good practice to rely entirely on blowing of air through the system.

Drifting Snow

It is quite likely that some drifting will occur on every system that is adjacent to a wall or vertical surface. The designer should try to anticipate this condition and add extra piping in these areas. If possible, coils should be added in the vertical surface. Another expedient is to carry the drainage to the area expected to be drifted. The drainage will tend to wash away some of the snow.

Design Example

Example 2: Assume that there are three independent hydraulic systems to be used in the snow-melting application described in *Example 1*. One system will be for the Class I area, another for the Class II, and the third for the emergency area, or Class III area.

Determine the hydraulic requirement for the Class II system, assuming an area of 100,000 sq ft.

Solution: The heating requirement has been determined as 160 Btuh per sq ft. Since an area of this type has a large perimeter, assume the edge and back losses add about 40 percent or 64 Btu per sq ft. The total heat requirement, then, will be taken as 224 Btuh per sq ft or 22.4×10^6 Btuh for the 100,000 sq ft area.

The fluid temperature is determined by using Equation 7:

$$t_m = 0.5\, q_o + t_f$$

Since

$$q_o = 160 \text{ Btuh per sq ft, and } t_f = 33 \text{ F}$$
$$t_m = 0.5 \times 160 + 33 = 113 \text{ F.}$$

If an oil antifreeze is used, the specific heat, c_m, the kinematic viscosity ν_m, and the specific weight w_m, at 113 F can be found by interpolation in Table 5 as

$$c_m = 0.440 \text{ Btu per (lb) (F deg)}$$
$$\nu_m = 3.88 \times 10^{-5} \text{ sq ft per sec.}$$

An accurate determination of viscosity cannot be made by interpolation in Table 5 and should therefore be made by use of *ASTM Standard Viscosity Chart B*.

$$w_m = 59.4 \text{ lb per cu ft}$$

Then from Equation 12

$$G = \frac{A_p q_t}{160 w_m c_m} = \frac{10^5 \times 225}{160 \times 59.4 \times 0.440} = 5380 \text{ gpm}$$

Equation 12 is based on a temperature drop of 20 F deg, therefore, the fluid must enter the slab at 123 F and leave at 103 F in order to have the desired mean temperature of 113 F.

Assume that the coil with the longest *equivalent* length is 300 ft and has a flow of 12 gpm. Such an arrangement would suggest the use of a coil of 1 in. IPS. From Table 1, Chapter 31, the ID = 1.049 in. = 0.0875 ft which gives an internal cross-sectional area of 0.00600 sq ft.

For 12 gpm the flow is 0.0268 cu ft per sec and the velocity is

$$V = \frac{0.0268}{0.00600} = 4.47 \text{ fps}$$

and Reynolds number, N_{Re}, is

$$N_{Re} = \frac{VD}{\nu_m} = \frac{4.47 \times 0.0875}{3.88 \times 10^{-5}} = 10{,}100$$

From Fig. 4, Chapter 4, using a roughness factor $e = 0.00015$, whence $e/D = 0.00172$, the friction factor f is found to be 0.0333. Substituting in Equation 10 yields

$$h_f = f \times \frac{L}{D} \times \frac{V^2}{2g} = 0.0333 \times \frac{300}{0.0875} \times \frac{4.47^2}{2 \times 32.16}$$
$$= 35.8 \text{ ft of fluid} = 14.8 \text{ psia}$$

The friction loss could have been approximated two ways: (1) by using Fig. 2, and (2) by using Fig. 3 of this chapter in conjunction with Fig. 5 of Chapter 28.

In order to use Fig. 2, the viscosity of the oil must be known in SSU. As given above the kinematic viscosity of the oil at 113 F is 3.88×10^{-5} sq ft per sec. From Table 6, 3.88×10^{-5} sq ft per sec is equal to 37.9 SSU. Entering Fig. 2 at 37.9 SSU and reading friction loss for 12 gpm gives 12.8 ft per 100 ft. For a 300-ft coil this is 38.4 ft of fluid. This compares fairly well with the more exact figure of 35.8 ft as found by using the Fanning equation.

If the friction loss were approximated by Fig. 3 and Fig. 5 of Chapter 28, the result would have been 40.5 ft which is determined as follows: From Fig. 5 of Chapter 28, the loss for *water* flowing at 12 gpm in a one-inch pipe is 1200 milinches per foot or 10 ft per 100 ft. In this case, then, the loss for *water* would be 30 ft.

The correction factor for oil as compared to water is given in Fig. 3 as 1.35. The friction loss then would be 1.35 × 30 or 40.5 ft. Both of the approximation methods are conservative, but for large systems, this conservative approach may be costly.

The pump must deliver 5380 gpm against a 14.8 psia head. The heat exchanger must deliver 22.5 million Btuh with an inlet temperature of 103 F and an outlet temperature of 123 F. Actually some allowance should be made for the temperature drop in the piping between the slab and the heat exchanger.

