

Table 27 . . . Rates of Heat Gain from Occupants of Conditioned Spaces*

Degree of Activity	Typical Application	Total Heat Adults, Male Btu/Hr	Total Heat Adjusted ^b Btu/Hr	Sensible Heat Btu/Hr	Latent Heat Btu/Hr
Seated at rest	Theater—Matinee Theater—Evening	390 390	330 350	180 195	150 155
Seated, very light work	Offices, hotels, apartments	450	400	195	205
Moderately active office work	Offices, hotels, apartments	475	450	200	250
Standing, light work; or walking slowly	Department store, retail store, dime store	550	450	200	250
Walking; seated Standing; walking slowly	Drugstore, Bank	550	500	200	300
Sedentary work	Restaurant*	490	550	220	330
Light bench work	Factory	800	750	220	530
Moderate dancing	Dance hall	900	850	245	605
Walking 3 mph; moderately heavy work	Factory	1000	1000	300	700
Bowling ^c Heavy work	Bowling alley Factory	1500 1500	1450 1450	465 465	985 985

* Note: Tabulated values are based on 80 F room dry-bulb temperature. For 78 F room dry-bulb, the total heat remains the same, but the sensible heat values should be increased by approximately 10 percent, and the latent heat values decreased accordingly.

^b Adjusted total heat gain is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is 85 percent of that for an adult male, and that the gain from a child is 75 percent of that for an adult male.

^c Adjusted total heat value for sedentary work, restaurant, includes 90 Btu per hour for food per individual (30 Btu sensible and 30 Btu latent).

^d For bowling figure one person per alley actually bowling, and all others as sitting (400 Btu per hour) or standing (350 Btu per hour).

up a lower limit to the heat gain to a room from the appliance when in operation.

Experienced judgment must be used in the application of data given in Table 28. Consideration must be given to the heat contributed by appliances which are in use at the time of peak load. The quantity of heat will depend upon whether products of combustion are vented to a flue, whether they escape into the space to be conditioned, or whether appliances are hooded to allow part of the heat to escape through a stack. There are no generally accepted data available on the effects of venting and shielding heating appliances, but it is believed that when they are properly hooded with a positive fan-exhaust system through the hood, 50 percent of the heat will be carried away and 50 percent dissipated in the space to be conditioned. In the case of gas-burning appliances where the heat of combustion adds considerably to the total heat output, it is believed that a well-designed hood will remove more than 50 percent of the heat generated at the appliance. The same effectiveness of the hood should be figured for both latent and sensible heat.

For each meal served the heat transferred to the dining space is approximately 50 Btu per hr, of which 75 percent is sensible and 25 percent latent.

Heat gains from cooking appliances, motors, lights, and people are not considered in cooling load calculations if the appliances are located in a separate kitchen that is not part of the conditioned area.

LOAD FROM MOISTURE TRANSFER THROUGH PERMEABLE BUILDING MATERIALS

The diffusion of moisture through all common building materials is a natural phenomenon which is always present to a greater or lesser degree.

The permeability and permeance values for various building materials are given in Table 2 of Chapter 10, together with an explanation of moisture transmission through these materials.

In the usual comfort air-conditioning application, it is common practice to neglect moisture transfer through walls, because the actual rate is quite small and the corresponding latent-heat load is hardly significant. Vapor barriers are frequently employed in modern construction for the purpose of keeping moisture transfer to a minimum, and reducing the deteriorating and insulation-destroying effects of moisture.

Industrial jobs, on the other hand, frequently call for a low moisture content to be maintained in a conditioned space. Here the matter of moisture transfer cannot be neglected; indeed, it is quite possible to have the latent-heat load accompanying this transfer be of greater magnitude than any other latent-heat load. The equation for computing this load is:

$$\left(\frac{q_m}{A}\right) = \frac{M}{7000} \times \left(\frac{\text{Vapor-Pressure}}{\text{Difference, In. Hg.}}\right) \times 1076, \text{ Btu/sq ft (17)}$$

where

M = permeance of the specimen in perms, or grains per (square foot) (hour) (inch of mercury vapor-pressure difference).

7000 = grains per pound.

The factor 1076 is defined in list of symbols at Equation 9. (Sensible cooling of the water vapor is included in the factor 1076.)

The only means of preventing moisture transfer is to use a vapor proof wall, or to apply a special lining, which is

Cooling Load

vapor proof. All openings in moisture proof construction must be equipped with special gaskets to prevent entrance of moisture.

When moisture transfer contributes an appreciable part of the latent-heat load, it is recommended that estimates should be made intentionally liberal in order to avoid later difficulties with insufficient dehumidifying capacity. Storage spaces, for example, would require sufficient dehumidifying capacity to handle the moisture brought in with goods to be stored, in addition to moisture leaking in subsequently.

