U. S. DEPARTMENT OF COMMERCE NATIONAL BUREAU OF STANDARDS

RESEARCH PAPER RP1086

Part of Journal of Research of the National Bureau of Standards, Volume 20, April 1938

CROSS-CONNECTIONS IN PLUMBING SYSTEMS

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ABSTRACT

This paper deals principally with the technical aspects of the problem of preventing the backflow of water from plumbing fixtures into water-supply systems. It starts with a general review of the subject, including a brief history of previous work on the subject, a classification of cross-connections, and a brief discussion of vacua and siphon action. This is followed by a mathematical and experimental analysis of the conditions tending to produce backflow into a supply line. This analysis makes it possible to determine the worst conditions, as regards backflow, that can occur in any building supply system, and to determine minimum requirements for the positive prevention of backflow under these conditions. Specifically, the minimum pressure that can occur in any system, the maximum rate at which water can be removed from the supply risers under this minimum pressure, the smallest air gap between a faucet and plumbing fixture that can be safely allowed under the worst conditions, and the essential performance characteristics of a siphon-breaker are determined. The effectiveness of various types of siphon-breakers in preventing backflow is discussed, and the operation of one type of flush valve is explained in order to show the essentials of a stable flush valve, that is, one which will not open under any possible reduction in supply pressure. Finally, there is given a brief review of the entire subject of preventing backflow from plumbing fixtures, in which two distinct methods of attack are pointed out, and the merits of each are discussed. The conclusions relate only to the technical aspects of the subject and do not take the form of proposed health or plumbing regulations.

CONTENTS

I. INTRODUCTION

1. EARLY ATTENTION TO CROSS-CONNECTIONS

For several decades, cross-connections and their menace to health have been subjects of active interest to engineering, public health, and trade organizations, but as yet no generally satisfactory conclusions, either as to the extent to which cross-connections may be a menace to health or as to practical regulatory measures for controlling this condition, have been reached. The subject was presented to the New England Water Works Association as early as 1894 [1]* and was later made the subject of committee investigations and reports by the New England Water Works Association [2], the American Water Works Association [3], and the American Society of Sanitary Engineering [4]. The subject has also been frequently before the American Public Health Association [5], the National Association of Master Plumbers [6], and other national, sectional, and local organizations for discussion and official action. A large number of papers on the subject have been published in engineering, public health, medical, and trade journals. For the most part, publications on the subject deal with the occurrence of cross-connections and their menace to health rather than with a practical correction of the situation.

Numerous epidemics have been traced or attributed to cross-connections in plumbing systems [7]. Probably the most publicized and most thoroughly investigated case on record was that of an outbreak of amoebic dysentery in a hotel during the World's Fair in Chicago in 1933 [8]. Lists of epidemic outbreaks attributed to cross-connections in plumbing systems have been compiled, but relatively few of the available records give detailed information, either regarding the cross-connections to which the epidemic was attributed or the means by which it was determined that the epidemic was caused by the cross-connection.

Among the published information dealing more directly with the correction or removal of the hazards of cross-connections in plumbing systems, the reports of investigations at the University of Wisconsin under the direction of Professor F. M. Dawson [9], and at the Massachusetts Institute of Technology under the direction of Professor T. R. Camp [10] are of particular interest. Two more recent papers by Professor Dawson and Mr. Kalinske continue with the analysis of the problem [11, 12]. A number of reports of investigations in Germany [13 to 19] on the problem of cross-connections have been published. None of these papers contains a complete analysis of the problem in respect to (1) a classification of cross-connections into physical types; (2) a determination of the limiting conditions in water-supply systems as they affect backflow through a cross-connection; and (3) a determination of the minimum requirements for the prevention of backflow through a cross-connection of a given type.

2. PURPOSE AND SCOPE OF THE INVESTIGATION

In February 1936, the National Bureau of Standards commenced an investigation of cross-connections in plumbing systems for the purpose of obtaining the technical information and data required for

*Numbers in brackets indicate literature references at the end of this paper.

the determination of minimum requirements for preventing backflow of water into the water-supply lines from plumbing fixtures or drains and for putting this information into a form available for the use of Federal, State, and municipal authorities in the formulation of effective health regulations applying to plumbing.

The investigation as planned included: (1) A survey of the literature on the subject, with the collection and analysis of available data; (2) the definition and classification of cross-connections; (3) an analysis of the physical conditions and principles involved; (4) experimental verification of the analysis; and (5) correlation of data for practical application in eliminating the hazards of cross-connections.

The ultimate purpose of the investigation is to establish an organized body of information on this subject that will afford a basis for decreasing, or even eliminating entirely, the hazards due to cross-connections. Since the accomplishment of this purpose is dependent on the cooperative endeavors of a number of agencies and interests—governmental, engineering, construction, and owners or operators—no complete and satisfactory solution of the problem of cross-connections by a single agency or individual can be expected. However, a common ground or basis of agreement must be found, and this paper is offered as a contribution to that end, rather than as a complete solution from the standpoint of health regulations.

In fact, the paper should be looked upon as a progress report, rather than as a complete and final report on the investigation. It will be some time before all phases and details of the project as it is now conceived will be finished; and since the theoretical analyses and experimental verifications of certain aspects of the problem have been completed, it seemed advisable to publish these data and analyses without delay, so that they would thus be made conveniently available to other investigators and to those who may wish to make practical application of the results.

II. OUTLINE OF THE PROBLEM

1. DEFINITION AND CLASSIFICATION OF CROSS-CONNECTIONS

A cross-connection may be defined as any physical connection or arrangement of pipes between two water piping systems whereby water may flow from one system to the other, the direction of flow depending on the direction of the pressure differential between the two systems. A cross-connection becomes a hazard to health when one system carries a water used for human consumption and the other carries an impure or contaminated water. It will be convenient for purposes of discussion and analysis to divide cross-connections into two general classes, *direct* and *indirect*, although there is no sharp demarcation between the two in principle.

A *direct cross-connection* may be defined as a continuous inclosed interconnection between two piping systems, such that the flow of water from one system to the other may occur whenever a pressure differential is set up in the connection between the two systems. Examples: Interconnections between dual water-distributing systems, completely submerged inlets from water-supply lines to closed plumbing fixtures, tanks and vats, continuous water connections between the supply and drain systems, priming lines to pumps, etc.

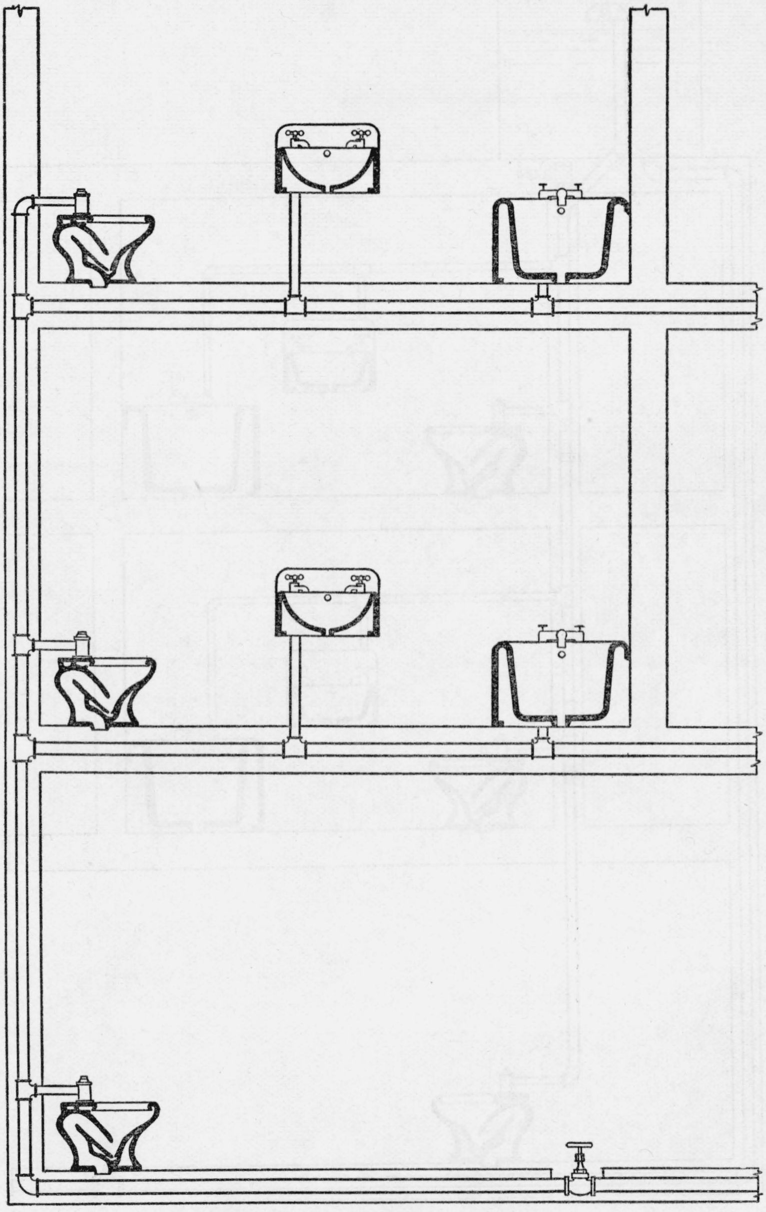

FIGURE 1.—*Up-feed supply system.*

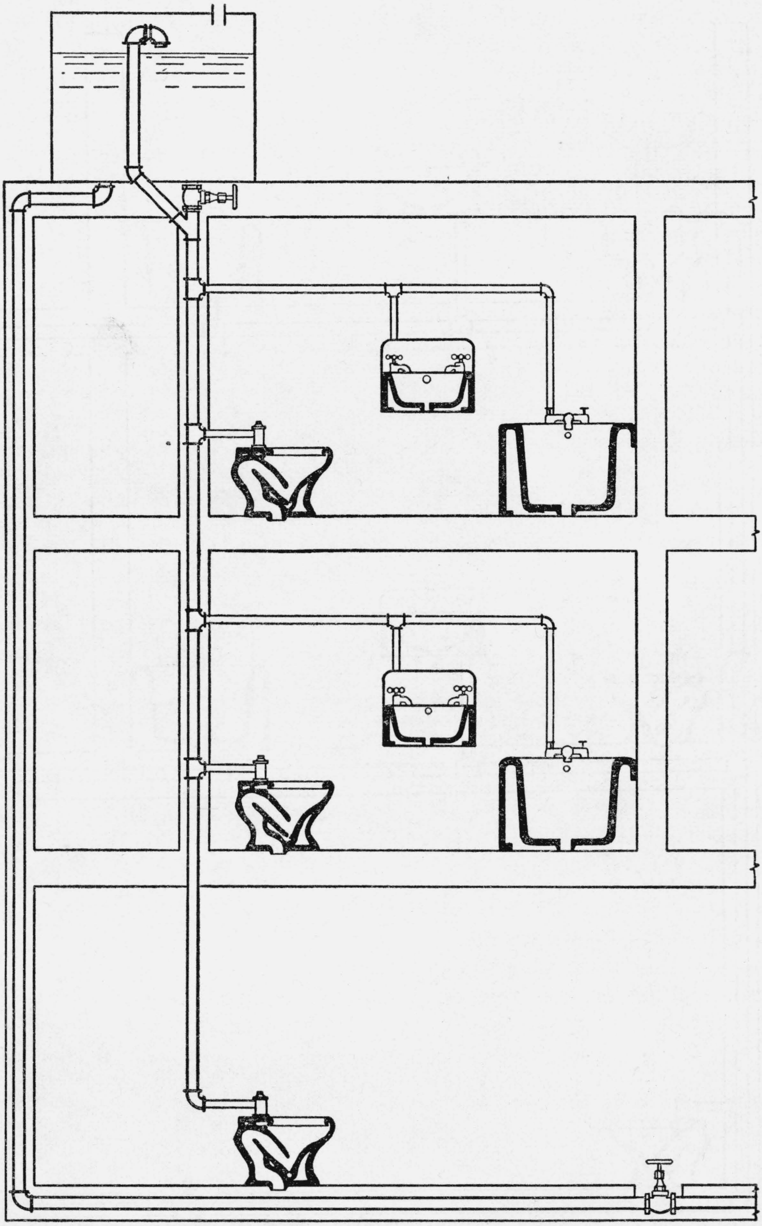

Figure 2.—*Down-feed supply system.*

An *indirect cross-connection*, frequently referred to as a potential cross-connection, is one in which the interconnection is not continuously inclosed, and the completion of the cross-connection depends on the occurrence of one or more abnormal conditions. Examples: Water closets with direct flush-valve supply, bathtubs and lavatories with faucet openings that may become submerged, and other plumbing fixtures and equipment whose supply inlets may become partially or wholly submerged.

Among the abnormal conditions, using the term to include all conditions not contemplated or intended, as well as in the sense of unusual, the following are the conditions that may occur or may be necessary to complete the cross-connection: (1) A drop in the static pressure in the supply system to such a point that a pressure differential acting in the direction of the supply system is produced in the supply connection to the fixture; (2) the formation of a vacuum by displacement of water from the supply line; (3) a flooding of the fixture by stoppage or other causes; and (4) an open or leaking faucet or valve. In general, the simultaneous occurrence of two and sometimes all of these conditions is necessary to produce flow from the fixture into the supply line.

2. TYPES OF INDIRECT CROSS-CONNECTIONS

Although there are many variations in the details of indirect cross-connections as they occur in plumbing systems, they may be grouped into two classes in respect to the general principles involved in the prevention of backflow: (1) Cross-connections between the water-supply system and an open-top fixture with faucet supply, such as washbasins, bathtubs, sinks, laundry trays, water-closet flush tanks, etc. (figs. 3 and 4); and (2) cross-connections between the water-supply system and an open-top fixture requiring a direct connection for pressure flushing, such as water closets and pedestal urinals (fig. 5).

A cross-connection between a water-supply and drainage system through a closed fixture illustrated diagrammatically by figure 6 may be classed as an indirect cross-connection by some, since a combination of two or more abnormal conditions may be required to produce backflow into the water-supply system; for example, an abnormal decrease in the service pressure, an abnormal increase of pressure in the drainage system, an abnormal filling of the fixture, tank, or vat, as the case may be, or a combination of all these conditions. However, a cross-connection of this type complies with the definition of a direct cross-connection in that it presents a continuous inclosed passage between two piping systems through which the pressure in one system may be transmitted to the other, and in that it presents the same problem in prevention of backflow as a direct pipe connection between the two systems.

3. SYSTEMS OF WATER SUPPLY

There are three fairly distinct methods of supplying and distributing water in buildings: (1) The up-feed system with a direct service pipe to a service main (fig. 1); (2) the down-feed system from an overhead tank exposed to atmospheric pressure (fig. 2); and (3) the pressure-tank supply with either up-feed or down-feed distribution, the tank pressure being maintained either by a direct connection with a pressure

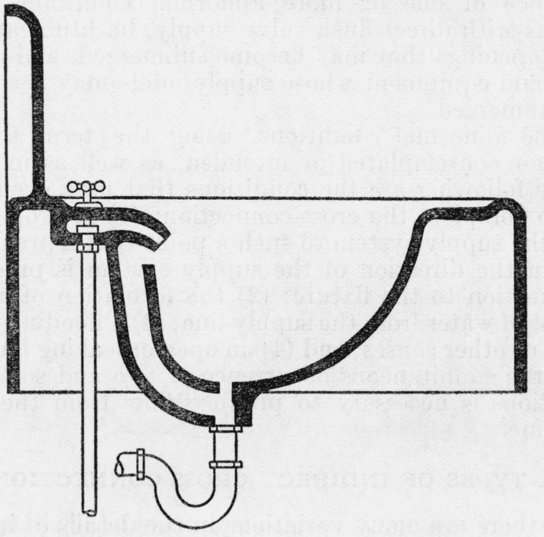

FIGURE 3.—*Indirect cross-connection between open-top fixture and faucet supply.*

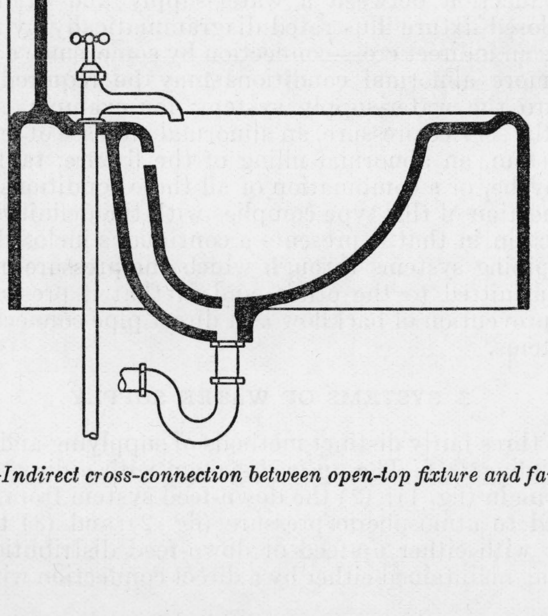

FIGURE 4.—*Indirect cross-connection between open-top fixture and faucet supply.*

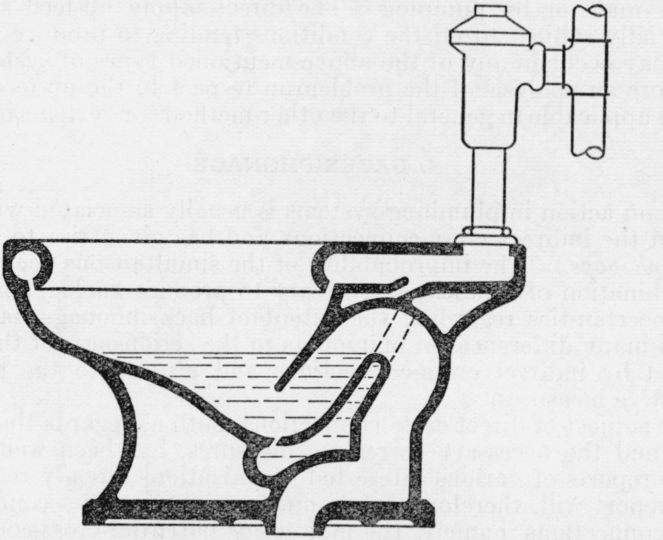

FIGURE 5.—*Water closet with direct connection for pressure flushing.*

Indirect connection.

