air being processed is determined by the cooling water temperature and the amount of moisture removed in the equipment. Control of leaving air temperature may be obtained by precooling the absorbent solution in a suitable surface cooler, by tap, well, or chilled water.

The excess water of condensation, which dilutes the brine, is removed in the solution concentrator. This is a low pressure steam heat exchanger which over-concentrates a portion of the weak liquor, and returns it to the main brine reservoir for re-cycling. The concentrator operates in the manner of an evaporative condenser, whereby moisture is evaporated from the brine by the heating coils into a stream of regeneration air taken from, and rejected to, the outside atmosphere. Low pressure steam is normally used for heating the brine. 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

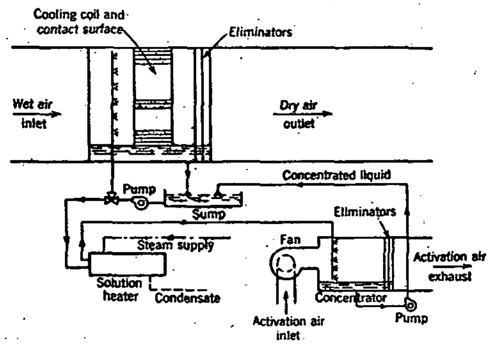

FIG. 5. LIQUID ABSORBENT EQUIPMENT IN WHICH SOLUTION COOLER AND CONTRACTOR ARE COMBINED

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, instead of being condensed out of, the conditioned air stream. This requires dilution of the brine externally to the contactor, rather than concentration.

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 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:	$\frac{6000}{60 \text{ x } 13.56} \times (118.4 - 24.1) =$	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.