

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 Btu 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 Btu per sq ft or 22.4×10^3 Btu for the 100,000 sq ft area.

The fluid temperature is determined by using Equation 7:

$$t_m = 0.5 t_a + t_f$$

Since

$$q_a = 160 \text{ Btu per sq ft, and } t_f = 33^\circ \text{ F}$$

$$t_m = 0.5 \times 160 + 33 = 113^\circ \text{ F}$$

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

$$c_p = 0.440 \text{ Btu per (lb) (F deg)}$$

$$\nu = 3.88 \times 10^{-4} \text{ sq ft per sec.}$$

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*.

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

Then from Equation 12

$$G = \frac{A q_a}{160 \nu c_p} = \frac{10^4 \times 225}{160 \times 59.4 \times 0.440} = 5380 \text{ gpm}$$

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 0.00690 sq ft.

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

$$V = \frac{0.0268}{0.00690} = 4.47 \text{ fps}$$

and Reynolds number, N_{Re} , is

$$N_{Re} = \frac{VD}{\nu} = \frac{4.47 \times 0.0875}{3.88 \times 10^{-4}} = 10,100$$

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

$$h_f = f \times \frac{L}{D} \times \frac{V^3}{2g} = 0.0333 \times \frac{300}{0.0875} \times \frac{4.47^3}{2 \times 32.16}$$

$$= 35.8 \text{ ft of fluid} = 14.8 \text{ psia}$$

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^{-4} sq ft per sec. From Table 6, 3.88×10^{-4} 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 millinches 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 Btu/h 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 $14.8 = 15.6$ psia and $5380 = 5550$ 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 $+123^\circ \text{ F}$, the expansion would be $\frac{62.7 - 59.1}{59.1} = 6.1$ percent of the total volume in the system.

REFERENCES

- W. P. Chapman and S. Katunich: Heat requirements of snow melting systems (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 359).
- W. P. Chapman: Design of snow melting systems (*Heating and Ventilating*, April 1952, p. 95, and November 1952, p. 88).
- W. P. Chapman: Calculating the heat requirements of a snow melting system (*Air Conditioning, Heating and Ventilating*, September 1956 through August 1957).
- W. P. Chapman: Snow melting system hydraulics (*Air Conditioning, Heating and Ventilating*, November 1955).
- F. B. Gordon: Antifreeze protection for snow melting systems (*Heating, Piping and Air Conditioning Contractors National Association Official Bulletin*, February 1950, p. 21).
- C. S. Crago: Properties of Ethylene Glycol and Its Aqueous Solution (National Bureau of Standards, Society of Automotive Engineers, CKC report No. 9).
- W. P. Chapman: Are thermal stresses a problem in snow melting systems? (*Heating, Piping and Air Conditioning*, June 1955, p. 104, and August 1955, p. 92).

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:

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

(Text continued on p. 705)

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

Process	Temp. F	R.H. %	METHODS OF MIXER COOLING
BAKERY			1. 35-40 F water circulated through mixer jacket.
Mixer (bread dough).....	75-80	40-50	2. 15-25 F brine.
Fermenting.....	75-80	70-75	3. Direct expansion, refrigerant circulated through mixer jacket.
Proof box.....	92-96	80-85	4. Cracked ice added to dough in mixer.
Bread oven.....	375-450		5. Cold air introduced into mixer during mixing process.
Cake oven.....	300-430		6. Cooled agitators are used in mixers.
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	

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 66% 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.