# Fit The Air-Conditioning System To The Load

**This concluding article shows how to solve design problems. Part I gave reasons for studying building requirements and making a complete physical survey before designing a system**

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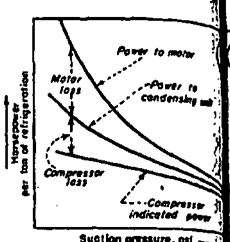
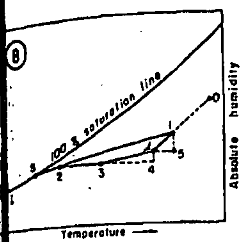
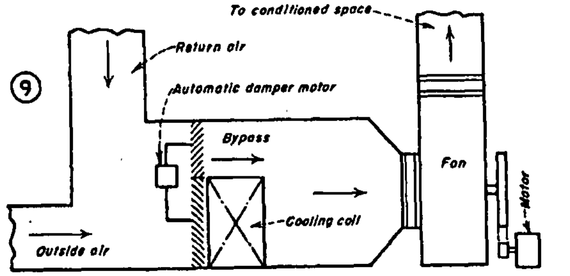



ON A GIVEN LOAD, lowering the suction pressure will increase the horsepower




HIGH LATENT-HEAT LOAD is indicated if line 1-2 fails to touch saturation curve




CHANGES IN DESIGN of the surface cooler and bypassing do not increase the dehumidification with respect to the sensible cooling if the air conditions are fixed

▶THE SIMPLE EXPEDIENT of installing less cooling surface and operating it at lower temperature sometimes materially reduces the first cost of equipment. Because the combined condensing unit and evaporator fit the load at a lower suction pressure, the horsepower requirements per ton of refrigeration increase, Fig. 7. But this increased power use is offset by the attendant improvement in comfort conditions at the lower indoor humidities.

**Change in Humidity.** Conversely, a slight increase in design-indoor humidity may use greater coil surfaces and, therefore, a more expensive evaporator assembly, but lower power-consuming refrigeration equipment. Of course, less comfortable conditions of humidity prevail. Since condensing-unit capacity varies in steps according to motor size, a change in design-indoor humidity may easily determine economies in first cost of the condensing unit. Variations in design-indoor dry-bulb temperature affect equipment selection similarly but to a different degree.

Economic selection of equipment thus requires a balance between operating expense and fixed charges. Although total cost may be reduced by slight changes in design-indoor conditions, these are beyond the control of the application engineer in industrial and process applications, and often in comfort installations, because conditions are either defined by the process or set by contract. He must confine his efforts, therefore, to variations in air flow and characteristics of heat-transfer surface in arriving at the most economical arrangements.

**Conditions That Need Reheat.** The straight line joining points 1 and 2 in Fig. 8 may fail to intersect the saturation line (Part I, Jan 1948, p 80) because of high latent-heat load requirements. This indicates that the ratio of latent- to sensible-heat loads (curve slope) is so great as to require an additional air-treating device. Remember that surface-type equipment has definite limitations with respect to dehumidification. Once entering-air conditions, point 1, are set, the maximum ratio of latent to sensible capacity attainable with that particular coil is fixed by the steepest line that can be drawn through initial point 1, and yet terminate on the saturation curve. This is a tangent to the 100% saturation curve. Thus, with the entering-air point fixed, changes in surface-cooler design and modifications to air flow, such as bypassing, Fig. 9, will not further increase the amount of dehumidification with respect to sensible cooling. This holds true for extended surface coolers of all types.

**Reheat or Sorbent.** Only alternative is to supplement the surface cooler by some device that alters the temperature or humidity of the air. One method is to reheat the air, Fig. 10, after it has been cooled and dehumidified. Another way is to dehumidify the air chemically, Fig. 11, before passing it to the surface cooler. Reheat is commonly used when the ratio of latent to sensible heat is greater than can be handled simultaneously in a single process. Air entering the conditioner may be cooled by a surface maintained at some temperature below the initial dewpoint corresponding to 2 on Fig. 8, to some point where the correct amount of moisture has been removed. In removing the latent heat and sweating out moisture from the air stream, more than the required amount of sensible heat is removed. This results in a low dry-bulb temperature.

