparatus is expressed as the percentage ratio of the actual cooling range to the possible cooling range. Since the wet-bulb temperature of the entering air is the lowest temperature to which the water could possibly be cooled this is:

$$\text{Percentage cooling efficiency of atmospheric water cooling equipment} = \frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}$$

Efficiencies of various types of atmospheric water cooling apparatus vary through wide limits, depending upon air velocity, concentration of water per square foot of area, and the type of equipment. The commercial range of efficiencies is 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.
Condenser turbine.....	950 to 980 Btu per pound of steam.
Steam jet refrigerating apparatus.....	1030 to 1150 Btu per pound of steam.
Diesel engine.....	2800 to 4500 Btu per horsepower.

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

TABLE 3. EFFICIENCY OF ATMOSPHERIC WATER COOLING EQUIPMENT

EQUIPMENT	COOLING EFFICIENCY—PER CENT		
	Minimum	Usual	Maximum
Spray Ponds.....	30	45 to 55	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