At the same air mass velocity, varying performance can be obtained depending upon the turbulence of the air flow into the coil, and upon the uniformity of distribution of air over the coil face. The latter is very important in obtaining reliable test ratings, and in realizing rated performance in actual installations. The air resistance through the coils will assist in distributing the air properly, but where the inlet duct connections are brought in at sharp angles to the coil face, the effect is frequently bad and there may even be reverse air currents through a portion of the coils. This reduces the capacity, but can be avoided by proper layout or by the use of vanes or baffles.

Generalized heat-transfer information on bare pipe coils has been developed and verified through many tests. In the case of finned coils, the heat transfer from the cooling or heating medium to the air stream is dependent on so many factors that reliable rating and performance information for any design of coil must be based upon actual tests of the specific coil. Mathematical comparisons of different designs of coils on a square foot of surface and face area basis may be misleading. The selection of finned coils should be made from curves or tables of coil performance prepared from a series of adequate and reliable tests. There are cases in which the engineer must extend available data or design for a single unique installation. For such purposes the following coil calculations will be useful.

PERFORMANCE OF DRY COOLING COILS

The performance of dry cooling coils depends in general upon:

1. The overall coefficient of sensible heat transfer from the fluid within the coil to the air it cools.
2. The mean temperature difference between the fluid within the coil and the air flowing over the coil.
3. The physical dimensions of the coil.

Thus, for any one definite operating condition, the cooling capacity of a given coil is expressed by the following basic formula:

$$q_t = U \times (\Delta t_m) \times A \times N \qquad (1)$$

where

q_t = total heat transfer of the coil, Btu per (hour) (square foot of coil face area).
U = overall coefficient of sensible heat transfer, Btu per (hour) (square foot of external coil surface) (Fahrenheit degree temperature difference between the fluid within the coil and the air flowing over the coil).
Δt_m = mean temperature difference, Fahrenheit degrees, between the fluid within the coil and the air passing over it. (This is commonly taken as the logarithmic mean temperature difference.)
A = external surface area of the given coil, square feet per (square foot of coil face area) (row of coil depth).
N = number of rows of coil depth.

Overall Coefficient of Sensible Heat Transfer

While the overall coefficients of sensible heat transfer for bare pipe coils have been defined by numerous tests within close limits, the verification of a bare coil design by a series of tests is an accepted commercial practice. The overall coefficients of sensible heat transfer for finned coils should always be obtained from tests. The data from a series of tests may be used for purposes of interpolating and extending coil data beyond test range but such extrapolated data should later be verified by test.

Considering any coil, whether of bare pipe or of finned type, the overall heat-transfer coefficient for a given size and design of coil can always be considered as a combined effect of three individual heat-transfer coefficients, namely:

1. The film coefficient of sensible heat transfer between air and the external surface of the coil, usually given in Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference).
2. The conductivity of the coil material—tube wall, fins, ribs, etc., usually given in Btu per (hour) (square foot of surface) (Fahrenheit degree per inch).
3. The film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, usually given in Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature difference).

These three individual coefficients acting in series result in an overall coefficient of heat transfer in accordance with the basic laws given in Chapters 5 and 10. For a *bare-pipe* coil, the overall coefficient of heat transfer for cooling (without dehumidification) can be expressed by a simplified basic formula as follows:

$$U = \frac{1}{\frac{R}{f_i} + \frac{L}{k} + \frac{1}{f_o}} \qquad (2)$$

where

U = overall coefficient of sensible heat transfer, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and fluid within the coil).
f_i = film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature difference between that surface and the average fluid temperature).
f_o = film coefficient of sensible heat transfer between air and the external surface of the coil, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between the mass of air and the external surface).
k = conductivity of material from which the bare pipe is constructed, Btu per (hour) (square foot) (Fahrenheit degree per inch thickness).
L = thickness of tube wall inches.
R = ratio between external and internal surface of the bare tube, usually varying from 1.03 to 1.15 for the tube used in typical heating or cooling coils. This ratio R is inserted in the formula in order to place internal fluid coefficient of heat transfer on the basis of external surface.

Frequently, when pipe or tube walls are thin and of material having high conductivity (as is the case in construction of typical heating and cooling coils) the term L/k in Equation 2 becomes negligible and is generally disregarded. (The effect of the term L/k in typical bare-pipe cooling coils seldom exceeds 1 to 2 percent of the overall coefficient.) Thus, in its simplest form, for bare pipe:

$$U = \frac{1}{\frac{R}{f_i} + \frac{1}{f_o}} \qquad (3)$$

For finned coils the equation for the overall coefficient of heat transfer can be conveniently written:

$$U = \frac{1}{\frac{R}{f_i} + \frac{1}{\eta f_o}} \qquad (4)$$

in which the term η, called the *fin effectiveness*, is introduced to allow for the resistance to heat flow encountered in the fins.