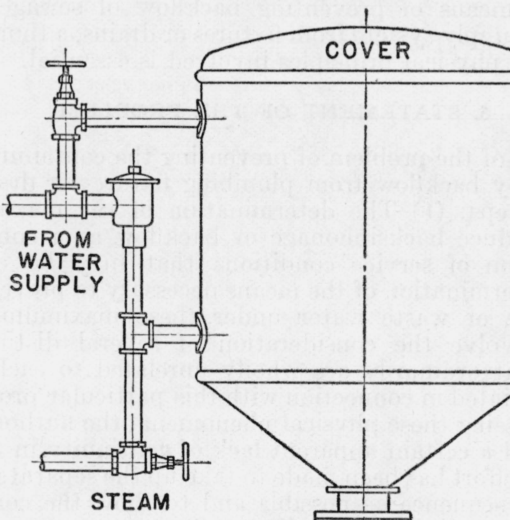

FIGURE 6.—*Direct cross-connection through a closed fixture.*

supply main or by pumping. The direct-supply up-feed system is potentially subject to all the conditions tending to produce backflow that may occur in any of the above-mentioned types of systems, and therefore an analysis of the problem in respect to the up-feed system will be applicable in general to the other methods or systems of supply.

4. BACKSIPHONAGE

Siphon action in plumbing systems is usually associated with backflow in the indirect cross-connections and has given rise to the term *backsiphonage.*[1] The improbability of the simultaneous occurrence of a combination of conditions necessary to produce backsiphonage and the uncertainties regarding the extent of backsiphonage have given rise to many differences in opinion as to the seriousness of the hazard created by indirect cross-connections and also as to the necessary corrective measures.

The subject of direct cross-connections, both as regards their prevalence and the necessary corrective measures, has been well covered in the reports of various interested organizations already referred to. This report will, therefore, be confined mainly to the second class of cross-connections, namely, the indirect or potential cross-connections between the water supply and the drainage system through plumbing fixtures and other mechanical equipment.

Since the final purpose of the investigation is to indicate an effective and practical means of preventing backflow of sewage and waste water into the supply system from fixtures or drains, a thorough understanding of the physical principles involved is essential.

5. STATEMENT OF THE PROBLEM

The solution of the problem of preventing the contamination of the water supply by backflow from plumbing fixtures or drains involves two distinct steps: (1) The determination of the maximum effects tending to produce backsiphonage or backflow as a consequence of any combination of service conditions that may be encountered; and (2) the determination of the means necessary to prevent the backflow of sewage or waste water under these maximum conditions. Both steps involve the consideration of several distinct physical phenomena that ordinarily are wholly unrelated to each other, but which are associated in connection with this particular problem. As a result, in discussing these physical phenomena, the authors have been unable to avoid a certain apparent lack of continuity in the development. Every effort has been made to take up the separate phenomena in as logical a sequence as possible and to show the connection between them when they are first discussed.

In order to avoid any misunderstanding as to the use of terms and to make the discussion complete, some commonly understood physical phenomena, as well as some that are not so familiar, are discussed in the following pages. To a large extent, the detailed discussion of physical principles and phenomena, together with the experimental data pertaining to them, are grouped in sections III and IV in order to break the continuity of the general discussion of the subject as little as possible.

[1] The corresponding term, *Rücksaugung*, has come into common use in Germany.

6. DEFINITIONS OF TERMS RELATING TO PRESSURES AND VACUA

(a) ABSOLUTE PRESSURE AND GAGE PRESSURE

The pressure in a water-supply system is ordinarily expressed as *gage pressure;* that is, the pressures are based on atmospheric pressure as a datum. Hence, when the line pressure falls below atmospheric pressure, the gage pressure is negative. In some cases it is desirable to refer gage or manometer readings to the absolute scale of pressures, the zero of which is approximately 14.7 lb/in.^{2} below normal atmospheric pressure at sea level. Pressures that are so measured are called *absolute pressures.* All pressures involved in the analysis of the phenomena discussed in what follows will be expressed in this scale of pressures. If a gage pressure is used, this will be expressed as a difference between the corresponding absolute pressure and atmospheric pressure.

(b) PRESSURE UNITS USED

The pressure in a water-supply system is ordinarily expressed in pounds per square inch (when English units are used), but for the purposes of this paper it will frequently be more convenient to express it as the height in feet of a water column that would exert at its base a pressure equal to the given pressure in pounds per square inch.

Atmospheric pressures are commonly expressed in inches height of a column of mercury, but obviously they may be expressed equally well in pounds per square inch or in feet height of water column.

(c) DEFINITION OF A VACUUM

A vacuum is usually defined as space devoid of matter. This is the hypothetical perfect vacuum, in which, if it were obtainable, the pressure in the evacuated space would be zero on an absolute scale of pressures. Actually, an absolute-zero pressure cannot be produced over water, since some water vaporizes into the space over the water and continues to vaporize until equilibrium is established. This equilibrium pressure is called the vapor pressure of water, and its value depends on the temperature and purity of the water. At 0° C the vapor pressure of water is small, but it increases rapidly with increase in temperature, and at 100° C it is equal to standard atmospheric pressure, approximately 34 feet of water at sea level.

Similarly, if the water contains dissolved air or other gases, as is usually the case, some of the gases will come out of solution into the evacuated space until equilibrium is established, and each gas will exert a pressure independently of the vapor pressure and of the partial pressures of the other gases present.

The total pressure in an inclosed vacuum in equilibrium with water containing air or other gases in solution is equal to the sum of the partial pressures of the water vapor, air, and other gases at the existing temperature. This total pressure varies with the temperature, and, for dissolved gases, also with the saturation and the ratio of the evacuated volume to the volume of water remaining in the system when the latter comes to equilibrium. The equilibrium pressure, which will be referred to as the *limiting vacuum pressure* and designated by the symbol h_e, is small in cold-water systems, but in hot-water systems its value may equal or exceed the atmospheric pressure h_a if the temperature equals or exceeds the boiling temperature at atmospheric pressure.

The following definitions will be observed in this paper in referring to vacua in water-supply systems:

A *vacuum* is any space in a water-supply system from which water has been displaced by water vapor, air, or other gases, and in which the pressure is less than the prevailing atmospheric pressure.

A *limiting vacuum* is a vacuum in an inclosed space in which the absolute pressure is equal to the equilibrium pressure h_e.

A *partial vacuum* is any vacuum in which the pressure lies between the prevailing atmospheric pressure h_a and the limiting pressure h_e.

It is to be understood that the values of h_a and h_e depend on many different factors, some of which have been mentioned above. Hence, in general, we cannot assign definite values to these two pressures. However, it will be convenient in the following discussion to regard h_a as having a limiting value of 34 feet, expressed as height of water column, and to regard h_e as representing in general a very low absolute pressure. With this understanding, the symbols h_a and h_e will usually be employed in this paper in the discussion of vacua as if they had definite fixed values.

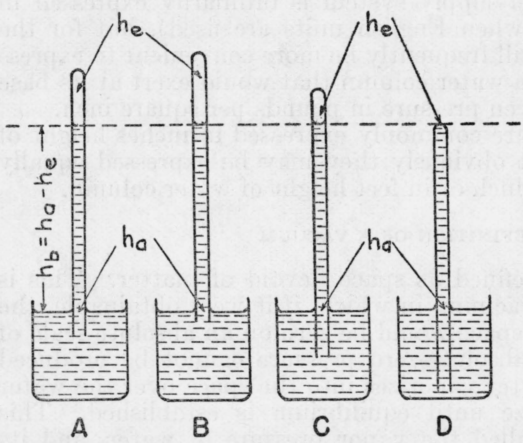

FIGURE 7.—*Simple liquid barometer.*

7. ATMOSPHERIC PRESSURE AND SIPHON ACTION

The part played by the atmospheric pressure in producing siphon action or backflow in plumbing systems apparently has not been understood thoroughly by some who have discussed this subject. Consequently, it seems advisable to state here some of the relations involved in siphon action.

The atmosphere at sea level exerts a pressure of approximately 14.7 lb/in.^{2}, equivalent to the pressure at the base of a column of water 34 feet in height or at the base of a column of mercury 30 inches in height. Therefore, the atmosphere at sea level will support a column of water approximately 34 feet in height in a tube against a perfect vacuum (zero absolute pressure). Actually, at any given time and place, the height of the water column, illustrated by a simple water barometer in figure 7, will be given by the equation $h_b = h_a - h_e$. If there are no dissolved gases in the water, then when the barometer tube is raised and lowered successively, figure 7 (*B*, *C*, and *D*), the evacuated volume over the column of water will increase or decrease correspondingly, and the length of the column of water h_b will be the same in each case, once the system has come to equilibrium. If the water contains air in solution, the system will come to equilibrium with slightly different values of the equilibrium pressure h_e in the cases illustrated, since the change in the relative volumes of vacuum and water exposed produces a change in the partial pressure due to the dissolved air, as explained in the preceding section. In any case, h_b represents the limit in the height that water can be lifted from an open vessel by siphon action in a completely filled siphon tube, since the atmospheric pressure supplies the entire lifting force. This limit, as defined here, refers to atmospheric pressure at sea level, and it decreases proportionately as the atmospheric pressure decreases with elevation above sea level.

Since the pressure exerted at the base of a liquid column varies directly as its density, the maximum height to which a liquid can be lifted by siphon action varies inversely as the density of the liquid. Therefore, the limit in the height to which a mechanical mixture of air and water, such as may flow into a partially submerged inlet, can be lifted will be approximately equal to $h_b\ (\rho_w/\rho_m)$, where ρ_w is the density of the water, and ρ_m is the density of the mixture, provided the velocity of flow through the siphon is sufficient to prevent the air from separating from the mixture in the siphon tube. If the velocity is too low, the air will separate from the water and collect at the top of the tube (air-lock) and will stop the siphon action. It is impossible to connect this theoretical limit in the height of lift of a mixture of air and water by siphon action numerically with the height of lift in a water-supply system because there is no known means of determining the exact conditions, such as velocity and height of lift, under which air-lock of the siphon will or will not occur. The most that can be positively stated in this respect is: If siphonage of a mixture occurs, the limit in lift may be many times the limit of lift for pure water through a completely submerged siphon inlet.

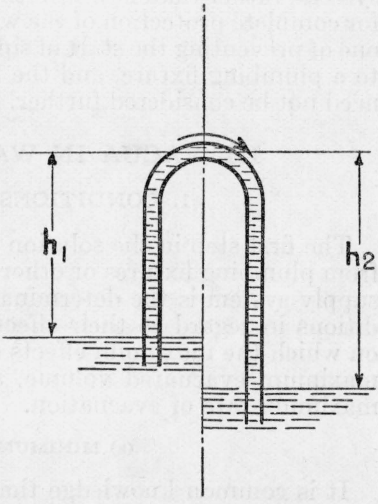

FIGURE 8.—*Simple siphon.*

The quantitative relation of the forces acting to produce siphonage may be demonstrated by reference to the simple siphon, shown in figure 8. Considering the component of pressure acting to produce flow in the direction of the arrow as positive, there will be a component of $+h_a$ acting on the inlet,[2] a component of $-h_a$ acting on the outlet,[3] a component of $-h_1$ produced by the column of water in the inlet leg and a component of $+h_2$ produced by the column in the outlet leg. The resultant pressure will be the sum of the components

$$+h_a-h_a+h_2-h_1=h_2-h_1$$

There is, therefore, a head of h_2-h_1 available to produce flow in the tube, and the velocity at any time will be such that this available head h_2-h_1 is exactly equal to the friction loss in the tube plus the entrance and exit losses.

[2] [3] At the level of the external water surface in each case.

It should be pointed out that the sole functions of the outlet leg are: (1) To reduce the pressure at the top of the siphon tube so that the atmospheric pressure can lift the water in the inlet leg, and (2) to provide a passage for flow. Hence, flow in the inlet leg will take place in exactly the same manner, for a given reduction in pressure at the top of the siphon, regardless of how the reduction in pressure at the top of the siphon is produced. Therefore, the existence of a vacuum, whether it be a partial vacuum or the limiting vacuum, in the water-supply system will produce backflow through a supply branch, as a result of atmospheric pressure acting through the supply branch, in exactly the same manner and with the same limits of lift as would be produced by a complete siphon tube with an outlet leg capable of maintaining the same pressure in the top of the siphon as exists in the vacuum.

If the positive prevention of backflow into any part of the water system, rather than flow into the water mains, is taken as the criterion for complete protection of the water supply, the problem then becomes one of preventing the start of siphon action in any water-supply branch to a plumbing fixture, and the limits of height of lift and rate of flow need not be considered further.

III. VACUA IN WATER-SUPPLY SYSTEMS

1. CONDITIONS TO BE CONSIDERED

The first step in the solution of the problem of preventing backflow from plumbing fixtures or other mechanical equipment into the water-supply system is the determination of the limits applying to the conditions in regard to their effects in producing backflow. The limits on which the maximum effects depend are the minimum pressure, the maximum evacuated volume, and, closely related to the latter, the maximum rate of evacuation.

(a) MINIMUM SERVICE PRESSURE

It is common knowledge that the service pressure in water-supply systems may sometimes fail completely, as by the breaking of a supply main by a shutdown or breakdown of a pumping system, and, so far as a building water-supply system is concerned, may approach complete failure under heavy demands on water mains, as from fire pumps, or from other unusual heavy service demands. Since no one can predict with certainty when or where such failures in the main service pressure will occur, there is only one minimum limit that can be established for service pressures for buildings in general, namely, a drop in service pressure that will reduce the pressure at the base of the building water-supply system to the limiting vacuum pressure over water, h_e.

(b) MINIMUM PRESSURE IN BUILDING SUPPLY SYSTEMS

Since we can set no definite limit for the main supply pressure short of the limiting vacuum pressure, the same limit must apply to an unvented building supply system. However, the minimum limit will be reached more frequently in the building supply system than in a supply main because the former is higher than the latter. The limiting pressure h_e will occur at the top of any unvented up-feed system whenever the service pressure falls to atmospheric pressure at a level 34 feet or more below the top.

(c) MAXIMUM EVACUATED VOLUME IN BUILDING SUPPLY SYSTEMS

Just as occurs in the tube of a water barometer, a vacuum will form in the top of an unvented water-supply system whenever the service pressure from any cause falls to atmospheric pressure at a level 34 feet below the top, and the pressure in the vacuum will be the limiting vacuum pressure h_e. If the system is vented or leaking at the top, a partial vacuum will be formed in which the pressure will lie between the limiting pressure h_e and atmospheric pressure h_a while the service pressure is dropping and will become equal to h_a when the system comes to equilibrium. In either case, whether the system is vented or unvented, the evacuated volume obviously will be exactly equal to the volume of water that has flowed out of the system and may extend to occupy its entire internal volume, depending on a drop in service pressure at the base of the system to h_e in the case of an unvented system, and on a drop to h_a in the case of a vented or leaking system. Therefore, the only definite limit that can be set for the maximum evacuated volume is the internal volume of the system expressed in terms of the diameter and length of the pipe of which the system is constructed.