MISCELLANEOUS HEAT LOADS

This designation is intended to cover the various small heat gains from exposed piping, ducts, work done by circulating fan, and unforeseen contingencies. Where sufficient data are available, these various heat gains may be estimated individually. In the majority of cases, however, common practice is to lump these factors together and combine them with a safety factor according to the experience and judgment of the estimator. On this basis, a small safety factor is added to the calculated cooling load to compensate for miscellaneous effects. No rules can be given for this procedure, as experience in air conditioning is indispensable for application of suitable safety factors.

APPARATUS DEW POINT AND REQUIRED AIR QUANTITY THROUGH CONDITIONING EQUIPMENT

In ordinary practice, with commercial apparatus, complete saturation of the air is seldom obtained. Four-row finned cooling coils contact approximately 80 percent of the air, whereas six-row finned coils contact approximately 95 percent of the air. In spray-type dehumidifiers of good design the air leaves the dehumidifier at 1 to 2 deg higher wet-bulb temperature than the spray water leaving the dehumidifier, and the difference between the dry-bulb and wet-bulb temperatures leaving the dehumidifier may be as low as 1 deg. A spray-type dehumidifier having sufficient length of spray chamber and density of spray, together with proper arrangement of nozzles, may approach saturation very closely.

As explained in Chapter 3, and shown in Fig. 5, the slope of the line on the psychrometric chart connecting the room condition with the apparatus dew point on the saturation line, determines the ratio of sensible heat absorbing capacity to the moisture absorbing capacity of the supply air. Therefore the room condition can be maintained as long as the supply-air temperature lies on this line, but a greater volume of supply air must be used to satisfy the room load if the cooling coil does not contact 100 percent of the air. For a given room load, the same apparatus dew point will be required whether the cooling appliance contacts all the air or only part of the air.

From the point of view of satisfying the given cooling load requirements, the air passing through the apparatus without being cooled below the dew-point temperature produces two effects:

1. The air quantity which must be passed through the dehumidifier must be increased. Thus, if 20 percent of the air passing is not contacted, then $(20 \div 80) \times 100 = 25$ percent more air must be used than would be necessary if all of it were contacted.
2. Passing untreated air may change the room cooling load, which in turn may change the enthalpy-humidity difference ratio (sometimes called the sensible-heat factor). If return air only is passed through the dehumidifier or if room air only is bypassed, the room load will not change, but if some outdoor

air is passed through, the room sensible-heat gain and room latent heat gain will be changed due to the addition of untreated outdoor air which changes the enthalpy-humidity difference ratio. When a load calculation is made, it is necessary to know the percentage of air treated in the dehumidifier, and calculation must be made accordingly.

If the ventilation air is drawn through the dehumidifier before it goes into the room, only that portion of the air not saturated must be included in the room load for the purpose of determining the apparatus dew point and supply-air quantity. It should be noted when evaluating the load added by untreated outdoor air that the temperature difference between room air and outdoor air, and the moisture content difference between room air and outdoor air, should be used, rather than the difference between outdoor air and apparatus dew point, since the rise from the apparatus dew point to room condition is charged against the dehumidifier as the cooling and dehumidifying load.

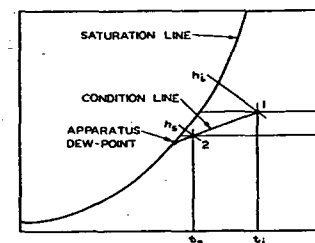


Fig. 5 . . . Apparatus Dew-Point Shown on ASHAE PSYCHROMETRIC CHART

The procedure for determining the required air quantity is based upon the thermodynamic principles of Chapter 3 and the use of the ASHAE PSYCHROMETRIC CHART. Readers are advised to review these principles, paying particular attention to the illustrative examples.

Calculation of the cooling load for a conditioned space is equivalent to making, for the space, a heat balance in which all heat, moisture, and infiltration are treated as directly entering the space. As explained in the section, Load from Outdoor Air, Ventilation, and Infiltration, the outdoor-air load normally does not become a part of the space load, because heat and moisture are removed in the air conditioner before this air gets into the conditioned space. The desired conditions are maintained by considering a certain quantity of air to be withdrawn from the space, passed through the conditioning equipment, and returned to the space with such a temperature and humidity ratio that its net effect will be to counterbalance or remove the given entering amounts of heat and water vapor. This quantity of indoor air, which is considered to be circulated in this manner, is called the required air quantity and its determination is normally part of every cooling-load estimate. The procedure is as follows:

1. Determine the total sensible and latent heat loads in Btu per hour for the space.
2. Compute the quantity called the enthalpy-humidity difference ratio (also referred to as heat-moisture ratio) of