The term R, in this case, is the ratio of *total* external surface to internal surface. For typical designs of finned coils for heating or cooling, this ratio varies from 10 to 30. Term R is again introduced to place the internal surface coefficient of heat transfer on a basis of external surface.

The performances of all dry cooling coils are influenced by these same factors. But, when cooling coils operate wet or act as dehumidifying coils, the performance cannot be predicted on the basis of overall sensible heat coefficients since the effect of air-side moisture transport (or latent heat removal) must be included (see Chapter 5).

PERFORMANCE OF DEHUMIDIFYING COILS

When the dew point of the air leaving a cooling coil is lower than the dew point of the air entering the coil, some *moisture* removal has been accomplished. A coil that normally accomplishes (or is designed to accomplish) moisture removal in addition to sensible-heat cooling is termed a dehumidifying coil.

In most air-conditioning processes, the air may be considered as a mixture of water vapor and the dry components. Both dry components and the water vapor enter an air-conditioning coil at the same dry-bulb temperature; both the dry components and the water vapor lose sensible heat during the contact with the first portion of the cooling coil in the same manner as in a dry cooling coil. As the dry-bulb temperature of the mixture approaches the dew point of the water-vapor components, moisture removal starts.

The psychrometric path of air through a cooling coil is generally assumed to follow a path similar to that shown in Fig. 12. When the dry-bulb temperature of the air mixture in the coil falls below the entering dew-point temperature, moisture removal proceeds. The indications are that the lower the leaving dry-bulb temperature below the entering dew-point temperature, the less will be the difference between the leaving dry-bulb temperature and the leaving dew-point temperature.

The first portion of a cooling coil (in the direction of air flow) may function in the same manner as a dry cooling coil. Where the moisture removal starts, the cooling surfaces also continue the removal of sensible heat, thereby carrying the load due to both. As saturation is approached in the cooling coil, each degree of sensible cooling is approximately matched by a corresponding degree of dew-point decrease. However, while the sensible heat removal from the dry air remains approximately constant per degree change, the amount of latent heat removal per degree of dew-point change varies considerably because moisture content varies widely at different temperatures.

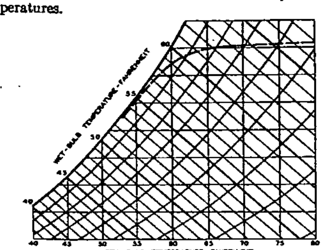

Fig. 12 . . . . Performance of Dehumidifying Coil

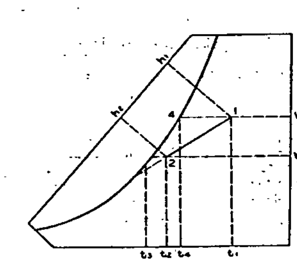

Fig. 13 . . . . Psychrometric Performance of Cooling And Dehumidifying Coil

For example, the following tabulation compares the amount of moisture removal involved in a reduction of one degree of dew point from 60 to 59 F with the removal from 50 to 49 F:

Dew Point	$W_s \times 10^3$ lb/(lb)	*Dew Point*	$W_s \times 10^3$ lb/(lb)
60	11.080	50	7.658
59	10.690	49	7.374
Difference	0.390	Difference	0.284

The above values are given in Table 2, Chapter 3.

When cooling coils act as dehumidifying coils, the performance can be predicted accurately only from tests at a sufficient number of points to establish the definite performance characteristics of the coil under varying conditions of loading and of entering air. Dehumidifying coils employing volatile refrigerants are generally rated in conjunction with specific refrigerant distributing and flow control equipment. The combination of the coil with its refrigerant control equipment (such as distributor and expansion valve, and capillary tube or float valve) must be tested at both the higher and lower capacities of its rated range. The lower capacities impose a test on the distributor to provide equal distribution and on the control to modulate without *hunting* at the lower capacities. The higher capacities result in a greater pressure drop through the coil system and a test of the maximum feeding capacity of the flow control device at various *head pressures*.

Most coil manufacturers have their own methods of producing performance rating tables from a suitable number of coil-performance tests. A method of testing coils is given in *ASHRAE Standard* 33–64, *Method of Testing for Rating Forced-Circulation Air-Cooling and Air-Heating Coils*. A basic method of rating to provide a fundamental means for establishing thermal performance of dehumidifying coils by extension of test data, as determined from laboratory tests on prototypes, to other operating conditions, coil sizes, and row depths of a particular surface design and arrangement, is given in *Air-Conditioning and Refrigeration Institute Standard* 410–64 *for Forced-Circulation Air-Cooling and Air-Heating Coils.*

DETERMINING REFRIGERATION LOAD

The following determination of the refrigeration load shows a division of the true *sensible* and *latent heat* loss of the air, which is accurate within the limitations of the data. This division will not correspond to load determination obtained from approximate factors or constants.