cooling and dehumidification, a special measure of the propelling force is used as described later. Logarithmic differences are generally employed in practice although there are special flow relationships used, such as cross-flow, where they do not strictly apply. With volatile refrigerants there is often an appreciable pressure drop and corresponding change in evaporating temperature through the refrigerant circuit. The problem is further complicated by the fact that the refrigerant is evaporating in part of the circuit and superheating in the remainder. In spite of this, heat transfers and ratings for coils using volatile refrigerants are usually based in practice on a refrigerant temperature corresponding to the pressure at the coil outlet.

The design and surface arrangement of the coil includes such items as materials, type, thickness, height and spacing of the fins, and the ratio of this surface to that of the tube, the use of the staggered or in-line tube arrangement, and provisions to increase the air turbulence such as the use of corrugated as against flat fins. Staggered tubes increase the total heat transfer as against the in-line arrangement and corrugated fins are more effective than flat. Of especial importance is the bond between fin and tube.

The velocity of the air usually considered is the coil face velocity. This bears a varied relation to the actual velocity over the surface, depending upon the individual coil design. As long as a fixed design of coil is under consideration face velocities may be used, but they may be unsatisfactory in comparing different designs, as it is the actual surface velocity that is significant. The air volume is often based on standard air at 70 F and a barometric pressure of 29.92 in. Hg. The use of air volume in coil rating information may be misleading. The significant value is mass velocity in pounds per minute and not cubic feet per minute, because for a fixed volume the corresponding weight may vary widely, depending upon the air density, temperature and barometric pressure under consideration.

At the same mass air 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 practical installations. The resistance through the coils will assist in properly distributing the air, 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 the coils. This reduces the capacity, but can be largely avoided by proper layout or by the use of directing baffles.

The heat transfer depends also upon the velocity of the medium in the tubes and upon its character, whether flowing water, condensing steam or evaporating volatile refrigerant. In the latter case, the effect is complex because of the combination of evaporation and superheating, and because of the influence of pressure drop in the refrigerant circuit. Heat transfer rates expressed as Btu per square foot of internal surface per degree logarithmic mean effective temperature difference between the fluid and tube wall are, for example, about 150 to 300 for evaporating dichlorodifluoromethane, about 350 to 1200 for water at 2 and 6 fps and about





1200 for condensing steam. The influence of the medium in the tubes on the overall heat transfer rate is, therefore, apparent.

Because of these variables, reliable rating and performance information for any design of coil must be based on actual tests on that coil under the expected conditions of operation. A comparison between the performance of two designs, unless based on such tests on each, may lead to entirely erroneous conclusions.

## Heating and Dry Cooling Coils

To find the surface requirements for heating or dry cooling coils, the heat transfer coefficient $U$ and the dry-bulb mean temperature difference must be known. It has been found that $U$ can be expressed as an exponential function of the mass air velocity. If $U_1$ is known at a given velocity $V_1$, its value at another velocity $V_2$, can then be found by the Equation 1.

$$U_1/U_2 = (V_1/V_2)^n \tag{1}$$

This relationship can also be represented on logarithmic paper as a straight line of slope $n$. The value $n$ must be determined by test as it is dependent upon the coil design. For coils in common use, $n$ ranges from about 0.4 to 0.8. For a given design, $n$ will depend upon the coefficient of heat transfer from the fluid in the tubes to the tube wall, so that $n$ will be higher as the water velocity in the tube increases, and will be higher for condensing steam than for water at low velocities. A coil that is several rows deep will have a higher $n$ than one of the same design with fewer rows. For heat transfer information as well as values of $n$ for typical coils, see Table 1.

To avoid the labor of coil selection by use of heat transfer coefficients and mean temperature differences, the ratings of heating and dry cooling coils are frequently set up in tables from which, for known conditions, a coil can be selected directly.

## Dehumidifying Coils

When air passes through a cooling coil, the temperature of which is lower than the dew-point of the air, there is a removal of both sensible and latent heat. The sensible heat transfer process is exactly the same as in heating and non-dehumidifying cooling coils. The moisture passes from the air to the cold surface by diffusion.

It is evident that the coil may be wet throughout or, because of temperature gradient through fins or temperature range in the refrigerant, may be partially dry and partially wet. In this case, part of the coil acts as dry and part as dehumidifying surface. Various approximations are used for this condition as the percentages of wet and dry surface and the temperatures applying to each are difficult to ascertain. One approximation is to assume that for ratios of total to sensible heat of 1.10 or more the coil is to be treated entirely as wet, and for ratios less than 1.10 entirely dry.

Although much research has been already conducted and more is in progress, there is no general agreement as to the most satisfactory and convenient manner in which to rate dehumidifying coils. A large number of methods are now in use, most of them combinations of theory and