Table 3 . . . . Sequence of Mechanical Integration Tower Performance

1	2	3	4	5	6	7	8
Water Temp. θ	Enthalpy of Film h''	Enthalpy of Air h_a	Enthalpy Diff. $h''-h_a$	$\frac{1}{h''-h_a}$	$\frac{\Delta\theta}{A(h''-h_a)}$	$\Sigma\frac{\Delta\theta}{h''-h_a}$	Cooling Range
84	48.22	38.61	9.61	0.1042			
					0.1042	0.1042	1
85	49.43	39.81	9.62	0.1041			
					0.1039	0.2081	2
86	50.66	41.01	9.65	0.1038			
					0.1034	0.3115	3
87	51.93	42.21	9.72	0.1030			
					0.1025	0.4140	4
88	53.23	43.41	9.82	0.1020			
					0.1013	0.5153	5
89	54.56	44.61	9.95	0.1006			
					0.1002	0.6155	6
90	55.93	45.81	10.12	0.0988			
					0.4640	1.0795	11
95	63.32	51.81	11.51	0.0868			
					0.3965	1.4760	16
100	71.73	57.81	13.92	0.0718			
					0.3220	1.7980	21
105	81.34	63.81	17.53	0.0570			
					0.2537	2.0517	26
110	92.34	69.81	22.53	0.0445			

h_1''. Air enters the tower at wet-bulb temperature t_2' having an enthalpy of h_2, which is plotted vertically below h_2''. Since the heat removed from the water equals the heat added to the air, the enthalpy increase per pound of dry air equals the temperature change of the water multiplied by the liquid/gas (L/G) ratio. The enthalpy of the air, therefore, increases along a straight line having a slope equalling L/G and terminating at h_1 which is vertically below h_1''. The driving force at any section of the tower is $h'' - h_a$, the vertical distance between the two operating curves.

Example 1: It is desired to cool 300 gpm (150,000 lb of water per hr) from 110 to 84 F with 3050 cfm (125,000 lb of dry air per hr) with an entering wet-bulb temperature of 75 F. Calculate the (NTU) showing the successive steps in tabular form.

Solution: The sequence of steps in the mechanical integration is shown in Table 3.

Water temperature is entered in Column 1 in even, but not necessarily equal, increments. One-degree increments are used from 84 to 90 F where the potential difference is small, and the effect of the reciprocal is large. Little accuracy is sacrificed by using 5-degree increments from 90 to 110 F where the potential difference is larger and its reciprocal smaller. Column 2 shows the enthalpy of saturated air corresponding to the temperatures in Column 1. Air enters the tower at a wet-bulb temperature of 75 F, having an enthalpy of 38.61 Btu per lb of dry air. This value is entered at the top of Column 3. The enthalpy of the air increases by $\Delta h = \frac{Ld\theta}{G}$. The L/G ratio is $\frac{150,000}{125,000} = 1.2$ so Δh equals 1.2 Btu from 84 to 90 F and 6 Btu from 90 to 110 F. These increments are added to the initial enthalpy to obtain the successive values in Column 3. The enthalpy potential difference in Column 4 is the difference between Columns 2 and 3. This driving force appears in the denominator of Equation 2 so the reciprocal of Column 4 is entered in Column 5. The (NTU) within each increment equals the temperature change multiplied by the average of the entering and leaving values from Column 5. The products are entered in Column 6. The summation of Column 6 in Column 7 gives the (NTU) for the cooling ranges in Column 8. Hence, the mechanical integration reduces the given conditions to the numerical value of 2.0517 *Transfer Units*.

Example 2: The cooling tower designed to handle the conditions in *Example 1* has a fill height of 8 ft and a plan area of 96 sq ft. Calculate the unit-volume and overall coefficients and explain their significance.

Solution: The (NTU) of 2.0517 as determined in *Example 1* signifies the tower must cool the water 2.0517 deg per Btu of mean driving force. This cooling is accomplished in 8 ft of fill height, so the unit-volume coefficient is $\frac{2.0517}{8} = 0.2565$. This means that each cubic foot of tower must cool the water 0.2565 degrees per Btu of mean driving force. The water loading is $\frac{150,000}{96} = 1563$ lb per (sq ft) (hr). The overall coefficient is $0.2565 \times 1563 = 400.91$. This signifies that the tower must transfer 400.91 per cu ft (hr) (Btu of mean driving force).

Column 7 of Table 3 shows how the (NTU) increases with cooling range. A plot of Columns 7 vs. 8 can be used to determine the (NTU) for intermediate cooling ranges. This curve can also be used to determine temperature distribution because equal increments of transfer units correspond to equal increments of tower height. The linear relationship between (NTU) and height provides the basis of the procedure used to analyze crossflow cooling towers.[12]

Since the water temperature varies both horizontally and vertically in a cross-flow tower a double integration is required. This is accomplished by using equal increments of *transfer units*, representing equal increments of distance, and calculating the corresponding temperatures. Counter-flow integration progresses from the bottom of the tower upward. The cross-flow integration starts at the top of the air inlet, and progresses downward and inward.

