the advantage of locating the mechanical equipment away from the humid exhaust-air stream.

Air velocities usually vary from 300 to 400 fpm in counter-flow towers. The lower friction losses in cross-flow towers make it practicable to operate them at velocities up to 650 fpm. The water loading of counter-flow towers varies from 1.2 to 8 gpm per sq ft of plan area. The upper limit is largely determined by the blanketing effect of the distribution system. Since this blanketing is less pronounced in cross-flow towers, the loadings, based on plan area of the filling, can go up to 14 gpm per sq ft or higher.

Mechanical-draft towers require less plan area than ponds or atmospheric towers, because of the higher water loadings. This is quite important in roof installations where the available space may be limited. This feature can also be of great importance in large industrial applications where space may be at a premium. Reduction in plan area will usually result in a saving in piping costs.

Hyperbolic Towers

The natural-draft, or chimney type, tower depends on the difference in density of the heated air and outdoor air to produce a flow of air through the tower. The concrete hyperbolic tower has been common in Europe for many years and is now being offered in this country. Such towers are 50 to 260 ft in diameter at the base and from 100 to 340 ft high. Air enters around the periphery at the base and moves upward through filling, which is 10 to 20 ft high.

The performance for a given air flow is calculated on the same basis as a mechanical-draft tower. Variations in atmospheric conditions, however, do not have the same effect on the two types of towers. Fans tend to move a constant volume of air, so the mass flow rate varies with the density. An increase in air temperature results in a decrease in density and a corresponding decrease in air flow. A decrease in air temperature results in an increase in the mass flow rate. These variations are rather easily calculated for mechanical-draft towers.

The air leaving a mechanical-draft tower is frequently cooler than the dry-bulb temperature of the ambient air (although the wet-bulb temperature is higher) during warm weather. This is particularly true when the relative humidity is low. The outgoing air will then be heavier than the ambient air. Such a condition will not produce a draft in a hyperbolic tower, so it must operate at a lower air rate and a greater heat rise to obtain sufficient density difference to produce a draft. The density differential is quite small, so an increase in ambient air temperature and a corresponding decrease in density will cause the air flow to decrease until the heat rise is sufficient to produce a draft. Variations in atmospheric conditions have a greater effect on the performance of a natural-draft tower. If the two types of towers are designed for the same conditions, the natural-draft tower will deliver colder water when the atmospheric temperatures are lower than the design point, but the mechanical-draft tower will perform better at higher temperatures.

COMPARING TYPES OF EQUIPMENT

Each type of equipment has advantages and disadvantages that must be considered when selecting equipment for a particular situation.

Space Requirements

The spray pond occupies the greatest ground area and renders adjacent space unsuitable for most uses because of drift nuisance. The other types, in order of amount of space required, are spray towers, deck towers, and mechanical-draft towers, which occupy the least space.

Performance Limitations

Atmospheric water-cooling equipment has no control over cooling range. The cooling range is a function of the heat load applied by the process it serves, and the circulating water rate. The capacity of the equipment merely determines the temperature level at which the heat load is rejected to the atmosphere. This is usually measured as the approach of the cold water to the entering wet-bulb temperature.

Spray ponds provide the shortest time of contact between falling water and air, thus offering the poorest performance. They are usually designed for cooling ranges that do not exceed 10 or 15 F, and will obtain about a 10 F approach during hot summer weather. They are seldom practical for longer cooling ranges because the approach increases drastically under such conditions.

Spray towers have a larger number of small, closely spaced nozzles and a greater spray height. Finer break-up of the water and greater time of contact result in greater capability. The difference, however, is not great, and they are seldom designed for conditions that vary appreciably from those recommended for spray ponds.

Mechanical-draft and atmospheric deck towers can perform over a wide range of operating conditions. The capacity of both types is increased by (1) increasing effective height, (2) installing more closely spaced filling, or (3) increasing plan area to reduce water loading. The capacity of mechanical draft towers can be increased by an increase in air rate. These characteristics provide the designer with considerable flexibility, so the design conditions are limited by initial cost rather than by ability to perform.

Reliability

Natural-draft equipment occupies a long, narrow space that should be oriented broadside to the prevailing *summer* wind. Capacity varies appreciably with wind velocity and direction. Mechanical-draft equipment is less subject to the vagaries of the wind.

Natural-draft equipment is not subject to temporary loss of capacity due to failure of fans or drives.

Drift Loss

Spray ponds have the greatest drift loss, followed closely by spray towers and deck towers. This renders adjacent downwind areas unsuitable for many uses. It also creates a hazard to roadways, especially in cold weather. Mechanical-draft towers have the least drift loss. They are usually guaranteed not to exceed 0.2 percent of the circulating rate, and seldom lose that much. The nuisance from drift is therefore much less with mechanical-draft towers.