Both the pump and heat exchanger manufacturers should be advised that the fluid is an oil and will operate over a wide temperature and viscosity range. If a low-limit thermostat is placed in the fluid line so that the minimum fluid temperature is −10 F, it can be seen from Tables 5 and 6 that the viscosity can reach about 300 SSU. From Table 7, it can be found that such a viscosity can reduce the pump head 5 percent and the capacity 3 percent. This means that the pump should be rated at $\frac{14.8}{0.95} = 15.6$ psia and $\frac{5380}{0.97} = 5550$ gpm.

The expansion tank should be designed in accordance with the procedure outlined in Chapter 28, Hot Water Heating Systems. The amount of expansion should be based on the change in volume from the low temperature limit to the high temperature limit. Assuming the temperature range to be −10 F to +123 F, the expansion would be $\frac{62.7 - 59.1}{59.1} = 6.1$ percent of the total volume in the system.

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CHAPTER 50

PROCESS AND PRODUCT AIR CONDITIONING

General Requirements for Manufacture, Processing, and Preservation; Design Conditions and Application Data; Classification of Problems; Moisture Content and Regain; Conditioning and Drying; Chemical and Biochemical Reactions; Crystallization; Control for Machining, Polishing, and for Static Electricity Elimination; Laboratory Conditions; Calculations

PROCESS and product air conditioning is concerned with the design and application of equipment for obtaining proper conditions for the manufacturing, processing, and preserving of material, equipment, and commodities. This chapter includes a general discussion of these conditions and also a comprehensive list of specific requirements for various types of products.

GENERAL REQUIREMENTS FOR MANUFACTURE, PROCESSING, AND PRESERVATION

In order to apply air conditioning to industrial processes, the air-conditioning engineer must have a thorough understanding of the processing problems involved. Since individual processes and machines are changing rapidly, air conditions must be revised constantly to meet new requirements.

Table 1 lists the temperatures and relative humidities required for storage of certain commodities, and for manufacturing and processing of others. In some cases the temperatures and relative humidities listed in Table 1 have no direct influence upon the product itself, but do affect the efficiency of employees and, in turn, the workmanship, uniformity, and cost of production. Sometimes, a compromise between the known optimum condition for processing and that required for worker comfort is unavoidable.

Air conditioning for industrial processes is so extensive and involved that a detailed treatment is beyond the scope of this chapter. In many industries the exact conditions to be maintained are determined and known only by the manufacturer who specifies them. In other industries there is a wide variance between manufacturers' requirements, depending on results desired, experience, and cost considerations.

CLASSIFICATION OF PROBLEMS

In general, any industrial air-conditioning problem in processing may be classified under one or more of the following:

1. Control of regain.
2. Control of rate of chemical reactions.
3. Control of rate of biochemical reactions.
4. Control of rate of crystallization.
5. Control of temperature for close tolerance machining, and grinding.
6. Control of dew point for protection of highly polished surfaces.
7. Control of humidity for static-electricity elimination.
8. Control of conditions for material-testing laboratories.

(*Text continued on p. 687*)

Table 1 .... Temperatures and Humidities Applicable to Industrial Air Conditioning*

Process	*Temp. F*	*R.H.%*
BAKERY		
Mixer (bread dough)	75–80	40–50
Fermenting	75–80	70–75
Proof box	92–96	80–85
Bread oven	375–450	
Cake oven	300–430	
Bread cooler (room or tunnel) Vacuum 28.6 in.	70–75	80–85
Cold room	40–45	
Make-up room	75–80	65–70
Cake mixing	70–75	65
Cake mixing (sponge)	95–110	
Crackers and biscuits	60–65	50
Wrapping	60–65	60–65
Dried ingredients, storage	70	55–65
Fresh ingredients, storage	30–45	80–85
Flour storage	65–80	50–65
Shortening (depending on type), storage	45–70	55–65
Sugar, storage	80	35
Water, storage	32–35	....
Wax paper, storage	70–80	40–50

METHODS OF MIXER COOLING

1. 35–40 F water circulated through mixer jacket.
2. 15–25 F brine.
3. Direct expansion, refrigerant circulated through mixer jacket.
4. Cracked ice added to dough in mixer.
5. Cold air introduced into mixer during mixing process.
6. Cooled agitators are used in mixers.

MIXER LOAD CALCULATIONS

Refrigeration is required to remove: excess ingredient, if any; heat generated by the beating and mixing of dough; excess heat in mixer body; heat of hydration of flour and water; and heat absorbed by mixer from atmosphere during mixing.

Additional factors are the design and speed of mixer, consistency, kind and mass of dough.

Data Used in Calculations:

1 bbl. flour = 200 lb.
Heat of hydration = 6.5 Btu per lb of flour.
Specific heat of flour = 0.42 Btu per lb.
Water is 65% of weight of flour.
Flour is 65% of batch.
Sponge is 60% of batch.

Total motor output is converted to heat in the mixer.

In fermenting rooms recent practice is to use direct radiation for heating, atomizing sprays for humidifying, and gravity con-

* Information in Table 1 is drawn from many sources. See bibliography at end of chapter.