(d) MAXIMUM RATE OF EVACUATION OF BUILDING SUPPLY SYSTEMS

If the system is vented or leaking, the rate of evacuation (rate of outflow of water) is quite as important a factor in the determination of maximum effects as the evacuated volume at any particular time. The maximum rate of evacuation will have a definite application in determining the capacities of vents required to control the pressure in a supply system within any selected limits found necessary to prevent backflow through its supply branches. The volume rate of evacuation of a supply system will be exactly equal to the volume rate of outflow (displacement) of water from the system and can be computed from the sectional area of the pipe and the velocity of outflow, provided these factors are known. While it is not possible to predict definitely the maximum velocity of outflow that will take place in particular supply systems, it is possible to determine from simple hydraulic principles a maximum limit of velocity in terms of the diameter of the pipe through which the outflow takes place, which limit is unlikely to be reached and will not be exceeded in any system under actual service conditions. These hydraulic principles are given in section IV–1 of this paper in connection with the experimental data pertaining to this phase of the problem.

(e) ASSUMPTIONS RELATIVE TO CONDITIONS IN BUILDING SUPPLY SYSTEMS

Since we cannot determine definite limits for the pressure conditions that will occur in particular systems, it becomes necessary to assume that the extreme conditions that can occur in any system may occur in particular systems, in order to determine minimum requirements for the prevention of backflow or backsiphonage for general application to all plumbing systems under all conditions of service. This assumption is equivalent, as will develop later, to the assumption that the maximum effects possible from any vacuum, in combination with the atmospheric pressure acting through the supply openings of the system, may be encountered at some time in any particular up-feed supply system.

2. MAXIMUM EFFECTS OF VACUA AND PRESSURE CONDITIONS IN WATER-SUPPLY SYSTEMS

Acting on the preceding assumptions, we may proceed with an analysis of the effects of vacua as they apply to different types of cross-connections. The maximum or limiting cases are not dependent on direct or simple siphon action, as the term siphon action is commonly understood, but on certain phenomena connected with the maximum rate of air-flow through the various types of supply openings presented in the plumbing systems.

(a) FLOW OF AIR THROUGH ORIFICES

The theory of the flow of gases through orifices at high velocities has been fully developed by other investigators, so the subject may be introduced here by the statement of certain facts proved by experiment and generally accepted.

It is an accepted fact that rate of flow of air through an opening from the free atmosphere at a pressure P_a into an inclosed space (tank or water-supply system) at a lower pressure P_r increases as P_r is decreased until a critical pressure ratio $r_c = P_r/P_a$ is reached, after which the mass rate of flow remains sensibly constant as the ratio P_r/P_a is reduced below the critical ratio. In the range of pressure ratios $r_c > P_r/P_a > 0$, the velocity of flow in the minimum section or vena contracta of the jet is equal to the velocity of sound in air at the pressure and density of the air in the minimum section of the jet.

This phenomenon of critical flow of gases has a direct application in two phases of the problem: (1) The maximum effect it can exert in lifting water across a vertical air gap into a water-supply opening, and (2) the maximum effect it can exert in lifting water in the submerged portion of a partially submerged outlet to a point where it can mix with the stream of air flowing through the unsubmerged part, the maximum effects in both cases being determined by the maximum rate of air-flow.

(b) SAFE AIR GAP

Water may be lifted across a vertical air gap into the water-supply lines by the combined effect of the atmospheric pressure and a vacuum in the supply system, as illustrated in figure 9, in which the glass jar represents an open-top plumbing fixture, the glass tube represents a faucet spout and the evacuated tank to which the tube is connected represents an evacuated water-supply system. The safe air gap will be defined as the minimum vertical distance x across which water cannot be lifted by the effects of atmospheric pressure in combination with the vacuum in the tank (see fig. 9). The problem is to determine the minimum safe gap x in terms of the dimensions of the faucet for the maximum (critical) flow of air through the faucet.

(c) SAFE PARTIALLY SUBMERGED SUPPLY OUTLETS

Again the maximum (critical) backflow of air into the supply system serves as a means of determining the safety of a partially submerged supply outlet under all possible service conditions. (See fig. 5.) In this connection, a relation or principle of arrangement that applies to and determines the limit of effectiveness of all partially submerged supply outlets may be pointed out. This arrangement

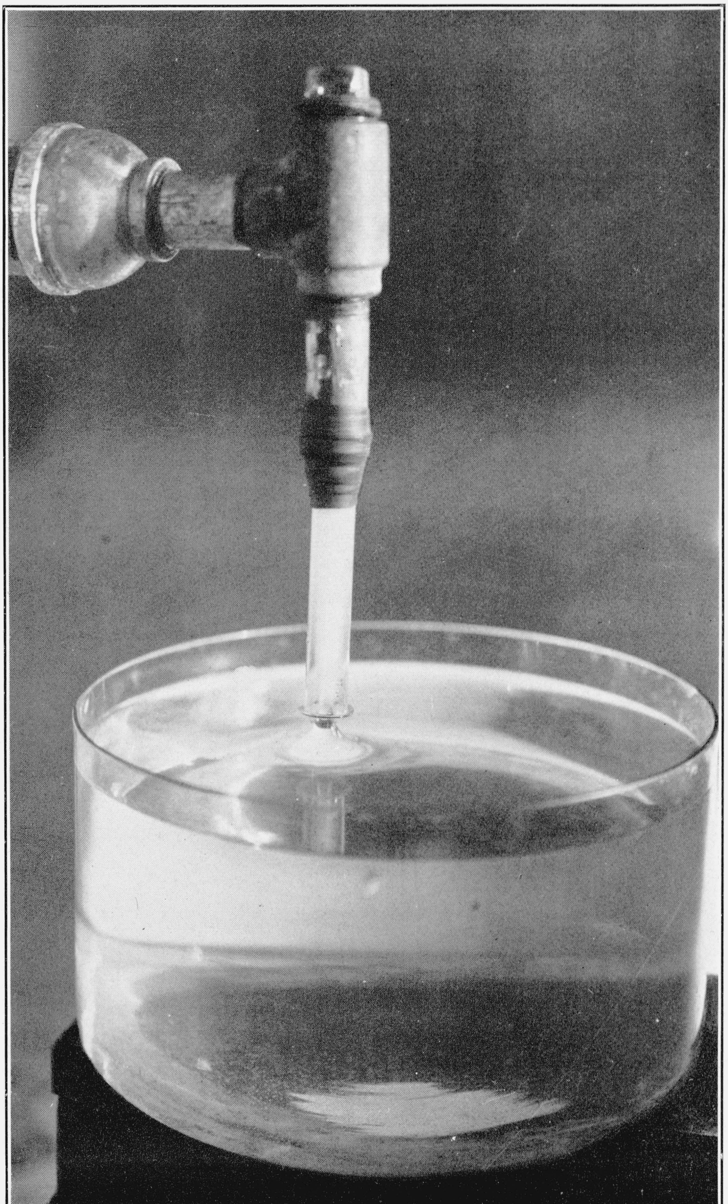

FIGURE 9.—*Backflow across an air gap.*

may be described as two orifices in series, one an inner passage leading from the flush chamber to the supply branch, indicated in figure 18 by the symbol i, and the other an outer orifice leading from the flush chamber to the outer air, indicated by the symbol O, either or both of which orifices may be a single or multiple passage. There is also a third passage leading from the fixture to the flush chamber, this one submerged, through which backflow of water may take place under certain conditions.

Practically any indirect cross-connection may be considered as a modification of the arrangement described. In the case of a jet water closet with flush-valve supply, the passage through the flush valve forms the inner orifice, the rim ports the outer orifice, and the jet passage the submerged third opening. In the case of an air gap, the passage through the valve is the inner orifice, the passage over the water surface to the faucet spout the outer orifice, and the space over the water surface directly under the faucet spout corresponds to the submerged inlet.

The special application of this combination of two orifices in series and a submerged outlet which is of immediate interest and which lends itself to a definite analysis is the device commonly known as a siphon-breaker or vacuum-breaker. The effectiveness of the device in the prevention of backflow from a plumbing fixture is determined by the relative cross sections of the inner and outer orifices and the height to which water can rise in the submerged outlet before it mixes with the air stream flowing back into the water-supply system. Effectiveness is defined, and methods of determining the effectiveness of the device are given in the next section of the paper.

IV. MAXIMUM EFFECTS OF VACUA AND MINIMUM REQUIREMENTS FOR PREVENTION OF BACKFLOW

The preceding section of this paper contains a descriptive analysis of the conditions that may occur in water-supply systems in respect to their limits, and of the effects of these conditions in producing backflow or backsiphonage. This section gives a more detailed theoretical and experimental analysis of a number of special phenomena and physical problems associated with the general problem of preventing backflow, including: (1) The limit in rate of evacuation in water-supply lines; (2) safe air gaps for prevention of backflow; (3) effectiveness of siphon-breakers in preventing backflow; (4) classification of commercial siphon-breakers; (5) tests for effectiveness; and (6) stability of flush valves as related to backsiphonage. As previously stated, these different phenomena and the problems growing out of them are related mainly through their association in the plumbing systems and are, therefore, treated here as separate problems. The reader who is already familiar with or who does not care to give the time to a critical reading of the mathematical analyses of these problems may pass immediately to the summary of the results at the beginning of section V without breaking the continuity of the descriptive analysis of the problem as a whole.

1. RATE OF EVACUATION—MAXIMUM LIMIT

As stated in Section III of this paper, the evacuated volume and the rate of evacuation at any time in a water-supply system are exactly equal to the volume of water displaced from the system and to the volume rate of displacement of the water, respectively. It is impossible to compute the maximum rate of displacement that can occur in any particular system, because of its complexity and because of the impossibility of determining the minimum pressure that will occur in that system. However, if we know the diameter of the main water-supply pipe (service pipe) of the building, we can determine by applying simple hydraulic principles an upper limit to the possible volume rate of displacement in terms of the pipe diameter, as will be shown in the following section. The maximum velocity of evacuation thus determined will not be exceeded in any building water-supply system whatever.

(a) TERMINAL VELOCITY

In any piping system filled with a liquid at rest, if we suddenly apply a definite pressure difference to the system and maintain it unchanged, the liquid will be accelerated, and the mean velocity of flow will approach asymptotically a constant value, reaching it when the frictional resistance opposing the flow becomes equal to the forces—pressure and gravitational—tending to produce flow.

As will be shown presently, the particular case of pipe flow in which we are interested here is that of flow out of a vertical pipe open at the top, the pipe being filled initially with water at rest, after which a valve at the bottom is opened suddenly, and the water is allowed to flow out freely into the atmosphere under the action of gravity. When the valve is opened, the column of water accelerates until, if the pipe is long enough, it attains a constant velocity for which the frictional force opposing flow is equal to the gravitational force producing flow, and falls thereafter at this constant velocity until the pipe is empty. This maximum velocity of fall attained by the water column will be called the "terminal velocity" for the pipe. It will be shown later that the maximum rate of evacuation of a water-supply system in a building is determined by the terminal velocity for the pipe used in the system. Correspondingly, the distance that the water falls in the pipe before attaining the terminal velocity will be called the "terminal length" of the pipe for the assumed conditions.

(b) METHOD OF DETERMINING TERMINAL VELOCITIES IN BUILDING SUPPLY SYSTEMS

The customary rational formula for computing the velocity of flow in a pipe line may be expressed in the form:

$$\frac{H}{L}=\lambda\frac{l}{d}\frac{v^2}{2g}, \tag{1}$$

where

H=the loss of head in the length L feet of pipe, measured in feet of the liquid flowing,

d=the diameter of the pipe in feet,

v=the mean velocity of flow in ft/sec,

g=the acceleration of gravity in ft/sec^2, and

λ=a dimensionless friction factor that depends only on the Reynolds number for smooth-walled pipe.

This equation enables us to compute the loss of head due to friction in a straight pipe of uniform diameter d and of length L between any two measuring sections between which the mean velocity is v, provided we know the correct value of λ that should be used. Or conversely, if we have a definite length of pipe L, and if there is available a definite head H to produce flow, we can compute the velocity that will result. It is clear from the equation that, the greater the value of the ratio H/L (called the "hydraulic gradient"), the greater will be the velocity of flow in the system, and our problem is to determine what conditions that may be encountered in building supply systems will produce the highest velocity of evacuation of the system; or, in other words, the greatest value of H/L that may occur under service conditions.

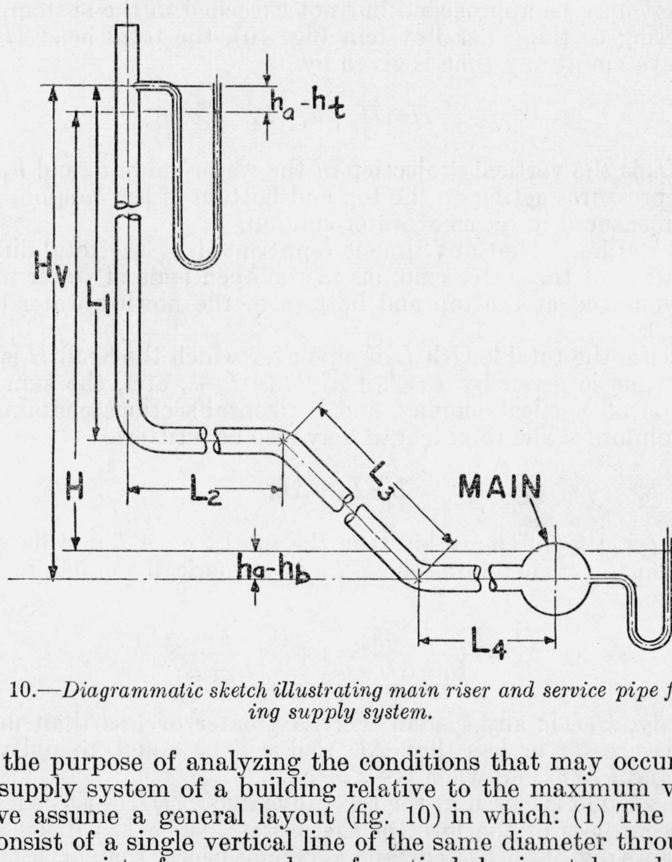

FIGURE 10.—*Diagrammatic sketch illustrating main riser and service pipe for building supply system.*

For the purpose of analyzing the conditions that may occur in the water-supply system of a building relative to the maximum value of H/L, we assume a general layout (fig. 10) in which: (1) The system may consist of a single vertical line of the same diameter throughout, or it may consist of any number of vertical, horizontal, and sloping sections of the same diameters connected by elbow bends; (2) the pipe may be either open or closed at the top and may be open at the bottom or connected by a service pipe to a street main or other source of supply; and (3) the pipes are originally filled with water.

If we now assume a sudden and complete failure in the supply pressure produced by a break in the street main or service pipe, a shutdown in a pumping system, excessive water demands on the main supply, or other comparable cause, water will flow from the system, accelerating until it approaches the terminal velocity for the system, as given by eq 1.

We note from eq 1 that the velocity v varies directly as the square root of the hydraulic gradient H/L and the diameter d of the pipe and inversely as the square root of the friction factor λ. Perfectly smooth pipe has the lowest possible friction factor, and therefore smooth pipe will yield the highest terminal velocity, the factors H/L and d remaining constant for the comparison. Similarly, the velocity v increases as d increases; and, therefore, if the building supply piping is not greater in diameter than the service pipe, the usual form of installation, the terminal velocity determined from the maximum possible value of H/L for a system of the same diameter as the service pipe will set a limit that may be approached but not exceeded in the system.

Referring to the general system (fig. 10), the total head H acting on the system at any time is given by

$$H=H_v+h_t-h_b, \tag{2}$$

where H_v is the vertical projection of the water column, and h_t and h_b are the pressures acting on the top and bottom of the column, respectively, measured in terms of water column.

The total head H at any time is represented by the total difference in elevation of the water columns in the open ends of water manometers connected at the top and bottom of the flowing water column (see fig. 10).

Likewise, the total length L of pipe over which the head H is acting at any time is given by $L=L_1+L_2+L_3+L_4+$, etc., the sum of the lengths of all vertical, sloping, and horizontal sections containing the water column. The total length may also be written

$$L=L_v+\Delta L,$$

where ΔL is the difference between the total length L and its vertical projection L_v. Therefore, since L_v is numerically equal to H_v by definition,

$$\frac{H}{L}=\frac{L_v+h_t-h_b}{L_v+\Delta L}=1+\frac{(h_t-h_b-\Delta L)}{L_v+\Delta L} \tag{3}$$

Obviously, H/L in any system will be greater or less than unity as h_t-h_b is greater or less than ΔL, and will be equal to unity when $\Delta L=0$ and $h_t=h_b$ and when $h_t-h_b=\Delta L$.

