

established by Babcock in formula 1.

$$P = 0.000000367 \left(1 + \frac{3.6}{d} \right) \frac{W^2 L}{D d^5} \quad (1)$$

or

$$W = 5220 \sqrt{\frac{P D d^5}{\left(1 + \frac{3.6}{d} \right) L}} \quad (2)$$

where

P = loss in pressure, pounds per square inch.

d = inside diameter of pipe, inches.

L = length of pipe, feet.

D = weight of 1 cu ft of steam.

W = weight of steam flowing per hour, pounds.

Example 1. How much steam will flow per hour through 100 ft of 2-in. pipe if the initial pressure is 1.3 lb per square inch and the pressure drop is 1 oz?

Solution. $P = \frac{1}{16} = 0.0625$ lb; $d = 2.067$ in. (Table 1, Chapter 18); $L = 100$ ft; $D = 0.04038$ lb (Table 8, Chapter 1). Substituting these values in Formula 2:

$$W = 5220 \sqrt{\frac{0.0625 \times 0.04038 \times 2.067^5}{\left(1 + \frac{3.6}{2.067} \right) 100}} = 97.2 \text{ lb per hour.}$$

Formula 2 does not allow for entrained water in low-pressure steam, condensation in pipe, and roughness in commercial pipe as found in practice.

The latent heat of steam (h_{fg}) at atmospheric pressure (Table 8, Chapter 1) is 970.2 Btu per pound. Inasmuch as the heat emission of an equivalent square foot of heating surface (radiation) is 240 Btu, 1 lb of steam at this pressure will supply $\frac{970.2}{240}$ or 4.04 sq ft of equivalent heating surface. This figure is usually taken as 4 even. In Example 1, the weight of steam flowing per hour would therefore supply 4×97.2 or 388.8 sq ft of equivalent heating surface.

PIPE SIZES

The determination of pipe sizes for steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and the end of the return system.
2. The maximum velocity of steam allowable for quiet and dependable operation of the system.
3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.
4. Unusual conditions in the building to be heated.

Initial Pressure and Pressure Drop

Theoretically there are several factors to be considered, such as initial pressure and pressure required at the end of the line, but it is most important that (1) the total pressure drop does not exceed the initial pressure of the system; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems specially designed for varying initial pressures, such as the sub-atmos-

pheric which normally operate under controlled partial vacua, the orifice, and the vapor systems which at times operate under such partial vacua as may be obtained due to the condition of the fire; (4) there is sufficient difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, and the dry return, when considered in relation to the boiler water line.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first*, the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second*, a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

The total pressure drop should never exceed one-half of the initial pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. Laboratory experiments limit this to the capacities given in Tables 1 and 2 for vertical risers and in Table 3 for horizontal pipes at varying grades.

Maximum Velocity and Reaming

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensation present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically, (2) the pitch of the pipe if it runs horizontally, (3) the quantity of condensate flowing against the steam, and (4) freedom of the piping from water pockets which under certain conditions act as a restriction in pipe size.

Three factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided and which caused an actual difference of 20 per cent in the capacity of a 1-in. pipe in experiments carried on at the A.S.H.V.E. Research Laboratory (Table 4). The second is the reaming of the ends of the pipe after cutting, which, experiments indicate, might reduce the capacity of a 1-in. pipe as much as 28.7 per cent (Table 5). The third is the uniformity in grading the pipe line. All of the capacity tables given in this chapter include a factor of safety. However, the pipe on which Table 4 is based showed no particular defects or constrictions on the inside, and the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*