temperature for a period of time prior to the advent of the peak load, when the heat gain begins to increase to peak conditions, some of the increase is used in raising the temperature of the furniture, fixtures, etc., to the design conditions and the cooling load can be reduced accordingly. However, unless very accurate data with regard to the mass, surface, specific heat, etc., of the items within the space are available, due caution must be used in discounting the cooling load for this storage effect. In the absence of reliable data it is often a matter of experience rather than calculation.

Where air conditioning supply and return ducts pass through unconditioned spaces there will be a transfer of heat from these spaces to the air in the ducts, even though these ducts are well insulated. An allowance

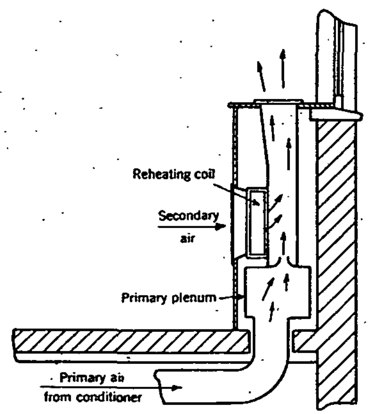

FIG. 10. INDUCTION UNIT (LOW PRESSURE TYPE)

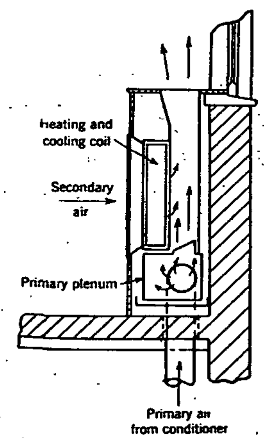

FIG. 11. INDUCTION UNIT (HIGH PRESSURE TYPE)

should be made for this heat gain and included in the heat estimate so that air can be supplied at a temperature low enough to offset the rise caused by this heat gain (see Chapter 43). There will also be some heat gain to the air in ducts passing through conditioned spaces, but since a cooling effect is produced in the space through which the duct passes, this is not a loss and usually can be compensated for by adjustment of air quantities between the various spaces.

Heating Load

Methods of calculating the heating load are shown in Chapter 6. Many of the factors outlined previously under Cooling Load, such as zoning, non-simultaneous peaks, and diversity, apply in the reverse manner due to the heating requirement instead of the cooling requirement. However, these factors enter into the heating load picture from a standpoint of control of inside conditions, overall performance and economy of operation more than from a capacity of equipment standpoint.

Where heating is concerned it is not only necessary to heat a building or space to its design conditions when there is but the merest fraction of normal occupancy, practically no lights, internal heat, or solar radiation, but it is also necessary to provide capacity to heat the building quickly after a shut-down such as when a sudden cold snap follows relatively warm weather, or after a week-end or holiday. However, in normal operation during week-ends and holidays, buildings are usually kept at a *holding* temperature to prevent the freezing of services and conserve fuel. In many cases it requires less fuel to keep a building or space at a temperature of 50 to 65 F for some time than to shut the system down and then bring the temperature up again.

Apparatus Dew-Point

The method of locating the condition line for a given air conditioning problem has been explained in Chapter 1, Examples 19 and 20. Briefly, the method consists in estimating the net energy gain (or loss) per hour and the net moisture gain (or loss) per hour from data on location, exposure, construction, appliances, occupants, ventilation requirements, inside and outside design conditions. In computing the quantities of energy and moisture introduced and displaced by the ventilating air, only that portion of the ventilating air admitted directly to the conditioned space is considered. With this understanding, the ratio of the net energy gain (or loss) to the net moisture gain (or loss) determines the slope of the condition line through the state point of the inside air on the Mollier Chart.

The condition line may or may not cross the saturation curve. If it does, the intersection is called the *apparatus dew-point*, with application to summer cooling in mind. Thus, if the air conditioning apparatus can be set to take inside air, process it, and return it to the conditioned space completely saturated at the apparatus dew-point, the cooling load requirements can be exactly met both as to removal of energy and simultaneous removal of moisture.

In actual practice with commercial apparatus, it is rarely possible to obtain complete saturation and there may be several degrees difference between the dry-bulb and wet-bulb temperatures of the air returned to the conditioned space. This causes no difficulty. In fact, the only special significance of the apparatus dew-point is that, when it exists, it provides a convenient control point at which to regulate the operation of the apparatus, provided complete saturation is attainable.

In order to illustrate the effect of incomplete saturation in the conditioning apparatus, consider the cooling load problem of Chapter 1, Example 19. In this problem, 114,600 Btu of energy and 15.92 lb of moisture per hour are to be removed simultaneously. The slope of the condition line is determined by the ratio $q = 114{,}600 \div 15.92 = 7205$ Btu per pound of water and the apparatus dew-point is 58.02 F, as shown in Fig. 12. (Point *B*).

But suppose that the saturation efficiency of the apparatus is only 95 per cent, by which is meant that the air delivered by the apparatus is only 95 per cent saturated. Then the temperature at which the condition line crosses the 95 per cent saturation curve is the proper temperature at which to regulate the apparatus. This temperature is easily found to be 59.6 F dry-bulb temperature (Point *A* in Fig. 12). Of course, a somewhat larger quantity of air will have to be recirculated, namely $114{,}600 \div (31.41 - 25.51) = 19{,}400$ lb of dry air per hour, where the enthalpy of the air at 95 per cent saturation on the condition line is 25.51 Btu per pound of dry air and $h = 31.41$ for the