If the system is closed at the top (unvented), H/L must be less than unity regardless of the form of the system, since h_t will equal h_e if water flows out of the system, and h_b cannot be less than h_e.

If the system is open at the top (completely vented), again H/L must be less than unity, unless h_t-h_b is numerically greater than ΔL. Since h_t must be sensibly less than atmospheric pressure (h_a=34 feet of water column, approximately), and h_b cannot be less than h_e, ΔL must be considerably less than 34 feet in any system, to make $H/L=1$. In the practical case, ΔL will include the offset of the main riser from the street water main, which will usually in itself be greater than 34 feet. Furthermore, the loss of head due to the flow of the water around the bends in the system is very considerable at velocities approaching the terminal velocity in the system and has the same effect on the value of H/L as increasing ΔL.

The maximum possible value of H/L would occur when $\Delta L=0$, a value given only by the impractical case of a completely vertical vented system connected directly to a water main without offset at its base. Even in this case, h_t-h_b will be less than h_a-h_e, since h_t must be less than h_a, and h_b cannot be reduced to h_e with water flowing from the building supply system into the water main at a high velocity. Furthermore, if the piping system is long enough for the velocity of evacuation to approach closely the terminal velocity for a hydraulic gradient of $H/L=1$, any possible value of h_t-h_b will be negligible compared with the projection of the water column L_v, since h_t must decrease as the velocity of evacuation and length of evacuated pipe increase, and at the same time the value of h_b will tend to increase.

In consideration of this analysis, it seems perfectly safe to assume that the velocity of evacuation of any actual building water-supply system under any possible service conditions will not exceed the terminal velocity for a smooth pipe having the same diameter as the service pipe of the system and a hydraulic gradient of $H/L=1$.

(c) DETERMINATION OF TERMINAL VELOCITIES FOR VERTICAL PIPES

The assumed upper limit of velocities of evacuation of water-supply systems will be given by eq 1, taking $H/L=1$, and writing v_t for v:

$$\frac{H}{L}=\lambda\times\frac{1}{d}\times\frac{v_t^2}{2g}=1 \tag{4}$$

The obvious method of determining v_t is to solve this equation for v_t, provided the corresponding value of λ can be obtained. The value of the friction coefficient, λ, which varies with the velocity, has been accurately determined as a function of the Reynolds number R_e for smooth-walled pipe and is known approximately for some classes of rough pipe.

The Reynolds number is defined as the dimensionless product

$$R_e=\frac{dv\rho}{\mu}=\frac{dv}{\nu}, \tag{5}$$

where

ρ= the density of the fluid flowing,
μ= its absolute viscosity, and
ν= its kinematic viscosity ($=\mu/\rho$).

A unique relation exists between λ and R_e for the flow of an incompressible viscous fluid through a smooth pipe, which can be expressed mathematically as

$$\lambda=\text{function}\left(\frac{dv}{\nu}\right). \tag{6}$$

This means that the relation between λ and the Reynolds number can be represented by a single curve for smooth-walled pipe. Figure 11 shows this relation between the friction factor and the Reynolds number dv/ν in the turbulent region for smooth pipes and is based on many experiments with different fluids [20–21].

(1) *Terminal velocities in smooth pipes.*—The terminal velocities for smooth-walled pipes 1, 2, 3, 4, and 6 inches in diameter were computed from eq 4, using values of λ taken from figure 11. Since v_t appears in the Reynolds number, the equation cannot be solved directly for v_t, and it is necessary to use the method of successive approximations. A value of v_t was assumed for a trial computation of the Reynolds number, and the corresponding value of λ was read from figure 11. Using this value of λ, a new value of v_t was

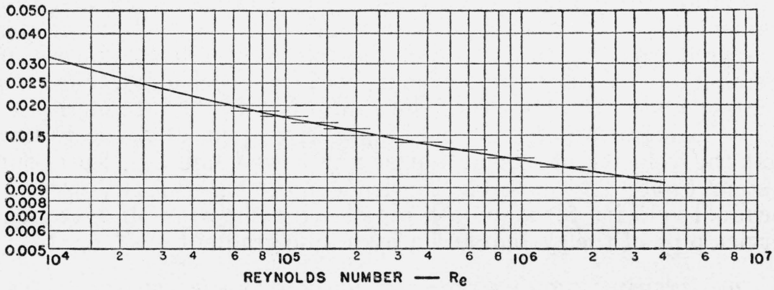

FIGURE 11.—*Friction factor* λ *as a function of the Reynolds number.*

computed by this equation. This new value of v_t was then used to compute a new Reynolds number, and the process was repeated until the last two values of v_t computed from the equation agreed with each other within the accuracy with which λ could be read from the curve. These computed terminal velocities for vertical smooth pipe from 1 to 6 inches in diameter are given, together with other pertinent data in table 1 and are plotted in figure 12.

TABLE 1.—*Terminal velocities, terminal lengths, and maximum rates of displacement for smooth, vertical pipes*

Diameter		Volume per foot length of pipe		Terminal velocity	Rate of displacement		Length required to attain 0.99 terminal velocity	
Nominal	Actual						Approx. (eq 13)	Exact method
in.	in.	cu ft	gal	fps	cfs	gpm	ft	ft
1	1.049	0.006	0.045	18.45	0.111	49.8	20.8	21.9
2	2.067	.023	.172	28.5	.664	299	49.2	52.6
3	3.068	.051	.382	37.3	1.91	861	84.8	88.4
4	4.026	.088	.659	44.3	3.92	1,760	119.4	124.1
6	6.065	.201	1.50	57.2	11.5	5,160	198.5	204.0

(2) *Terminal velocities in rough pipes.*—The $\lambda - R_e$ relation for rough pipes is not a single curve, as represented in figure 11 for smooth pipe. For any given class of rough pipes, a separate curve is obtained for each diameter of pipe, giving an approximately parallel series of curves covering a band in the $\lambda - R_e$ diagram. The reason that the curves for pipes of different sizes do not coincide is that the relative hydraulic roughness varies with the diameter. For these reasons, the loss in head per unit length of pipe for a given velocity of flow cannot be

predicted for rough pipe from past experience as accurately as for smooth pipe. However, the $\lambda - R_e$ curves for very rough pipe show one characteristic not shown by the curves for smooth pipe; the curves become horizontal for high Reynolds numbers, yielding a constant value of λ in the region of high velocities.

As a matter of general information, the terminal velocities in a number of sizes of very rough pipe were computed from values of λ taken from experiments on rough pipe [22] and are plotted in figure 12.

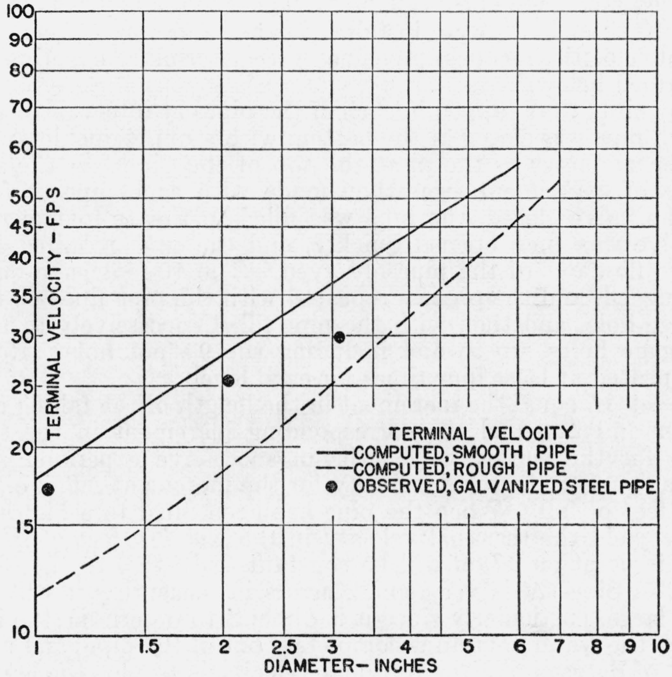

FIGURE 12.—*Terminal velocities for pipes 1 to 6 inches in diameter.*

(3) *Terminal velocities in brass, copper, and steel pipes.*—The friction factors for smooth copper or brass tubing with streamline fittings will differ but slightly from the values given in figure 11 for smooth pipe. Threaded brass or steel pipe may be expected to give $\lambda - R_e$ curves lying above the curve for smooth pipe; but, if the joints are carefully made, the curves will approach very closely the curve for smooth pipe. Experiments with steel pipe by different experimenters show a considerable variation in the $\lambda - R_e$ curves, probably in part owing to differences in installation and in the technique of measurement. However, the $\lambda - R_e$ curves for some of the more recent experiments on 4-inch galvanized-steel pipe[4] lie close to the curve for smooth pipe, indicating the probability that a smoother grade of pipe is now being manufactured than in the past. (See also reference 23.)

[4] Unpublished data by the authors.

(d) EXPERIMENTAL DETERMINATION OF TERMINAL VELOCITIES IN VERTICAL GALVANIZED-STEEL PIPES

The experimental determination of terminal velocities in open-ended vertical pipes requires long vertical pipes, and the existence of a plumbing tower 10 stories high at the National Bureau of Standards made the erection of such test lines for a limited range of diameters a simple matter. Hence, to demonstrate the validity of the statement made earlier in this paper that the rational pipe-flow equation could be used to compute terminal velocities in vertical pipes, galvanized-steel pipe lines, 1, 2, and 3 inches in diameter and approximately 100 feet high, were installed in the plumbing tower, and the terminal velocities for these pipe lines were determined experimentally, as described below.

Gage points were tapped in each of the pipes at intervals of 10 feet, and each pipe was closed at the bottom with a quick-opening valve of the same diameter as the pipe, the top of the pipe being left open. A series of experiments was then made with each pipe as follows: With the valve closed, the pipe was filled to the 10-foot gage hole. The valve was then opened quickly, and the time required for the water to flow out of the pipe observed. The 10-foot gage hole was then plugged, and the process repeated with the pipe filled to the 20-foot gage hole, and then with the pipe filled successively to the different gage holes, up to and including the 90-foot hole. Readings were repeated at least four times for each level.

If we let ΔL equal the increment in the length of the falling column of water and Δt equal the corresponding increment in the time of descent for the successive lengths in successive experiments, then $\Delta L/\Delta t$ is equal to the mean velocity for the increment ΔL, i. e., in the last 10 feet of fall. When the pipe has been filled to a high enough point to yield the terminal velocity in the last few feet of fall, then from this point on $\Delta L/\Delta t$ will be constant and equal to the terminal velocity. Since Δt is very small, errors in measuring it will be relatively large, particularly since it is difficult to determine the instant at which the water column is completely out of the pipe; and a better method of determining the terminal velocity is to plot the lengths of the descending columns as ordinates and the time of descent as abscissas. This will give a straight line in the region where the column is flowing at the terminal velocity, and the slope of the line will give the terminal velocity.

The terminal velocities determined in this way are plotted in figure 12 for comparison with the results computed for smooth pipes and for one class of very rough pipes, as described in sections IV–1–(c)–(1) and (2) of this paper.

Since these experimental values for galvanized pipe fall between the terminal velocities for smooth pipe and rough pipe, as they should, and since the problem is merely to set a limit for the maximum rate of evacuation of water-supply systems, it will be assumed that the velocity of evacuation in any practical water-supply system cannot exceed the terminal velocity of flow in a vertical smooth pipe of the same diameter as the service pipe of the water-supply system in question. The smooth-pipe curve in figure 12 can then be used to determine the terminal velocities for vertical pipes over the range of diameters given.

(c) TERMINAL LENGTHS

In addition to knowing what terminal velocities can be attained in vertical pipes of various diameters, it is important to ascertain whether the buildings in which vertical supply lines are constructed are high enough to permit these terminal velocities to be attained. This can be done with the aid of an equation developed in an earlier publication [24]. In deriving this equation, it is assumed (1) that the vertical pipe is wide open at both ends, so that atmospheric pressure (approximately) acts on the top and the bottom of the column making $h_t=h_b$ as nearly as possible, (2) that the cross section of the pipe remains full during the flow, and (3) that the resistance to flow varies with the square of the velocity. The following equation, giving a relation between the velocity attained by the falling water column as a function of the height through which it has fallen, was given in the earlier publication [24]:

$$v=\left(\frac{g}{k}-\frac{\frac{g}{k}-v_0^2}{e^{+2kS}}\right)^{1/2}, \tag{7}$$

where

v = the velocity attained after a fall through the vertical distance S,
v_0 = the initial velocity (when $S=0$),
g = the acceleration of gravity,
e = the base of the Naperian system of logarithms (=2.7183), and
k = a friction factor which depends on the diameter and roughness of the pipe.

In the case that we have to consider, the water column starts from rest; hence $v_0=0$, and we can write eq 7 as

$$v=\left(\frac{g}{k}\right)^{1/2}(1-e^{-2kS})^{1/2}. \tag{8}$$

Obviously the velocity of flow approaches the terminal velocity asymptotically; and, strictly speaking, the water column would have to fall through an infinite distance before it would attain its terminal velocity. Actually the velocity of fall approaches the terminal velocity so rapidly that the latter is reached, as nearly as can be measured, in a comparatively short length of pipe. An approximate value of this length for any given diameter of pipe can be obtained from eq 8 in the following manner. If we substitute S=infinity in the equation, the exponential term becomes zero and we have for the terminal velocity

$$v_t=(g/k)^{1/2}. \tag{9}$$

Now, if we agree arbitrarily to assume that the terminal velocity has been reached when the term $(1-e^{-2kS})^{1/2}$ in eq 8 has reached some value slightly less than 1, say 0.99, we can compute from this equation a finite value of S for a given value of k. Since values of k for different pipe diameters and roughnesses are not available, however, it will be convenient to replace k in terms of the pipe diameter d and the commonly used dimensionless friction coefficient λ, for which values are readily available for many kinds of pipe, particularly smooth pipe. From eq 9 we have

$$kv_t^2/g=1, \tag{10}$$

and, equating the left member of this last equation to the left member of eq 4, there results:

$$k=\lambda/2d, \tag{11}$$

and this value of k can be substituted back in eq 8 to give this equation the more convenient form

$$v=(2gd/\lambda)^{1/2}(1-e^{-\lambda S/d})^{1/2}. \tag{12}$$

Approximate values of the terminal lengths for smooth pipes ranging from 1 to 6 inches in diameter have been computed from the relation

$$(1-e^{-\lambda S/d})=0.99,$$

arranging this in the more convenient form

$$S=3.913\frac{d}{\lambda}, \tag{13}$$

and using a value of λ corresponding to the terminal velocity for the pipe in question. The values thus computed are given in table 1.

This method of computing terminal lengths is not exact, since λ varies considerably during the acceleration of the water column. Hence, although it is not important to determine these lengths with great accuracy, they were recomputed by a more nearly exact method, based on the differential equation from which eq 7 was derived, taking into account the variation of λ. These values of the terminal length are also given in table 1, and it will be observed that they are a little larger than the values yielded by eq 13.

This latter method of computation affords a simple and accurate method of computing the entire fall curve (distance through which the column has fallen plotted against the instantaneous velocity of fall). Consequently, these curves were computed for smooth pipes 1, 2, 3, 4, and 6 inches in diameter, in order to illustrate the rapidity with which the column of water approaches its terminal velocity. These curves are given in figure 13, together with the experimental data obtained for 1-, 2-, and 3-inch galvanized-steel pipe, as described in section IV–1–d of the paper.