**Watch Dry-Bulb Temperature.** If air is introduced to the conditioned space without further treatment, dry-bulb temperature will be below the design value. With normal ceiling height, architectural treatment and supply-grille setting, dry-bulb temperature of entering air should not be more than 15 F below design value. To avoid too low a temperature or too great a diffusion, the air stream leaving the cooling coil must be warmed by a reheat coil in the conditioner or in the supply duct.

If point 3, Fig. 8, is the fixed condition of air supply to the conditioned space, length of line 2 to 3 represents reheat requirements. Ideal setup is to condition the air down to the introduction point, but this is seldom done in practice.

**Problems on Selection.** The following example illustrates the principles involved in selecting equipment for cooling and dehumidifying. For comfort cooling, the design conditions could apply to areas in and around New York City and Tampa, Florida.

### DESIGN CONDITIONS

Outdoors, 95 F dry bulb, and 78 F wet bulb
Indoors, 80 F dry bulb, and 50% relative humidity
Outdoor air supply, 1340 cfm
Total sensible heat gain, 87,000 Btu per hr
Total latent-heat gain, 48,300 Btu per hr
Ratio, latent-to-sensible-heat gains 0.56
Total air circulation, 4000 cfm

The absolute humidities at design outdoor (o) and indoor (i) conditions are determined from the psychrometric chart (see Jan 1948 POWER, p 83, for explanation of symbols):

$$w_o = 117 \text{ grains per lb}$$
$$w_i = 77 \text{ grains per lb}$$

The conditions of air entering the cooling surface are determined from:

9. $$t_1 - t_i = \frac{M_o}{M}(t_o - t_i)$$

10. $$w_1 - w_i = \frac{M_o}{M}(w_o - w_i)$$

and thus,

$$t_1 = 80 + \frac{1340}{4000}(95 - 80) = 85 \text{ F}$$

$$w_1 = 77 + \frac{1340}{4000}(177 - 77) = 90.4 \text{ gr per lb}$$

**Air-Flow Rates.** On the psychrometric chart this corresponds to a wet-bulb temperature of 70.8 F. Note that we used volume-flow rates instead of weight-flow rates called for in the equations. Resulting error is so slight that this method may be used without affecting the final result within measurable limits.

Rating tables for cooling surface may be used if available. They give latent- and sensible heat capacities for various dry- and wet-bulb temperatures, refrigerant temperatures and airflow rates. Together with condensing-unit ratings of suction temperature or pressure, speed, condensing-water requirements and capacity, a graph similar to Fig. 6 in Part I may be developed.

For this type of application use an evaporator temperature of 45 F. All that is necessary is to plot two points and draw a straight line through them, treating both condensing unit and coil in the same manner. It is merely necessary to select a surface having a 56% ratio between latent- and sensible-heat capacities when extracting 87,000 + 48,300 = 135,300 Btu per hr.

This is equivalent to removing heat equal to 135,300/12,000 = 11.28 tons refrigeration from about 4000 cfm of air initially at 85 F dry-bulb and 70.8 F wet-bulb temperature.

Required temperature and humidity of the air leaving the cooling surface, point 2 in Fig. 8, may be determined from:

11. $$t_1 - t_2 = \frac{H_s}{1.06 \times Q} \text{ and}$$

12. $$w_1 - w_2 = \frac{H_l}{0.66 \times Q}$$

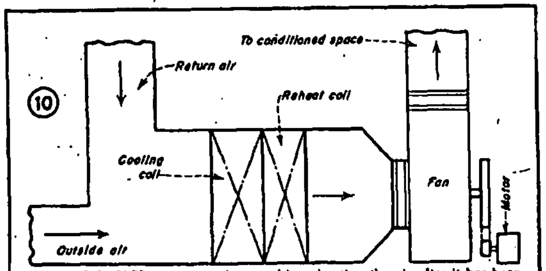
Accordingly, our figures show that:

$$t_2 = 85 - \frac{87{,}000}{1.06 \times 4000} = 64.5 \text{ F.}$$

$$w_2 = 90.4 - \frac{48{,}300}{0.66 \times 4000} = 72.1 \text{ gr per lb.}$$

**Surface Temperature.** Locating points 1 and 2 on a psychrometric chart, skeleton form Fig. 8, and extending the straight line joining them until it intersects the saturation curve, we find the required surface temperature, $t_s$, is 54 F. Another common term for this value is "apparatus dewpoint."

Mean temperature difference between




SURFACE COOLER can be supplemented by reheating the air after it has been cooled and dehumidified when the ratio of latent to sensible heat is high