Table 5 .... Condenser Design Data

Gas	Desired Pressure in Condenser	Gas Temperature in Condenser, F	Leaving Hot Water Temperature, F	
			Best Condenser Design	Average Condenser Design
Steam	28 in. vac	101.2	97	93
Steam	27 in. vac	115.1	110	105
Steam	26 in. vac	125.4	120	114
Ammonia	185 psia*	96.0	92	88
Carbon dioxide	1030 psig*	86.0	83	80
Methyl chloride	102 psig*	100.0	96	92
Dichlorodifluoromethane	117 psig*	100.0	96	93
	126 psig*	105.0	100	97
	136 psig*	110.0	104	101

* Head pressure.

Condensing Systems

Each vapor has a pressure at which condensation occurs for any given temperature. This pressure increases with temperature up to the critical temperature of the vapor, above which no amount of increased pressure causes condensation. Condensation in a process occurs at constant temperature because a constant pressure will exist in the heat exchanger. The condensing pressure is usually the controlling factor in the process, although this is obtained by designing for the corresponding temperature. Table 5 lists typical condensing data for commonly used vapors.

A transfer of heat from the vapor to the circulating water requires a temperature differential between the two media. Since the vapor must condense at a constant temperature, the condensing temperature t_2 in Fig. 9 must be higher than the hot water temperature θ_1. It is for this reason that the condensing temperature is largely determined by the temperature of the warm water leaving the heat exchanger, and the entering cold-water temperature has only a small effect. The terminal temperature difference is controlled by the amount of heat transfer surface in the condenser. Economically sized condensers are usually designed to operate with 3 to 10 F deg temperature difference between the vapor and hot water.

Fluid Cooling

Different conditions exist when cooling a fluid such as milk, natural gas, or distillate. When this is done in a counter-flow heat exchanger, two terminal differences are involved. The fluid can be cooled to a temperature approaching that of the entering cold water, and the circulating water can be heated to a temperature approaching the hottest temperature of the fluid. A terminal temperature difference must exist at both the hot and cold end of the exchanger, but the temperature of the fluid leaving the exchanger can be colder than the hot water temperature.

Water Circulating Rate

The heat load on the system is determined by the process being served, and when this value is fixed, the circulating water rate determines the temperature rise in the heat exchanger. The cooling range in the cooling tower is obviously equal to the rise in the exchanger when the two are operating in equilibrium. The type of heat load and the characteristics of a cooling tower govern the selection of the optimum circulating rate. Increasing the circulating rate while holding the other operating conditions of a cooling tower constant, will reduce the cooling range. The cold-water temperature will increase, but the hot-water temperature will decrease to a greater extent. A high circulating rate and a short cooling range is advantageous when condensing vapors because the lower hot-water temperature has more effect on condensing temperature than the smaller increase in cold-water temperature. This advantage can be offset by greater power requirements for pumping, and possibly by the increased cost for larger piping.

Cooling a fluid presents a different problem. The cooling tower must deliver the cold water at a temperature that is below that of the cooled fluid, and this is more easily accomplished with a low water rate. The process may involve several heat loads, one of which requires a low temperature and the other can be removed at a much higher level. The size of a cooling tower is governed more by the cold-water temperature than by the cooling range. Once the cold-water temperature is fixed, it is desirable to increase the range as much as possible. A single cooling tower serving several heat loads in series is usually more economical than a number of individual towers.

A refrigerating system rejecting 250 Btu per ton imposes a load on the cooling tower of 30 (F deg) (gallons) per ton. The standard practice is to design the towers for 3, 4 or 5 gpm per ton, corresponding to cooling ranges of 10, 7½ and 6 F deg, respectively. Towers serving steam condensers usually operate with a range of 12 to 19 F deg. Oil refineries usually design for a cooling range of 25 F deg or more.

Heat Exchanger

The cooling tower operates in conjunction with the heat exchanger in transferring heat from the process to the atmosphere. Increasing the size of either one will reduce the required size of the other. The optimum conditions are determined by considering the initial and operating costs of several combinations of cooling towers and heat exchangers.

Process Characteristics

The type of process must be considered when selecting the design conditions of the system. This is particularly true of the temperature at which heat is removed from the process. Ammonia refrigerating systems are frequently designed to operate at 185 psig head pressure, corresponding to 96 F condensing temperature. This will require cooling water leaving the condenser at about 90 F. It will be impractical in most localities to select a cooling tower that will not exceed that temperature during brief periods. An increase in the temperature of the circulating water will raise the head pressure and increase the power requirement of the compressor. The increased head pressure also reduces the refrigerating capacity, but the system can be designed to perform acceptably at such times. A carbon dioxide refrigerating system represents a more serious situation. The critical temperature of the refrigerant is 88.4 F so the system will not function if that temperature is exceeded. The design conditions are also influenced by the process served by a refrigerating system. The maximum refrigerating load does not necessarily coincide with peak wet-bulb temperatures so a reduction in capacity may not be serious. That may not be true in the case of air conditioning if the maximum capacity is needed when wet-bulb temperature may be at its highest.