The points plotted for these last three pipes do not represent the actual experimental data. As has already been explained, it was impossible to avoid making relatively large errors in measuring the time of fall when the height of fall was not very large. Consequently, the experimental data could not be relied on to give the form of the fall curves during the first part of the fall of the column. Hence this portion of the fall curve for each of the three pipes was computed by a method of approximations, working backward from the higher portion of the curve, which was obtained experimentally with reasonable accuracy. The points plotted are those for which the above-men-

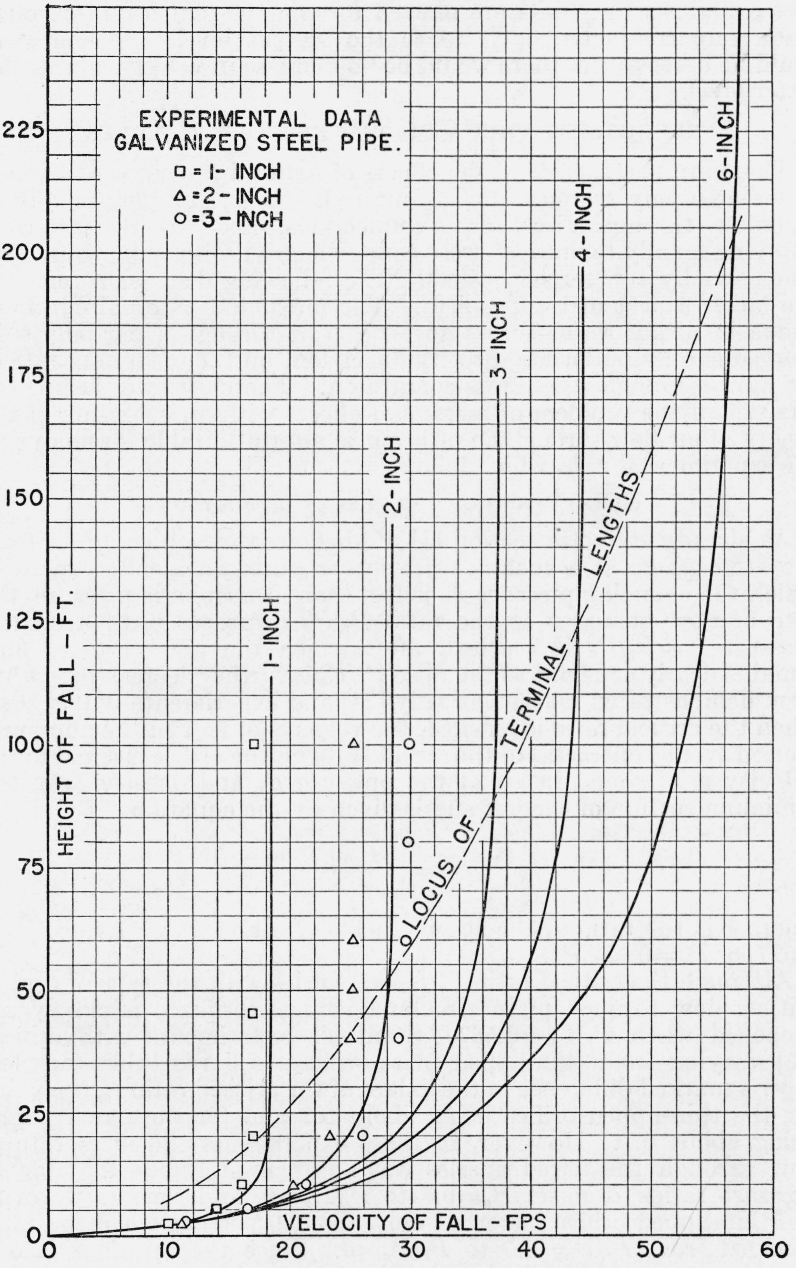

FIGURE 13.—*Fall curves for water columns in smooth vertical pipes 1 to 6 inches in diameter.*

tioned computations were made and yield curves that coincide accurately with the higher portion of each corresponding fall curve. This fact explains why points are plotted for the 100-foot level, although tests were conducted only up to the 90-foot level. As nearly as could be determined, there would be no increase in velocity above this latter level.

2. EFFECTS OF VACUA ON WATER-SUPPLY SYSTEMS

There are two aspects of the effects of reduced pressures and vacua in water-supply systems (1) The internal effects, applying mainly to backflow through direct cross-connections, and (2) the external, applying mainly to indirect cross-connections with inadequate air gaps and partially submerged outlets. The following discussion refers to the latter aspect of the problem. The maximum external effects of vacua in lifting water across an air gap or through a partially submerged supply outlet are directly dependent on the maximum rate of air flow through the cross-connection. Therefore we begin the analysis of the problem of maximum effects with an extension of the theory of air-flow through an orifice into a form suitable for analyzing the experimental data which follow.

(a) EXTENSION OF THE THEORY OF AIR-FLOW

As already stated in section III, if air flows through an orifice from the atmosphere at a constant pressure P_a into an inclosed space in which the changing pressure P_r is less than atmospheric pressure, the rate of flow increases as the ratio P_r/P_a decreases until a critical pressure $r_c=P_r/P_a$ is reached, after which the mass rate of flow remains nearly constant as the ratio P_r/P_a is further decreased. It has been demonstrated mathematically [24] and experimentally [25] that, when the critical ratio is reached, the velocity of flow in the minimum section (vena contracta) of the air jet through the orifice is equal to the velocity v_s of sound in air at the pressure P_c and density ρ_c in the minimum section of the jet, and is given by the equation

$$v_s=\sqrt{\frac{P_ck}{\rho_c}},$$

where k is the ratio of the specific heats of air. If $k=1.4$, $P_c/P_a=0.527$, or $P_c=0.527\ P_a$.

Although the critical ratio $P_r/P_a=r_c$ varies with the type of orifice, critical flow through plain circular openings (orifices) is closely approached when $P_r/P_a=0.527$. Stanton's experiments with orifices [25] varying from thin-lipped plate orifices to flared tubes (nozzles) show a marked difference in the values of the critical ratio r_c, the value for the thin-lipped orifice being about 0.2 and for the flared nozzle being about 0.7. However, in these experiments the mass rate of flow through the flared nozzles was approximately constant for all pressure ratios from $P_r/P_a=0.7$ to $P_r/P_a=0$, and the rate through the thin-lipped orifice increased only about 6 percent when the ratio changed from $P_r/P_a=0.5$ to $P_r/P_a=0$. Since the openings into a supply system cover an intermediate range between the extreme types used by Stanton, it seemed advisable to study experimentally the characteristics of air-flow through openings such as presented by faucets, flush valves, siphon-breakers, etc., in order to verify the existence of a critical ratio for these types of irregular orifices and to determine its approximate value for each. The data given later show that a critical ratio of $P_r/P_a=0.5$ or greater exists for the various types employed.

The equation for the velocity of a gas expanding through an orifice from a pressure P_a to a pressure P_r is [27]

$$v=\sqrt{\frac{2k}{k-1}\times\frac{P_a}{\rho_a}\left[1-\left(\frac{P_r}{P_a}\right)^{\frac{k-1}{k}}\right]}. \tag{14}$$

This equation is general except for the effect of any shape or size factors relating to the orifice. It may be applied therefore to the special case of air expanding from the pressure P_a (atmospheric pressure) to the pressure P_c, the pressure in the minimum section of the jet through the orifice, and becomes in this case

$$v_c=\sqrt{\frac{2k}{k-1}\times\frac{P_a}{\rho_a}\left[1-\left(\frac{P_c}{P_a}\right)^{\frac{k-1}{k}}\right]}, \tag{15}$$

in which v_c is the local velocity in the minimum section of the jet, this velocity being sensibly constant within the range $0<P_r/P_a<r_c$, the critical ratio for the particular orifice. The equation employed for analyzing the data in the straight-line portion of the curves, that is, in the critical flow range, for flow through irregular openings presented by siphon-breakers, flush valves, etc., may be derived as follows.

Assuming a flow of air from the atmosphere at a pressure P_a through some type of small opening (orifice) into a tank in which the initial pressure is zero, the gas flows at a sensibly constant mass rate until the pressure in the tank reaches a value of $P_r=r_cP_a$, after which the mass rate of flow decreases slowly at first, and then more and more rapidly as the ratio of pressures P_r/P_a approaches unity, where flow ceases entirely. An equation giving the ratio P_r/P_a at any time as a function of the time t is desired.

The mass rate in this range of constant flow is

$$m=CA\rho_c v_c, \tag{16}$$

in which

A=the cross-sectional area of the orifice,
C=the flow coefficient,
m=the mass flow per second, and
ρ_c and v_c=the density and velocity of the air in the minimum section of the jet, respectively.

The factors in this equation are all sensibly constant in the particular range of pressures in which the equation is to be applied, using the term *sensibly* to mean that no variation could be detected with the instruments used in the experiments to observe or measure the quantities involved. Therefore, the total mass of air M_r in the tank at any time will vary directly with the time t and we may write, since V_r, the volume of the receiving tank is constant,

$$M_r=\rho_r V_r=mt=CA\rho_c v_c t. \tag{17}$$

Now from the gas law $PV=MRT$, we may write

$$P_r V_r=M_r R T_r, \tag{18}$$

where P_r, V_r, M_r, and T_r are, respectively, the pressure, volume, mass, and absolute temperature in the tank at any time, and R is the gas constant. We may also write for the same mass at atmospheric pressure P_a:

$$P_aV_a=M_rRT_a. \tag{19}$$

Dividing eq 18 by eq 19:

$$\frac{P_rV_r}{P_aV_a}=\frac{T_r}{T_a}. \tag{20}$$

Also for the same mass,

$$V_r/V_a=\rho_a/\rho_r,$$

and T_r may be written

$$T_r=T_a-\Delta T;$$

from which by substitution in eq 20:

$$\frac{P_r\rho_a}{P_a\rho_r}=\left(1-\frac{\Delta T}{T_a}\right) \tag{21}$$

Multiplying each member of eq 17 by the corresponding member of eq 21:

$$\frac{P_r}{P_a}\times\frac{\rho_a}{\rho_r}\times\rho_rV_r=CA\rho_cv_ct\left(1-\frac{\Delta T}{T_a}\right),\text{ or}$$

$$P_r/P_a=\frac{CA}{V_r}\frac{\rho_c}{\rho_a}v_ct\left(1-\frac{\Delta T}{T_a}\right), \tag{22}$$

Substituting $A=\frac{\pi d^2}{4}$ and for v_c its value from eq 15, where d is the diameter of the orifice,

$$\frac{P_r}{P_a}=\frac{C\pi d^2}{4V_r}\times\frac{\rho_c}{\rho_a}\sqrt{\frac{2k}{k-1}\times\frac{P_a}{\rho_a}\left[1-\left(\frac{P_c}{P_a}\right)^{\frac{k-1}{k}}\right]}\left(1-\frac{\Delta T}{T_a}\right)t. \tag{23}$$

Now

$$\frac{\rho_c}{\rho_a}=\left(\frac{P_c}{P_a}\right)^{\frac{1}{k}}\text{ and }P_a=\rho_agh'_a,$$

where h'_a is the height of a hypothetical atmosphere of uniform density ρ_a giving the pressure P_a. Substituting these values in eq 23, there results

$$\frac{P_r}{P_a}=\frac{C\pi d^2}{4V_r}\sqrt{gh'_a}\sqrt{\frac{2k}{k-1}\left(\frac{P_c}{P_a}\right)\left[1-\left(\frac{P_c}{P_a}\right)^{\frac{k-1}{k}}\right]}\left(1-\frac{\Delta T}{T_a}\right)t \tag{24}$$

or

$$\frac{P_r}{P_a}=r=\frac{C\pi N}{4}\times\frac{d^2\sqrt{gh'_a}}{Vr}\left(1-\frac{\Delta T}{T_a}\right)\times t, \tag{25}$$

where

$$N=\sqrt{\frac{2k}{k-1}\left(\frac{P_c}{P_a}\right)^{2/k}\left[1-\left(\frac{P_c}{P_a}\right)^{\frac{k-1}{k}}\right]}.$$

(b) EXPERIMENTAL STUDY OF THE FLOW OF AIR THROUGH PLATE ORIFICES, SIPHON-BREAKERS, AND FLUSH VALVES

Equation 25 was employed in the study of critical flow through irregularly shaped orifices as follows:

The apparatus used in making these tests consisted of a vacuum tank of 318 gallons capacity, a vacuum pump capable of reducing the pressure in the tank to less than 4 inches of mercury absolute pressure (about 26 in. of mercury below atmospheric pressure) in a reasonably short time, a manometer for measuring the pressure in the tank at any time, a 2-inch gate valve connected to the tank by a short nipple, and fittings for mounting the various orifices. See figures 25 and 26, showing a siphon-breaker mounted on the same fittings.

The tank was evacuated for each test with the gate valve closed and the orifice or other device mounted in the end of a 2-inch pipe. Then the pump was shut off, and the initial pressure P_o in the tank recorded. Next the gate valve was opened, and the time intervals from the opening of the valve to the instants at which the pressure in the tank reached successive selected values between $P_r=P_o$ and $P_r=P_a$ were observed.

From these data there were obtained a series of ratios r_1, r_2, r_3, etc., and the corresponding time intervals t_1, t_2, t_3, etc., required to reach these ratios from the initial condition.

Three circular square-edged orifices of 0.125 inch (0.0104 ft), 0.250 inch (0.0208 ft), and 0.500 inch (0.0417 ft) diameters were drilled in circular disks cut from ⅛-inch brass plate and were tested as just described. The purpose of testing these plate orifices was to provide characteristic curves for a simple, easily reproduced form of orifice with which the curves obtained for the irregular and tortuous passages of siphon-breakers and flush valves could be compared. The data obtained are given in table 2.

TABLE 2.—*Data for thin-plate orifices*

t	Δt	$t_o=t+\Delta t$	$\frac{d^2\sqrt{gh'_a}}{V_r}\times 10^3$	$\frac{d^2\sqrt{gh'_a}}{V_r}\times t_o$	r
$d=0.125$ inch					
sec	sec	sec			
0.0	23.7	23.7	2.35	0.0556	0.029
127.9	23.7	151.6	2.35	.355	.184
239.9	23.7	263.6	2.35	.618	.322
372.4	23.7	396.1	2.35	.930	.454
465.5	23.7	489.2	2.35	1.150	.588
586.8	23.7	610.5	2.35	1.430	.724
730.9	23.7	754.6	2.35	1.770	.860
1,074.0	23.7	1,092.7	2.35	2.580	1.000
$d=0.250$ inch					
0.0	5.7	5.7	9.37	0.0534	0.029
30.8	5.7	36.5	9.37	.342	.184
57.3	5.7	63.0	9.37	.590	.322
84.0	5.7	89.7	9.37	.840	.454
111.6	5.7	117.3	9.37	1.100	.588
142.5	5.7	148.2	9.37	1.390	.724
179.1	5.7	184.8	9.37	1.732	.860
294.0	5.7	299.7	9.37	2.810	1.000
$d=0.500$ inch					
0.0 [a]	1.45	1.45	37.60	0.0545	0.029
7.98	1.45	9.43	37.60	.354	.184
14.66	1.45	16.11	37.60	.605	.322
21.39	1.45	22.84	37.60	.859	.454
28.24	1.45	29.69	37.60	1.117	.588
35.99	1.45	37.44	37.60	1.408	.724
44.37	1.45	45.82	37.60	1.723	.860
75.14	1.45	76.59	37.60	2.880	1.000
$h'_a=26{,}300$ ft. $V_r=42.5$ cu ft.					

[a] These times measured by electric chronograph.

TABLE 3.—*Data for moving-part siphon-breakers*

t	Δt	$t_e=t+\Delta t$	$\frac{d^2\sqrt{gh'_a}}{V_r}\times10^3$	$\frac{d^2\sqrt{gh'_a}}{V_r}\times t_e$	r
Siphon-breaker No. 1. (d_s=0.162 inch)					
sec	sec	sec			
0.0	48.0	48.0	4.02	0.193	0.102
13.0	48.0	61.0	4.10	.250	.132
76.0	48.0	124.0	4.03	.500	.263
142.0	48.0	190.0	3.95	.750	.398
208.0	48.0	256.0	3.91	1.000	.533
276.0	48.0	324.0	3.95	1.280	.668
352.0	48.0	400.0	3.98	1.590	.803
457.0	48.0	505.0	4.09	2.065	.937
488.0	48.0	536.0	4.12	2.210	.961
536.0	48.0	584.0	4.18	2.440	.987
590.0	48.0	638.0	------------	2.880	1.000
Siphon-breaker No. 2. (d_s=0.305 inch)					
0.0	25.0	25.0	13.80	0.345	0.184
11.0	25.0	36.0	13.88	.500	.263
29.0	25.0	54.0	13.88	.750	.398
47.0	25.0	72.0	13.88	1.000	.533
66.0	25.0	91.0	14.06	1.280	.668
87.5	25.0	112.5	14.12	1.590	.803
118.0	25.0	143.0	14.43	2.065	.937
197.0	25.0	222.0	------------	2.880	1.000
Siphon-breaker No. 3. (d_s=0.146 inch)					
0.0	64.0	64.0	3.02	0.193	0.102
99.2	64.0	163.2	3.06	.500	.263
181.5	64.0	245.5	3.05	.750	.398
265.0	64.0	329.0	3.04	1.000	.533
359.0	64.0	423.0	3.03	1.280	.668
466.2	64.0	530.2	3.00	1.590	.803
613.5	64.0	677.5	3.05	2.065	.937
811.0	64.0	875.0	------------	2.880	1.000
h'_a=26,300 ft. V_r=42.5 cu ft.					