Required and Available Coefficients

Any set of performance conditions can be accomplished by an infinite number of cooling tower designs. Calculating the (NTU) serves to evaluate the degree-of-difficulty of the problem. The *unit-volume coefficient* is more specific, but it also applies to an infinite number of towers of a given height. The *overall coefficient* has the most significance when comparing towers of different designs that are being considered for a given duty. It evaluates the effect of variations in air rate, height, and plan area on the *required* coefficient. A higher *required* coefficient indicates that more effective filling is needed. The effectiveness of a filling is determined almost entirely by the density of the splash surface and wetted surface per cubic foot of filled volume.

These calculations evaluate the problem, not the tower. The *available* coefficient of a specific cooling tower is determined from its operating characteristics. The tower must be tested at various conditions and the data from each test reduced to a corresponding coefficient. A cooling tower coefficient is not constant, but varies with operating conditions. Its characteristic is established by correlating variations of the coefficient with corresponding operating conditions.

The characteristic of a cooling tower closely approximates the relationship expressed by the general equation:

$$Ka = cL^aG^b \tag{7}$$

The cooling tower manufacturer uses this procedure to determine the characteristic of each cooling tower design.

An examination of the steps taken in the solution of *Example 1* will show that various air rates could have been specified. Increasing the air rate would have reduced the degree-of-difficulty and resulted in a smaller (NTU) for a given set of conditions. Decreasing the air rate would have the opposite effect. Therefore, the *required* coefficient for a given set of conditions varies with the air rate. A tower will operate at a given set of conditions when the *required* coefficient equals the *available* coefficient as determined from Equation 7.

DESIGN CONDITIONS

The design conditions used in selecting a cooling tower are (1) the circulating water rate, (2) cooling range, (3) entering wet-bulb temperature, and (4) the approach of the cold-water to the wet-bulb temperature. These conditions are selected by considering the function of the cooling tower in the overall system. The system includes a process having a heat load that is to be transmitted to the circulating water through a heat exchanger. The cooling tower rejects the heat load to the atmosphere. Certain design conditions of the system affect the design of the cooling tower. They include (1) the heat load, (2) the temperature at which it is to be removed, and (3) the wet-bulb temperature of the atmosphere to which the heat is finally rejected. The cooling tower and heat exchanger are then selected to serve these conditions. The relationship of the cooling tower to the system is shown schematically in Fig. 9, with the process represented by two heat sources operating in series.

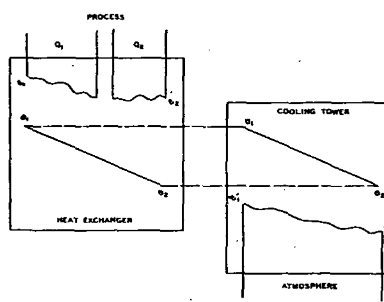

Fig. 9 . . . . Relation of Cooling Tower to Process

Heat Load

No definite sequence can be followed in selecting the design conditions because each item affects the choice of the others. The heat load comes closest to being a fixed quantity so can be considered first.

The amount of heat rejected by typical processes is shown in Table 4. The heat is rejected at a constant temperature when vapors are condensed, as illustrated by Q_2 in Fig. 9. A second type of heat load, represented by Q_1, consists of reducing the temperature of a fluid. Each type of heat load has specific requirements that affect overall design. The temperature at which the heat is removed from a process may be as important in the overall design as the heat load itself. As this temperature level decreases, the degree-of-difficulty increases at an accelerated rate, calling for larger and more expensive equipment. Higher temperatures usually result in higher operating costs, and loss of capacity in the process.

Table 4 . . . . Heat Rejection of Typical Processes

Equipment	Btu per min per ton	Btu per kw hr	Btu per bhp-hr
Refrigeration Compressors (reciprocating)	250		
Refrigeration Compressors (centrifugal)	300		
Refrigeration Absorption System	550		
Steam Jet Refrigerating System	550		
Steam Electric Power Plant:			
500 kw		11,210	
1000 kw		10,750	
5000 kw		8,150	
7500 kw		7,700	
10,000 kw		7,020	
Diesel Engine Jacket & Lube Oil:			
Four-cycle, Supercharged			2600
Four-cycle, Non-supercharged			3000
Two-cycle, Crank-case Compressor			2000
Two-cycle, Pump Scavenging (large unit)			2500
Two-cycle, Pump Scavenging (high speed)			2200
Natural Gas Engine:			
Four-cycle			4500
Two-cycle			4000