The efficiency, capacity and operating costs of a steam electric generating station all improve as the condensing temperature decreases. Designing for maximum performance at the most adverse conditions will result in a plant having excess capacity most of the time. The most adverse conditions occur during summer daylight hours, and peak power loads formerly came during winter evening hours. This gave the happy combination of peak demand during favorable operating conditions, and a low demand when the conditions were adverse. The power industry is profiting from the expansion of air conditioning, with the result that the peak load is beginning to coincide with the most adverse operating conditions. A generating plant can be designed to accept higher operating costs during such periods, but higher operating temperatures can reduce capacity below the required demand.

The capacity of an industrial operation such as an oil refinery will vary with the temperature of the cooling tower. Such a process may be so involved that it is not practicable to vary the output with fluctuation of the weather. These plants usually are compelled to operate at a constant rate that can be maintained from day to day. All processes using cooling water are faced with variations in water temperature. Higher temperatures increase operating costs and reduce capacity. These can be overcome by installing a larger plant or a larger cooling system, with the result that one portion of the equipment will have excess capacity during most of the year. The overall design must be based on (1) the minimum permissible capacity under the most adverse conditions and (2) an economic study showing the effect of changes in the size of the components on initial cost and operating expense.

Wet-Bulb Temperature

The wet-bulb temperature is beyond the control of the designer. It fluctuates during the day, and throughout the year. The annual averages should be studied and correlated with the characteristics of the process. Peak wet-bulb temperatures are usually of short duration so have little effect on annual operating costs, but the effect of these conditions in limiting capacity must be considered. The design wet-bulb temperature should be selected on the basis of the number of hours of the four summer months of June through September (or of the year) that it can be exceeded during an average year. A common practice in air conditioning is to select a wet-bulb temperature that will not be exceeded more than 5 percent of the time during the four summer months. Various groups are cooperating in compiling weather data for this type of study. A limited portion of the data are tabulated in Table 2, Chapter 13. More complete summer weather data, statistics, charts, maps, and technical analysis have been prepared by Albright.[13]

When the circulating rate and water temperatures have been selected, the size of the cooling tower will be increased by selecting a higher wet-bulb temperature. Selecting the design wet-bulb temperature makes it possible to consider various combinations of cooling towers and heat exchangers. The overall design will, of necessity, consider the worst possible conditions. This will correspond to the maximum wet-bulb temperature for the locality, and the highest permissible process temperature. All components should be selected with these conditions in mind, but testing will be easier if the same size equipment is designed and guaranteed around a lower wet-bulb temperature. Table 6 shows that an ammonia refrigerating system might be designed for a 78 F wet-bulb temperature. That temperature will prevail 12 hours a summer, and for brief periods of time. Designing the same equipment around a lower wet-bulb temperature greatly increases the amount of time that tests can be conducted at or near the design conditions.

Table 6 .... Alternate Design Conditions for Ammonia Refrigerating System, Showing Annual Hours the Design Wet-bulb Temperature is Attained

Wet-Bulb Temp. F	Hot Water Temp. F	Head Press. psi	Hours Attainable
78	92.0	185	12
75	88.5	174	161
70	86.0	166	750

SELECTION AND EVALUATION

The characteristics of the various types of water cooling equipment will influence the final selection. The requirements of the process, and physical conditions of the plant will impose further restrictions. These factors will determine the design conditions to be used and indicate the general type of equipment that will be most desirable. A wide variety of cooling towers will, nevertheless, be capable of meeting the design conditions. Competitive designs may be similar, but they will not be identical. The same design can be considered in various frame sizes, representing different plan areas or heights. An increase of frame size will enable a tower to meet the conditions with less total air flow, and will reduce horsepower requirements. As a general rule, an increase of frame size increases the initial cost but reduces operating costs. Optimum size is determined from an economic evaluation that will include such items as the following:

CAPITAL INVESTMENT

1. Total erected cost of cooling tower.
2. Basin, grillage, and value of the space occupied.
3. Cost of pump and prime mover.
4. Electric wiring to pump and fan motors.
5. Electrical controls and switch gear.
6. Cost of piping from tower inlet to point on tower nearest to plant.
7. Sump, screens, overflow, and make-up lines not furnished by manufacturer.
8. Shut-off and control valves not furnished by manufacturer.
9. Walkways, ladders, etc., providing access to tower.

OPERATING COSTS

1. Demand charge, if it enters into cost.
2. Energy costs for fans and pumps, based on 3000 to 7000 hr per yr.
3. Amortization of investment, over a period varying from 3 to 30 years.
4. Maintenance and repairs.

Other factors to consider are safety features, conformity to building codes, general design and rigidity of structures, availability of spare parts, experience and reliability of manufacturer, and operating flexibility for economical operation at varying loads or seasonal changes. Failure to perform in accordance with the guarantee usually proves costly to the