TABLE 4.—*Data for nonmoving-part siphon-breakers*

t	Δt	$t_o=t+\Delta t$	$\frac{d^2\sqrt{gh'_a}}{V_r}\times 10^3$	$\frac{d^2\sqrt{gh'_a}}{V_r}\times t_o$	r
Siphon-breaker No. 4. (d_e=0.679 inch)					
sec	sec	sec			
0.0	7.5	7.5	66.60	0.500	0.263
3.9	7.5	11.4	65.80	.750	.398
7.7	7.5	15.2	66.00	1.000	.533
Siphon-breaker No. 5. (d_e=0.465 inch)					
0.0	8.2	8.2	30.50	0.250	0.132
8.3	8.2	16.5	30.30	.500	.263
16.2	8.2	24.4	30.70	.750	.398
24.0	8.2	32.2	31.10	1.000	.533
31.8	8.2	40.0	32.00	1.280	.668
41.0	8.2	49.2	32.30	1.590	.803
53.8	8.2	62.0	33.30	2.065	.937
121.5	8.2	129.7	------------	2.880	1.000
Siphon-breaker No. 6. (d_e=0.365 inch)					
0.0	19.7	19.7	25.40	0.500	0.263
7.4	19.7	27.1	27.70	.750	.398
20.0	19.7	39.7	25.20	1.000	.533
30.8	19.7	50.5	25.30	1.280	.668
42.5	19.7	62.2	25.55	1.590	.803
59.5	19.7	79.2	26.10	2.065	.937
107.5	19.7	127.2	------------	2.880	1.000
h'_a=26,300 ft. V_r=42.5 cu ft.					

TABLE 5.—*Data for stable flush valves*

t	Δt	$t_o=t+\Delta t$	$\frac{d^2\sqrt{gh'_a}}{V_r}\times 10^3$	$\frac{d^2\sqrt{gh'_a}}{V_r}\times t_o$	r
Flush valve No. 1. (d_e=0.29 inch)					
sec	sec	sec			
0.0	18.5	18.5	12.70	0.235	0.125
19.8	18.5	38.3	12.78	.490	.260
40.2	18.5	58.7	12.68	.745	.395
61.5	18.5	80.0	12.42	.996	.526
83.9	18.5	102.4	12.35	1.265	.661
108.5	18.5	127.0	12.35	1.570	.796
143.8	18.5	162.3	12.53	2.035	.931
164.0	18.5	182.5	12.50	2.280	.970
176.0	18.5	194.5	12.38	2.410	.984
Flush valve No. 2. (d_e=0.193 inch)					
0.0	42.5	42.5	5.88	0.250	0.132
42.0	42.5	84.5	5.82	.500	.263
85.3	42.5	127.8	5.83	.745	.395
129.0	42.5	171.5	5.80	.996	.526
165.0	42.5	207.5	6.10	1.265	.661
228.0	42.5	270.5	5.80	1.570	.796
300.0	42.5	342.5	5.94	2.035	.931
405.0	42.5	447.5	6.44	2.880	1.000
Flush valve No. 3. (d_e=0.21 inch)					
0.0	40.0	40.0	5.88	0.235	0.125
43.2	40.0	83.2	5.89	.490	.260
86.7	40.0	126.7	5.88	.745	.395
132.0	40.0	172.0	5.79	.996	.526
180.7	40.0	220.7	5.74	1.265	.661
237.8	40.0	277.8	5.66	1.570	.796
322.0	40.0	362.0	5.62	2.035	.931
368.4	40.0	408.4	5.58	2.280	.970
397.5	40.0	437.5	5.51	2.410	.984
h'_a=26,300 ft. V_r=42.5 cu ft.					

Several siphon-breakers of both the moving-part and nonmoving-part types were tested in the same manner as the plate orifices (see for example, fig. 14). The data for three different siphon-breakers of the moving-part type are given in table 3. Similarly, the data for three different siphon-breakers of the nonmoving-part type are given in table 4. The data for three different flush valves tested are given in

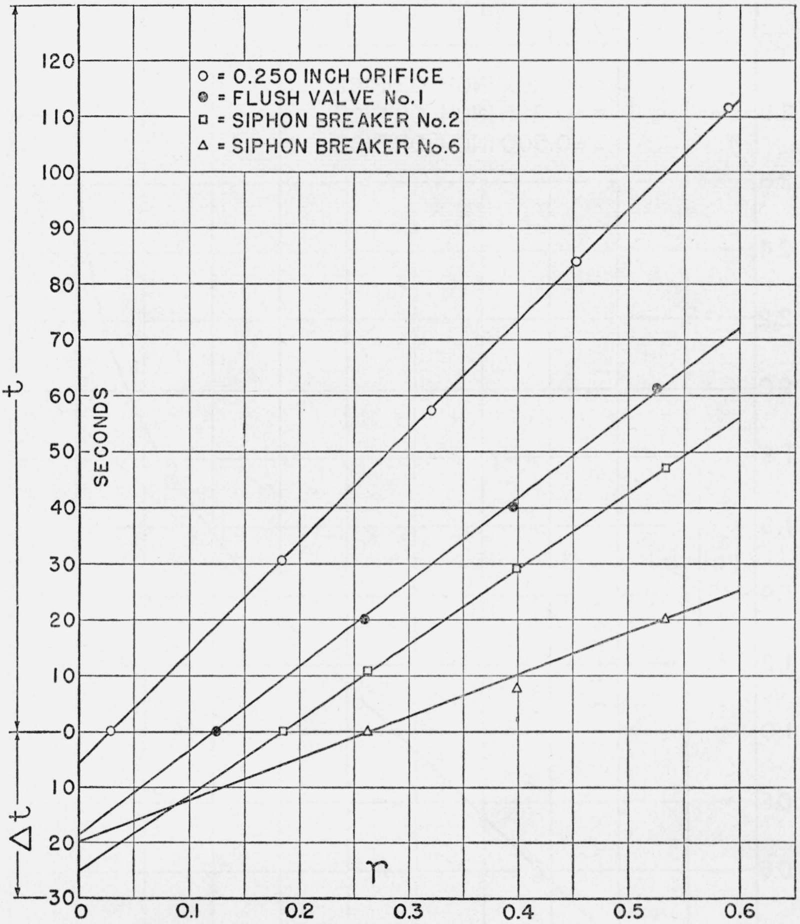

FIGURE 14.—*Air-flow through plate orifice, flush valve, and two siphon breakers in subcritical range of pressure ratio.*

table 5. (See fig. 27 for diagrammatic sketches of typical flush valves.) The data for representative devices from the above classes are plotted in figure 14 with time as ordinates and the pressure ratio r as abscissas.

Figure 14 shows that the data plotted yield straight lines, as nearly as can be determined by visual inspection, up to the highest value of r shown, indicating that the mass rate of flow is sensibly constant over the range of r plotted (subcritical range).

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The plate orifice data for the entire range of r were substituted in eq 25 and were then plotted in figure 15, using the dimensionless variables

$$\frac{d^2\sqrt{gh_a'}}{V_r}\, t_0 \text{ and } r.$$

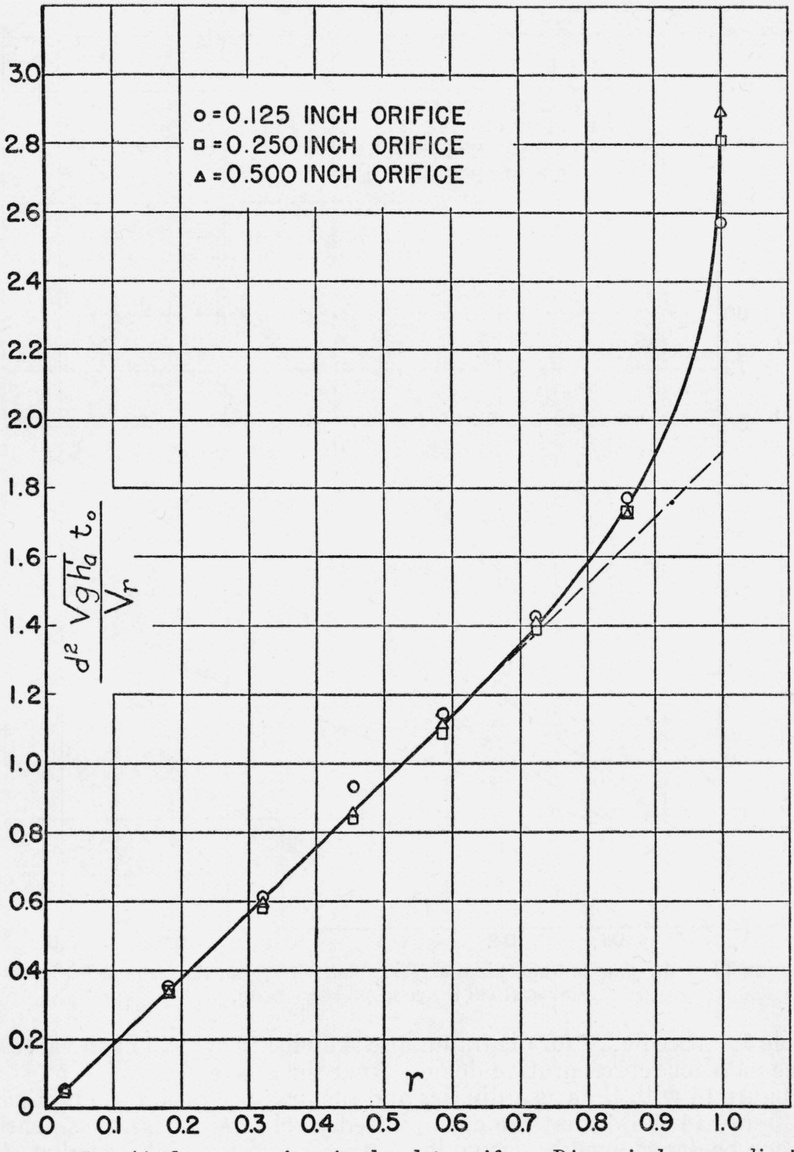

FIGURE 15.—*Air-flow curve for simple plate orifices: Dimensionless coordinates.*

By using these particular variables, the data for the three orifices were brought together in a single curve, instead of yielding three separate curves, as would have been the case had the variables t_0 and r been used, as was done in figure 14.

Before this could be done, however, it was necessary to correct the time interval t corresponding to any particular observed value of r by adding the increment Δt that would have been required to increase the pressure in the tank from absolute-zero pressure to the initial pressure P_0 that existed at the beginning of the run in question. These corrections Δt were obtained from plots similar to that shown in figure 14 by extending the straight lines to cut the axis $r=0$ and reading off the intercepts thus obtained. The time intervals to be used in plotting the following eq 26 corresponding to the observed pressure ratios r_0, r_1, r_2, etc., are then Δt, $t_1+\Delta t$, $t_2+\Delta t$, etc., respectively.

Work by earlier investigators made it appear that it would be legitimate to neglect the temperature change ΔT in eq 25 for the purposes of the present investigation; so, omitting the factor $\left(\frac{1-\Delta T}{T_a}\right)$ and writing t_0 for t, eq 25 becomes

$$r=\frac{C\pi N}{4}\times\frac{d^2\sqrt{gh'_a}\times t_o}{V_r} \tag{26}$$

Figure 15 shows the curve for the plate orifices tested with values of

$$\frac{d^2\sqrt{gh'_a}}{V_r}\times t_0$$

plotted as ordinates and values of r as abscissas. The data used in preparing this figure are given in table 2.

It will be observed that the data for the three orifices fall very closely on the same straight line over the range $0<r<r_c$. Since the coefficient C of the orifice was not taken into account in computing values of

$$\frac{d^2\sqrt{gh'_a}}{V_r}\times t_0,$$

this fact indicates that the three orifices had the same constant coefficient C over this range, within the limits of accuracy of the experiments. The points for the smallest orifice tend to deviate slightly from the straight line given by the two largest orifices, showing that the ratio of the thickness of the plate to the diameter of the orifice may have been a little too great for the smallest orifice, causing it to differ slightly from the characteristic performance of a thin-plate orifice.

(c) EQUIVALENT ORIFICES

It may be useful, especially to the manufacturers of flush valves and siphon-breakers, to make a comparison of the air-flow characteristics of these devices with the characteristics of simple plate orifices. This can be done by means of the curve for plate orifices in figure 15 and the data for siphon-breakers and valves given in tables 3, 4, and 5.

The curve of figure 15 is reproduced in figure 16. If we enter this curve with the value of r for any determination for a particular device, such as a given siphon-breaker, within the range $r<r_c$, we may read the corresponding value of

$$\frac{d^2\sqrt{gh'_a}}{V_r}\times t_0.$$

Knowing the value of t_0 corresponding to the value of r with which we entered the curve, we may now compute the diameter d_e of a circular orifice equivalent to the orifice in the device tested. The equivalence, to be more exact, is between C_1d_1 and C_2d_2, where C_1 and d_1 are, respectively, the flow coefficient of the orifice and the diameter for the plate orifice, and C_2 and d_2 are the corresponding factors for the device, since C_1 and C_2 have not been employed in plotting the curve and probably have different values for the different devices.

Now using the value of d_e thus determined, values of

$$\frac{d^2\ \overline{gh'_a\times t_0}}{V_r}$$

can be computed for the other observations for the device and can be plotted as has been done in figure 16. It is obvious that the flush valve and the two siphon-breakers for which data are given in this figure do have the same air-flow characteristics as a simple plate orifice, so that the performance of each can be expressed in terms of the performance of a plate orifice having the equivalent diameter d_e, the value of which can be determined as explained above.

All of the siphon-breakers and flush valves tested were compared with the plate-orifice curve in figures 15 and 16, and all were found to have the same characteristics as a plate orifice. The data selected for illustration in figure 16 are for three different types of devices—one flush valve, one moving-part siphon-breaker, and one nonmoving-part siphon-breaker.

This relation just established is important, because most flush valves and some siphon-breakers have inner passages or orifices which control the maximum rate of backflow of air but which cannot be measured readily. If the performance curve for a simple plate orifice is available in the form given in figure 15, a single test on a device having such irregular orifice or one of unknown diameter will supply the data necessary for determining the diameter of a simple plate orifice that will have the same capacity.

It will be observed from figure 16 that for the irregular orifices and for the plate orifices the linear relation between r and t_0 extends to a value of r greater than 0.527. The experiments thus appear to justify the assumptions that the mass rate of flow is constant for all pressure ratios between 0 and 0.527, and that this establishes the safe limit for computing the maximum rate of flow through an opening into an evacuated water-supply system and for testing for the maximum external effects of such evacuated systems.