Fog

The enthalpy rise of the air passing through the equipment depends on the heat load and the amount of air absorbing the heat. The exhaust air is approximately saturated, and fogging can occur when this air blends with the surrounding atmosphere. The fog is produced whenever the mixing ratio produces a moisture content exceeding the saturation point. Fogging is unavoidable if the ambient air is saturated, and the range of fog-producing mixing ratios decreases with lower ambient relative humidities. A high air rate through the equipment results in a lower outgoing wet-bulb temperature, which also reduces the range of fog-producing ratios. The type of equipment has little to do with the amount of fogging, although a spray pond or low tower produces fog near the ground. Higher towers may create less hazard by allowing the fog to dissipate before reaching the ground.

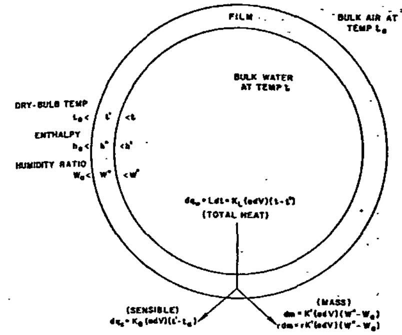

Fig. 5 .... Heat and Mass Transfer Relationships Between Water, Interfacial Film, and Air

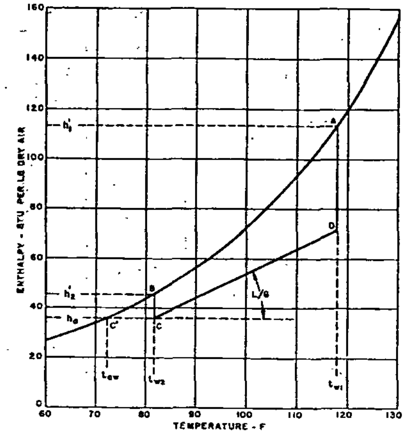

Fig. 6 .... Counterflow Cooling Diagram[3]

Recirculation

Wind blowing over and around a cooling tower, or other structure, creates a low pressure area on the leeward side. This creates down drafts that carry exhaust air to the ground where it can re-enter the equipment and increase the inlet wet-bulb temperature. Recirculation is worse in forced-draft towers because of the low exit air velocities. Equipment depending on the wind to provide air movement is not subject to recirculation.

COOLING TOWER THEORY[2]

Consider a cooling tower having 1 sq ft of plan area, a cooling volume, V, containing a sq ft of extended water surface per cu ft, a water rate, L, and an air rate, G, in lb per hr. The processes of mass and energy transfer are shown schematically in Fig. 5. Water at temperature t is surrounded by air at dry-bulb temperature t_a, having an enthalpy h_a and a humidity ratio W_a. The interface is assumed to be a film of saturated air having an intermediate temperature, t', and enthalpy, h'' and a humidity ratio, W''. Assuming a constant value of unity for the specific heat of water, the total energy transfer from the water to the interface is:

$$dq_w = L(dt) = K_L(adV)(t - t') \qquad (1)$$

where

q_w = rate of heat transfer, bulk water to interface, Btu per hour.
K_L = unit conductance, heat transfer, bulk water to interface, Btu per (hour) (square foot) (Fahrenheit degree).

The heat transfer from the interface to the air is:

$$dq_s = K_G(adV)(t' - t_a) \qquad (2)$$

where

q_s = rate of sensible heat transfer, interface to air stream, Btu per hour.
K_G = overall unit conductance, sensible heat transfer between interface and main air stream, Btu per (hour) (square foot) (Fahrenheit degree).

The diffusion of water vapor from the film to the air is:

$$dm = K'(adV)(W' - W_a) \qquad (3)$$

where

m = mass-transfer rate, interface to air stream, pounds per hour.
K' = unit conductance, mass transfer, interface to main air stream, pounds per (hour) (square foot) (pound per pound).

Considering the latent heat of evaporation a constant, r, the corresponding heat rate is:

$$r(dm) = dq_L = rK'(adV)(W' - W_a) \qquad (4)$$

The process will reach equilibrium when $t_a = t$ and the air becomes saturated with moisture at that temperature. Under adiabatic conditions, equilibrium is reached at the temperature of adiabatic saturation, or at the thermodynamic wet-bulb temperature of the air. This is the lowest attainable temperature in a cooling tower. The circulating water rapidly approaches this temperature when a tower operates without heat load. The process is the same when a heat load is applied, but the air increases in enthalpy as it moves through the tower so the equilibrium temperature increases progressively. The approach of the cooled water to the entering wet-bulb temperature is a function of the capability of the tower.

Merkel[3,4] utilized the Lewis[5] relationship (Equation 5) to combine the transfer of mass and energy into an overall coefficient based on enthalpy difference as the driving force.

$$\frac{K_G}{K'c_{pm}} \cong 1 \qquad (5)$$

where

c_{pm} = humid specific heat of moist air, Btu per (Fahrenheit degree) (pound of dry air).

This relationship also explains why the wet-bulb thermometer closely approximates the temperature of adiabatic saturation in an air-water vapor mixture. Simplification yields:

$$Ldt = Gdh = K'adV(h'' - h_a) \qquad (6)$$