

steam is normally used for heating the liquid absorbent. When it is desirable or necessary to use gas or electricity, an auxiliary low pressure steam boiler is usually added to the equipment. Concentrators operating on a simple boiler principle have not as yet been commercially practical.

It should be noted that the solution concentration phase is the reverse of the absorption process. During concentration, the aqueous vapor pressure of the solution is greater than that of the surrounding air, while during dehumidification, the reverse is the case. Utilization of this principle permits winter humidification by heating (instead of cooling) the solution pumped to the contactor. Water is thereby evaporated into, rather than condensed out of, the conditioned air stream. This requires dilution of the liquid absorber externally to the contactor.

CALCULATION OF MOISTURE LOAD

Calculation of the dehumidification required to maintain lower than normal moisture content in a given room begins with determination of the

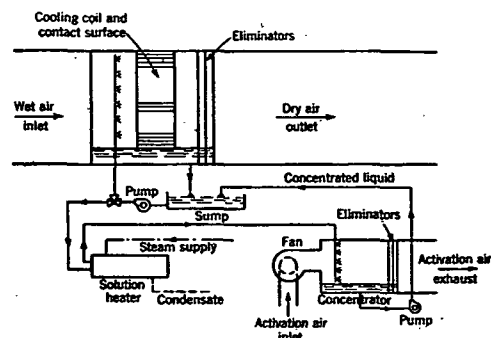


FIG. 5. LIQUID ABSORBENT EQUIPMENT IN WHICH SOLUTION COOLER AND CONTRACTOR ARE COMBINED

rate of moisture gain in the room from all sources. It is common practice, when maintaining a low humidity ratio, to recirculate a large percentage of the air in the room through the dehumidifier, and to add only enough outside air to meet the needs of the problem. The humidity ratio of the mixture of outside and recirculated air and the dehumidifier performance data can be used to calculate the humidity ratio of the air leaving the dehumidifier. The difference between the humidity ratio of the air in the room and that of the dehumidified air entering the room represents the effective dehumidification per pound of air. The rate of internal moisture gain in grains per minute, divided by the effective dehumidification in grains per pound of air, equals the air quantity required in pounds per minute. The following typical example using arbitrary values shows a general method of determining the dehumidifying requirements. Sensible heat determination considerations are discussed in other chapters, and are purposely omitted here.

Example 1: A solid absorbent dehumidifier having performance characteristics as shown in Fig. 6 is to be used to maintain inside conditions of 73 F and 20 percent relative humidity, i.e., 24.1 grains per pound of dry air, 30 F dew-point, in a room 20 ft x 30 ft x 10 ft high, having a total wall, ceiling, and floor surface area of 2200 sq ft. Outside design conditions are 72 F dew-point (118.4 grains per pound).

Internal sources of moisture are: 4 occupants; an open natural gas burner using 15 cu ft of natural gas per hour; an open top water tank, having an area of 2 sq ft exposed surface, in which water is maintained at 87 F, with air movement over the water surface being 100 fpm. Determine the quantity and condition of the dehumidified air to be supplied to the room.

Solution: The internal moisture gain consists of items 1 to 5.

- | | | Grains per Minute |
|---|--|-------------------|
| 1. From occupants: | $4 \times 1800/60 =$ | 120 |
| | 1800 grains per person per hour is obtained from Fig. 7, Chapter 6, by interpolation between curves C and D. | |
| 2. From burned gas: | $15 \times 650/60 =$ | 162 |
| | 1 cu ft natural gas produces approximately 650 grains of moisture. | |
| 3. From exposed water surface: | $2 \times 20 =$ | 40 |
| | Evaporation from water surface is assumed to be 20 grains per (minute) (square foot) at 87 F water with air movement of 100 fpm. | |
| 4. From infiltration: | $\frac{6000}{60 \times 13.56} \times (118.4 - 24.1) =$ | 696 |
| | One air change, 6000 cu ft, assumed per hour (see Chapter 11). | |
| 5. Moisture transmitted through room surface: | $\frac{2200}{60} \times 3 \times (0.783 - 0.176) =$ | 67 |
| | Permeability assumed to be 3 grains per (square foot) (hour) (inch Hg vapor pressure difference on two sides of wall). | |

Total moisture gain from internal sources 1085

Let q be the air delivered to the room, pounds per minute. Let it be assumed for this problem that 85 percent of the air is recirculated and 15 percent is outside air. Enough air must be supplied to replace leakage from the system or to satisfy normal ventilating requirements for the occupants of the room as given in Chapter 6, whichever is greater. The amount is estimated from experience or obtained by test.

The humidity ratio of the mixture of recirculated and outside air entering the dehumidifier is then:

$$\frac{0.85q(24.1) + 0.15q(118.4)}{0.85q + 0.15q} = 38.3 \text{ grains per pound entering dehumidifier.}$$

For the dehumidifier whose performance is shown in Fig. 6, for 38.3 grains per pound in entering air, the leaving humidity ratio will be 6.5 grains per pound.

Effective dehumidification in the room is $24.1 - 6.5$ or 17.6 grains per pound of supply air.

Then $q = \frac{1085 \text{ grains per minute}}{17.6 \text{ grains per pound}} = 61.6 \text{ lb air per minute}$ minimum that must be supplied to the room to maintain 30 F dew point.

Note that this figure represents the minimum requirement for the arbitrary conditions set forth and that in practice, safety margins should be added to the outside air percentage figure and to the calculated internal moisture gain.

VAPOR TRANSFER TO DEHUMIDIFIED SPACE

The walls enclosing a dehumidified space are subjected to a vapor pressure differential. The pressure of the vapor outside the walls tends to force moisture through the walls into the dehumidified zone of relatively low vapor pressure. As this process can be an unnecessary load on the