3. SAFE AIR GAPS

Reference has been made in the preceding section to the critical flow of air through an orifice and to the fact that the rate of flow through an opening or orifice in the critical range $0<P_r/P_a<r_c$ remains

sensibly constant throughout that range and is the maximum that can occur. In the problem of safe air gaps, we are concerned only with the maximum effects of the air-flow and may therefore restrict the problem to determining the minimum gap (fig. 17) that will preclude all

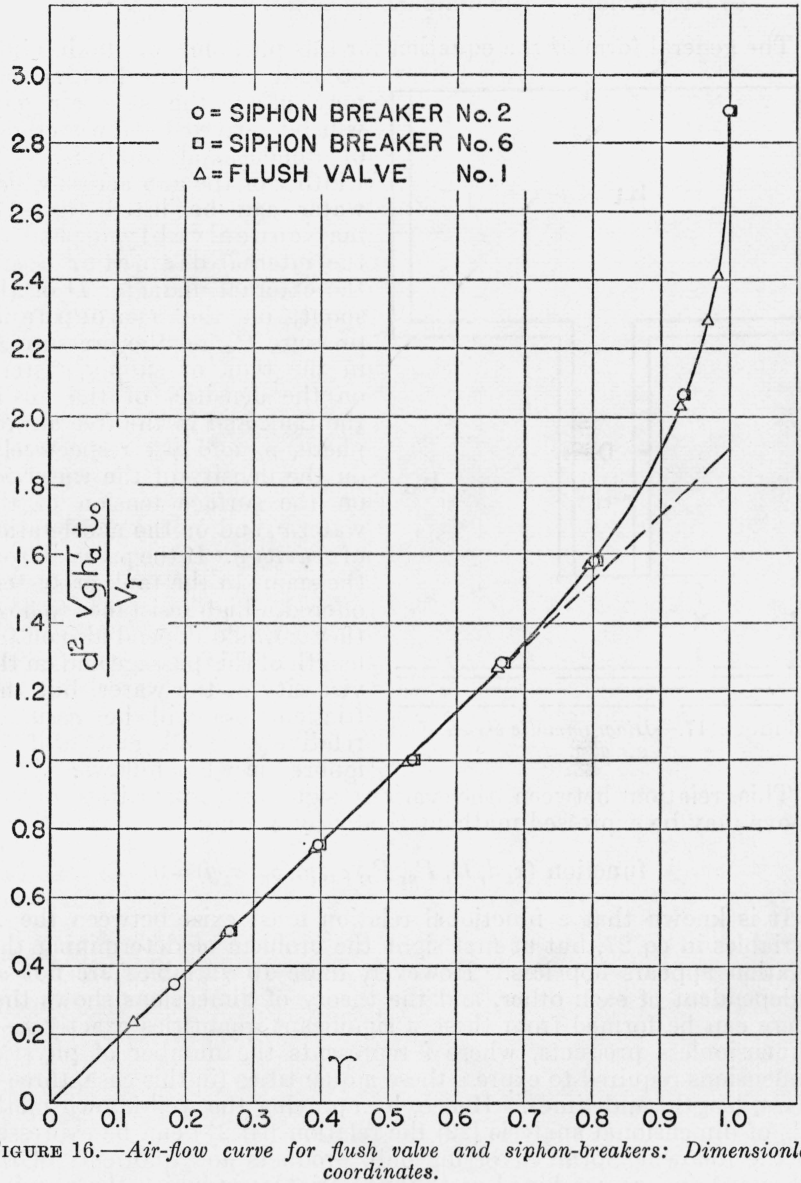

FIGURE 16.—*Air-flow curve for flush valve and siphon-breakers: Dimensionless coordinates.*

possibility of backflow of water under the most unfavorable pressure condition that can occur; that is, when there exists the maximum possible rate of air-flow across the gap. Mr. R. H. Zinkil, of the

Crane laboratories, has published the results of experiments on safe air gaps [28], and these results have been employed here in the derivation of an empirical equation for the safe air gap under any condition that may be encountered.

(a) DERIVATION OF THE GENERAL EQUATION FOR SAFE AIR GAP

The general form of the equation for this phenomenon, taking into account all of the factors that may affect the safe air gap, will be derived by the methods of dimensional analysis. The width x of the gap across which water can be lifted (fig. 17) may conceivably depend on the internal diameter d and the external diameter D of the spout, on the atmospheric pressure P_a, on the pressure P_r in the tank or supply system, on the densities of the air in the tank and in the free atmosphere, ρ_r and ρ_a, respectively, on the density of the water ρ_w, on the surface tension of the water σ, and on the acceleration of gravity g. If the passage from the spout to the tank or system offered a high resistance to flow, then x would depend also on the length of this passage and on the viscosity of the water, but this friction loss will be assumed relatively small and will be ignored in what follows.

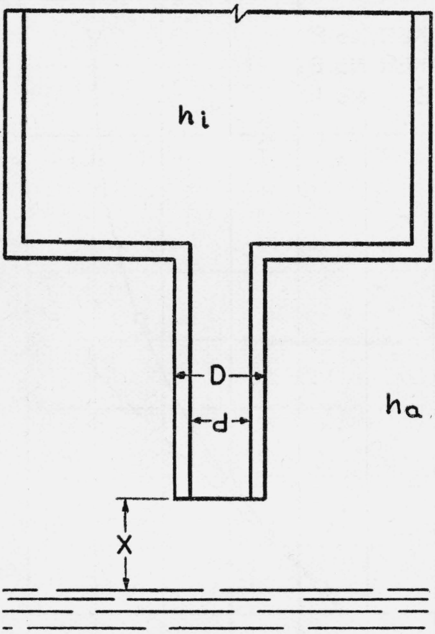

FIGURE 17.—*Diagrammatic sketch of air gap.*

This relation between the various significant quantities named above may be expressed mathematically by writing

$$\text{function}\ (x, d, D, P_a, P_r, \rho_a, \rho_r, \rho_w, \sigma, g)=0 \tag{27}$$

It is known that a functional relation must exist between the 10 variables in eq 27, but at first sight the problem of determining this relation appears hopeless. However, these 10 variables are not all independent of each other, and the theory of dimensions shows that there can be formed from these n significant quantities exactly $n-i$ dimensionless products, where i represents the number of physical dimensions required to express these n quantities (in this case, three—mass, length, and time). Hence, by applying the well-known methods of dimensional analysis [29] the relation (eq 27) can be expressed in the following form, involving only products and quotients of the 10 quantities, so combined as to give variables in which the physical dimensions cancel out, thus yielding a functional relation between $10-3=7$ dimensionless variables:

$$\text{function}\left(\frac{x}{d}, \frac{d}{D}, \frac{d^2\rho_w g}{\sigma}, \frac{d\rho_w g}{P_a}, \frac{\rho_w}{\rho_a}, \frac{\rho_r}{\rho_a}\right)=0 \tag{28}$$

The function has thus been simplified by reducing a relation between 10 quantities to one between 7 quantities, but further simplification is necessary if a simple, usable relation is to be found. The next step is to examine the variables in eq 28 to see whether any of them have no effect on the phenomenon as considered here, or whether they have so small an effect that they can be neglected safely. In the first place, previous experience with such phenomena suggests that surface tension should have only a negligible effect on the phenomenon. Furthermore, the experiments were made with pure water at approximately constant temperature, and under these conditions the variable $\frac{d^2\rho_w g}{\sigma}$ is constant, so that it can be omitted from further consideration. The variable ρ_w/ρ_a can also be omitted from consideration, since it is constant in the phenomenon under consideration, as long as the atmospheric pressure and the temperature of the air and the water remain constant. Finally, ρ_r/ρ_a is omitted from further consideration on the basis that it is a function of the pressure ratio P_r/P_a, which is also involved in eq 28. Thus it is possible to simplify eq 28 to

$$\text{function}\left(\frac{x}{d}, \frac{d}{D}, \frac{P_r}{P_a}, \frac{d\rho_w g}{P_a}\right)=0 \tag{29}$$

For convenience, the variable $\frac{d\rho_w g}{P_a}$ will be expressed in different form. Since $P_a=\rho_w g h_a$, where h_a is the height of a water column of density ρ_w that will exert the pressure P_a at its base,

$$\frac{d\rho_w g}{P_a}=\frac{d\rho_w g}{\rho_w g h_a}=\frac{d}{h_a},$$

and eq 29 can now be written

$$\text{function}\left(\frac{x}{d}, \frac{d}{D}, \frac{P_r}{P_a}, \frac{d}{h_a}\right)=0 \tag{30}$$

Zinkil's [28] experiments on safe air gaps show that x/d increased with decrease in P_r/P_a until P_r/P_a reached the value 0.5, to a rough approximation, and remained constant as P_r/P_a was increased still further. This is due to the fact that the flow of air through the spout increases as the pressure ratio P_r/P_a decreases, until the critical pressure ratio (see section IV-2-(a) of this paper) is reached after which the air-flow remains nearly constant as P_r/P_a decreases still further.

Since the immediate problem is to determine the air gap that is safe under the worst conditions that can occur, it is sufficient to consider only the conditions under which the backflow of air into the spout is a maximum; and it has already been shown that this is the case when P_r/P_a is less than approximately 0.5. Within this range of values of the pressure ratio, x/d has a sensibly constant maximum value. With this restriction on the value of P_r/P_a, it is no longer necessary to treat the pressure ratio as a variable, and eq 30 simplifies to

$$\text{function}\left(\frac{x}{d}, \frac{d}{D}, \frac{d}{h_a}\right)=0, \tag{31}$$

which is simple enough to be fitted to the existing data.

(b) EMPIRICAL EQUATION FOR SAFE AIR GAP

Using Zinkil's data given in table 6 for spouts with a circular cross section, the following empirical equation, which applies within the limits $0.125<d<0.806$ inch, and $0.361<d/D<0.919$, was obtained in terms of the three dimensionless variables in eq 31:

$$x/d=2.45\left(1-0.26\,\frac{d}{D}\right)\left(1-114\,\frac{d}{h_a}\right) \qquad (32)$$

in which x, d, D, and h_a can all be taken in inches, in feet, in centimeters, or in any other unit of length that is desired. One advantage of using dimensionless variables is the fact that the constants in the equation remain the same when a change is made from one system of consistent units to another consistent system. If it is desired to have all these quantities in inches, then, neglecting local variations in atmospheric conditions, h_a may be taken as 408 inches head of water for standard atmospheric pressure at sea level.

This equation may be employed to compute the safe gap x for any faucet within the range of diameters given in table 6, namely, $0.127<d<0.806$ inch and $0.36<d/D<0.92$. These ranges willl include all practical faucets up to an internal diameter of 0.8 inch. However, it is desirable for purposes of construction and inspection to have the minimum permissible gap expressed in a simpler form.

TABLE 6.—*Minimum air gap for various orifices and vacua*

[Values in parenthesis obtained by extrapolation]

Orifices			Minimum air gap x for various vacua						Ratios	
Type of opening	Dimensions		Vacua in inches of mercury						d/D	x/D
	d	D	2	5	10	15	20	Over 20		
	in.	*in.*	*in.*	*in.*	*in.*	*in.*	*in.*	*in.*		
6 in. tube	0.127	0.188	0.16	0.22	0.23	0.24	0.24	0.25	0.675	1.97
Plate orifice	.127	.311	.19	.24	.26	.27	.28	.28	.408	2.20
6 in. tube	.187	.249	.23	.30	.33	.33	.34	.34	.752	1.82
Plate orifice	.187	.501	.26	.32	.36	.38	.38	.38	.374	2.02
6 in. tube	.227	.312	.28	.36	.39	.41	.41	.41	.728	1.81
Plate orifice	.227	.629	.31	.39	.42	.44	.45	.45	.361	2.04
6 in. tube	.403	.501	.50	.59	.66	.66	.66	.66	.806	1.64
Plate orifice	.403	.876	.52	.63	.69	.72	.74	.74	.461	1.84
6 in. tube	.570	.627	.65	.79	.86	.87	(.88)	(.88)	.910	(1.54)
Plate orifice	.570	1.122	.68	.82	.93	.95	(.97)	(.97)	.509	(1.70)
6 in. tube	.806	.873	.86	1.14	1.25	(1.26)	(1.26)	(1.26)	.924	(1.56)
Plate orifice	.806	1.623	.93	1.25	1.34	(1.35)	(1.36)	(1.36)	.497	(1.68)

It will be observed that in eq 32 the value of x/d is increased by decreasing either or both of the ratios d/D and d/h_a in the second member of the equation. Therefore, if a group of faucets falls within definite limits in dimensions, for example, $0.375<d<0.5$ inch and $d/D<0.5$, an approximate value for the safe gap for any faucet in the group, but slightly greater than the minimum safe gap for any particular faucet in the group, may be obtained by substituting maximum and minimum limits in eq 32 and solving it for x. Thus, if 0.5 inch is substituted for d in the ratio x/d, 0.5 for d/D, and 0.375 inch for d and 408 inches for h_a in the ratio d/h_a, the resultant equation is

$$x/0.5=2.45(1-0.26\times 0.5)\left(1-114\frac{0.375}{408}\right), \qquad (32a)$$

from which $x=0.96$ inch approximately. In this manner a safe air gap expressed as a concrete number may be obtained for any limited range in faucet sizes, and this range may be chosen to correspond to the sizes of faucets used with the different kinds of plumbing fixtures, for example, lavatory faucets, sink faucets, bathtub faucets, etc.

Data published in Zinkil's paper, but not tabulated here, indicate that, in the case of oval sections, d and D may be safely taken as the mean of the maximum and minimum diameters of the internal and external sections respectively, and that for faucets set at an angle, x may be taken safely as the mean of the maximum and minimum distances of the spout from the highest possible water surface.

It may also be pointed out that the data given in table 6 show a tendency for the ratio x/d to decrease as d increases. This signifies that a margin of safety will be given by employing eq 32 to compute the safe gap for faucets or other supply outlets larger than 0.8 inch in mean internal diameter.

4. SIPHON-BREAKERS

The principle of siphon-breakers presents another special case of the flow of air through orifices. A siphon-breaker, using the term broadly, includes any device for the prevention of backflow of water through a water-supply branch, the outlet of which is submerged, or may become submerged, under service conditions. Excepting the simple check valve, the principle on which all these devices depend in common for their effectiveness in preventing backflow, is the arrangement of two orifices in series. If a check valve is leaking, the same principle still applies.

The essentials of this arrangement of orifices are illustrated diagrammatically in figure 18. These are: an inner orifice i opening from the supply branch to the flush pipe, and an outer orifice O, opening from the flush pipe to the outer air, either orifice being a single passage or any number of separate passages. If a vacuum occurs in the supply system on which a siphon-breaker is installed, a stream of air will flow through the outer orifice into the flush pipe, then from the flush pipe through the inner orifice into the water-supply system.

The problem consists in determining the relation between the ratio of the cross sections or capacities of the two orifices and the permissible rise of water in the flush pipe (fig. 18) before it can mix with the stream of air and be carried through the inner orifice into the supply branch. The height of rise in the flush pipe will be equal to the difference in pressure Δh between the pressure in the flush pipe and the prevailing atmospheric pressure h_a; hence Δh determines the minimum value that x can have if the siphon-breaker is to be fully effective. Since the mass rate of flow through the two orifices is the same, the maximum value of Δh will occur when the mass rate of air-flow is a maximum. Also since x, the permissible rise, must be small for structural reasons, Δh must also be kept small, and therefore the outer orifice must have a capacity sufficient to hold the loss in pressure head Δh to the value of x or less when the flow through the inner orifice is a maximum; that is, when critical flow through the inner orifice occurs. These relations are given in the paper [10] by Professor Camp previously referred to but were not analyzed in detail for all types of siphon-breakers.

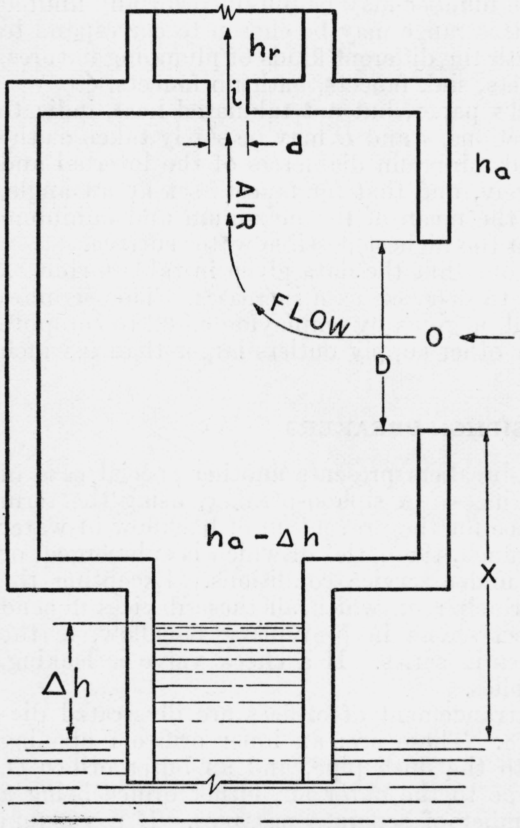

FIGURE 18.—*Diagrammatic sketch of siphon-breaker.*

(a) DEFINITION OF EFFECTIVENESS

We may define the effectiveness of any siphon-breaker as the ratio of the permissible rise of water x in the flush pipe to the maximum possible rise Δh under any vacuum that may occur, or $x/\Delta h$. If $x/\Delta h \gtreqless 1$, the device will be effective under any pressure conditions that can occur in the water-supply system. If $x/\Delta h < 1$, the device will be ineffective under the maximum possible effects of vacuum but may be effective for the conditions that actually occur in particular systems. It may be pointed out that, for any pressure conditions, the margin of safety of the device can be increased either by increasing x, the height it is placed above the water level in the fixture (permissible rise), or by decreasing Δh, the minimum limit of effectiveness, which can be accomplished by increasing the capacity of the outer orifice relative to the capacity of the inner orifice.

(b) DETERMINATION OF EFFECTIVENESS

If the dimensions of the inner and outer orifices of a siphon-breaker are known, so that the ratio of the cross-sectional areas or of the diameters d and D of equivalent circular openings can be determined, it is possible to compute approximately the effectiveness of the device. For this purpose, Δh can be expressed in terms of the diameter ratio d/D by considering the flow through the inner and outer orifices separately. (See fig. 18.) It will be assumed that the value of the pressure ratio

$$P_r/P_i = \frac{h_r}{h_a - \Delta h}$$

for the inner orifice is less than the critical value, $r_c = 0.527$; and, since Δh is very small in comparison with h_a, it will be ignored, so that the pressure ratio for the inner orifice will be taken as $P_r/P_a = h_r/h_a$.

The following equations then apply to the inner orifice, under the assumption that the flow is adiabatic:

$$m=\frac{C_i\pi d^2}{4}\rho_c v_c,\ v_c=\left(\frac{P_c k}{\rho_c}\right)^{1/2},\ \frac{\rho_c}{\rho_a}=\left(\frac{P_c}{P_a}\right)^{1/k},\ \text{and } P_c/P_a=r=0.527$$

where C_i is the flow coefficient for the inner orifice, and the other quantities have already been defined.

Since the pressure drop across the outer orifice will be very small, of the order of one inch of water, it will be assumed that the flow through the outer orifice is isothermal, in order to simplify the computation. The following equations then apply to the outer orifice:

$$v_0=(2g\Delta h')^{1/2},\ \text{and } Q=\frac{m}{\rho_a}=C_0\frac{\pi D^2}{4}v_0,$$

where C_0 is the flow coefficient of the outer orifice, Q is the volume rate of flow, and $\Delta h'$ is the drop across the outer orifice expressed in height of an air column of standard density ρ_a. Utilizing the fact that the mass rate of flow must be the same for both orifices, the following equation can be derived from the equations given above:

$$\Delta h=1/2h_a\times r^{\frac{k+1}{k}}\times k\times\left(\frac{C_i}{C_0}\right)^2\left(\frac{d}{D}\right)^4. \tag{33}$$

If the value 408 inches of water is substituted for h_a, 0.527 for r, and 1.4 for k, eq 33 becomes $\Delta h=95.2\left(\frac{C_i}{C_0}\right)^2\left(\frac{d}{D}\right)^4$ (inches of water). (33a)

Furthermore, if we take the maximum value of unity for C_i and an average value of 0.60 for C_0, i. e., $C_i/C_0=1.67$, then

$$\Delta h=265\ (d/D)^4 (\text{inches of water}). \tag{33b}$$

Values of Δh from eq 33a are plotted against d/D in figure 19 for three values of the ratio C_iC_0; 1.5, 1.75, and 2.0. If the diameter ratio d/D and the value of x are known for the siphon-breaker in question, it is possible to use eq 33a or a plot similar to figure 19, using the proper value of C_i/C_0, to determine the value of Δh.

Conversely, in the design of a siphon-breaker, if the equivalent diameter of the inner orifice is known, the equivalent diameter of the outer orifice that will keep Δh within any desired limit can be determined.

In general, the orifices in siphon-breakers and flush valves will not be simple circular openings. However, as a first approximation, the value of d or D, as the case may be, can be based on a circular area equal to the measured cross-sectional area of the irregularly shaped openings. In case the orifices are circular and are accessible, the diameters d and D can be measured directly. A reason for believing that this procedure is legitimate is the fact that, as figure 16 shows, the devices tested behaved exactly as if the orifices in them were simple circular openings, although actually in some cases they were very irregular and consisted of several parallel small passages.

(c) TYPES OF SIPHON-BREAKERS

Commercial siphon-breakers designed for the prevention of backflow into water-supply branches may be classified into two comprehensive groups: (1) Moving-part siphon-breakers, and (2) nonmoving-part siphon-breakers.

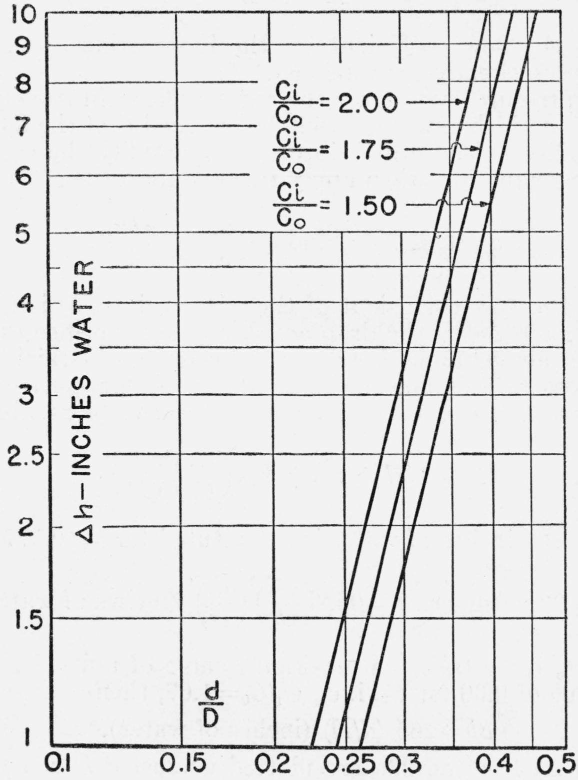

FIGURE 19.—*Rise of water level in flush pipe as a function of the ratio of the diameters of the inner and outer orifices.*

(d) MOVING-PART SIPHON-BREAKERS

Two forms of moving-part siphon-breakers are shown in figures 20 and 21, each of which is designed for insertion in the flush pipe between a flush valve and water closet or other plumbing fixture. The types shown consist of a check valve, through or around which a small opening (inner orifice) is provided, and of a larger opening (outer orifice) from the flush pipe to the outer air. Another suggested form,[5] in which the moving part consists of a single disk and loose-fitting guide, is illustrated in figure 22 and may be constructed in the offset

[5] Since the design for a siphon-breaker of the type illustrated in figure 22 was completed, a sample of one constructed on almost the same pattern has been submitted to the National Bureau of Standards for examination. Another siphon-breaker employing the same principle, but constructed in a form suitable for installation in the supply branch on the supply side of a valve or faucet, was submitted for examination in January 1936, prior to starting this investigation. Therefore, the authors make no claim of originality or priority in developing the principle in a form suitable for use in the flush pipes of flush valves.

form as shown, or in a symmetrical form, giving it an external appearance similar to figure 21.

In operation of the types shown in figures 20, 21, and 22, and others employing the same principle, the check valve remains closed in the protective position when water is not flowing through the device. When the flush valve is tripped, starting the flow of water through the

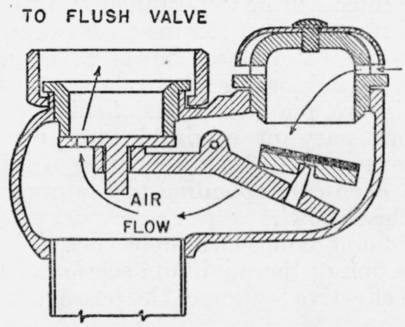

FIGURE 20.—*Moving part.*

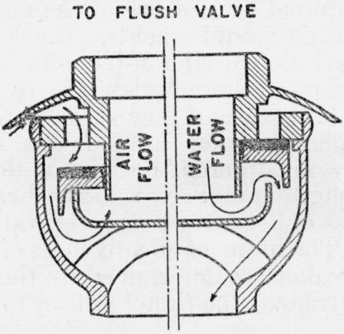

FIGURE 21.—*Moving part.*

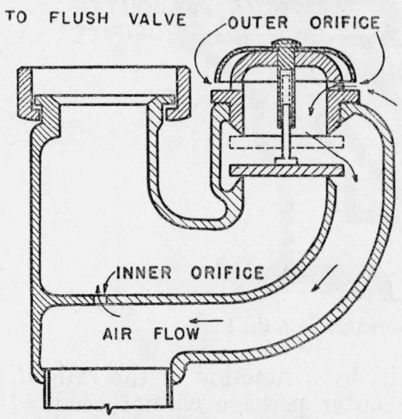

FIGURE 22.—*Moving part.*

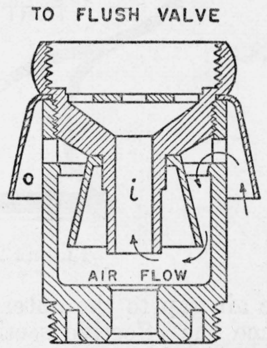

FIGURE 23.—*Nonmoving part.*

FIGURES 20–23.—*Types of siphon-breakers.*

device, the check valve opens, simultaneously closing the outer orifice to prevent an outflow of water through it, and again drops into the protective position when the water stops flowing, the moving part thus being forced through its complete cycle of motion with each flush of the fixture.

(c) NONMOVING-PART SIPHON-BREAKERS

Figure 23 illustrates the essentials of a nonmoving-part siphon-breaker, commonly referred to as the venturi type. It consists of an entrance tube tapering to a minimum section (inner orifice) i, an exit tube enlarging from a section slightly larger than the minimum section of the entrance tube to the same as that of the flush pipe, a definite air gap between the entrance and exit tubes, and an enclosed passage (outer orifice) O, between the air gap and outer air.

In order to function satisfactorily as a part of the flushing mechanism, the device must pass water under the available service pressure at the volume rate or rates required to flush the fixture satisfactorily and deliver the water to the fixture with a velocity head equal to or greater than the loss in head through the fixture, and therefore the required dimensions of the inner orifice will be determined mainly by flushing requirements. Since different types of fixtures; for example, wash-down, siphon-jet, and blow-out water-closet bowls have quite different characteristics in respect to the loss in head relative to the volume rate of flow through the fixture, manufacturers of this type of siphon-breaker have found it necessary or advisable to produce several different designs of the device, each design covering a fairly definite range in velocity heads, each corresponding to the range in loss of head in one type of water-closet bowl.

The limits of effectiveness of the nonmoving-part device as a siphon-breaker are determined by the relation of the minimum section of the entrance tube (inner orifice) to the effective section of the passage from

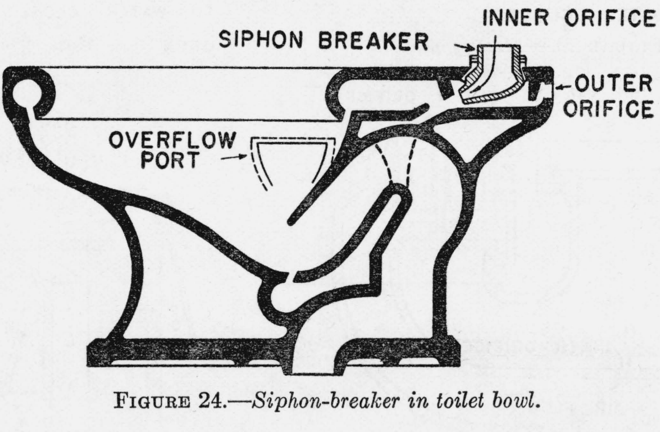

FIGURE 24.—*Siphon-breaker in toilet bowl.*

the air gap to the outer air; that is, by a function of the ratio d/D. Since the effective section of the outer passage cannot readily be determined by measurement, a test will usually be necessary to determine the limit of effectiveness $x/\Delta h$.

One manufacturer has installed a siphon-breaker of the nonmoving-part type in the flush chamber of the water-closet bowl (fig. 24). The principle involved is exactly the same as that already described, except that the closet bowl is also provided with an overflow below the rim ports which in effect gives two siphon-breakers in series, thereby increasing the over-all effectiveness. It may be pointed out in this connection, however, that, so long as the bowl is not clogged and the rim ports are open, there will be two siphon-breakers in series in any outfit in which a siphon-breaker is installed in the flushing mechanism of a jet-action water closet.

There are also adjustable nonmoving-part siphon-breakers being offered which are intended for application to any type of water-closet bowl. As none of these adjustable nonmoving-part devices has been examined in connection with this investigation, and since the authors have no record of their use in actual service, no conclusions as to their serviceability can be drawn here.

There is one other feature of nonmoving-part siphon-breakers to which attention may be called at this time. They all depend on an increase in the velocity of flow through an open gap in order to give the necessary velocity head for flushing the fixture and to reduce the pressure in the gap to atmospheric pressure in order to prevent the flow of water from the gap through the outer orifice. The noise produced by a flushing device is principally a function of the velocity of flow, and therefore a siphon-breaker of the nonmoving-part type is inherently noisy.

5. FLUSH VALVES AS SIPHON-BREAKERS

In the preceding paragraphs, moving-part and nonmoving-part siphon-breakers have been discussed from the standpoint of effectiveness inherent in the device itself. Obviously, if the siphon-breaker is used in connection with a valve that presents a smaller orifice than the inner orifice of the siphon-breaker proper, the effectiveness $x/\Delta h$ of the combination is materially greater than that of the siphon-breaker alone.

(a) STABLE FLUSH VALVES

Some flush valves are stable under a vacuum. By a "stable flush valve" is meant one which closes automatically under a line pressure sufficiently greater than atmospheric pressure and remains closed under any pressure that may occur in the supply line, either greater or less than atmospheric pressure.

In a stable valve, the passage open to backflow is the small by-pass, and such a valve in operating condition can be converted into an effective siphon-breaker simply by giving it an outer orifice the cross section of which bears the proper relationship to the minimum section of the by-pass f (see section IV–4–(b)). The limit of effectiveness may be determined by test in exactly the same manner as the limit for the separate siphon-breaker. The stable flush valve, when used as a siphon-breaker, is open to the same criticism as the moving-part detached siphon-breaker, namely, that the moving part may get out of order, thereby destroying its effectiveness.

The characteristics of construction on which stability in flush valves depends are discussed in section IV–7–(c).

(b) SIPHON-BREAKERS INTEGRAL WITH FLUSH VALVES

There are two types of construction in which the siphon-breaker may be said to be integral with a flush valve; one in which the siphon-breaker is constructed in the flush chamber of the valve body and functions independently of the valve mechanism proper, and the other in which the valve mechanism is utilized as a part of the siphon-breaker. The principle of the former does not differ from that of the detachable siphon-breaker except that the characteristics of the particular valve may be taken into account in determining its over-all effectiveness. The latter type is applicable only to the so-called stable valve.

It should be pointed out that, in installing any siphon-breaker, the minimum projected elevation x of the lower of the two orifices above the highest possible level of water in the fixture should not be less than the highest possible value of Δh as determined from the dimensions of the orifices or by test.

6. TESTS FOR EFFECTIVENESS

(a) APPARATUS REQUIRED

In most cases the effective diameters or areas of the orifices of siphon-breakers cannot be determined readily by measurement of dimensions, and in these cases a test will be necessary to determine the limit of effectiveness $x/\Delta h$. Figure 25 shows the essential parts of the equipment for making a test for effectiveness. The apparatus shown consists of: (a) A vacuum tank; (b) a vacuum pump; (c) a gage or manometer for measuring the pressure in the vacuum tank; (d) a water manometer or sensitive gage for measuring the pressure drop Δh in the flush pipe below the device being tested; (e) fittings for mounting the device and closing the flush pipe of the device; and (f) the siphon-breaker under test. The piping between the

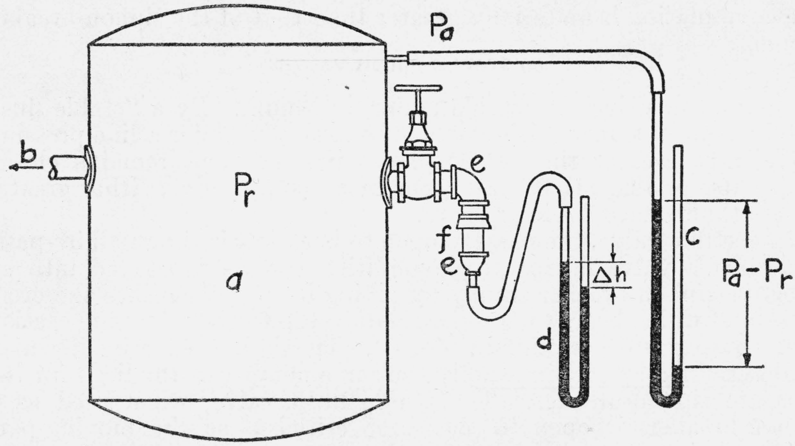

FIGURE 25.—*Sketch showing essential parts of apparatus for testing siphon-breakers.*

device and the vacuum tank should be short, and not smaller in diameter than the nominal size of the connection for which the device is designed. Figure 26 is a photograph of the equipment as used for these tests.

(b) ESSENTIALS FOR AN ADEQUATE TEST

A device can be tested adequately simply by evacuating the tank to a degree sufficient to maintain critical flow through the inner orifice for a long enough time to obtain a reading of the maximum drop in pressure Δh indicated on the gage *d*. For an atmospheric pressure of 30 inches of mercury, this means a pressure $P_r < 0.527\ P_a =$ 15.8 inches of mercury in the tank or a mercury manometer reading (gage *c*) of $(30 - P_r) > 14.2$ inches of mercury. All connections, especially manometer and gage connections, should be tight.

The test for effectiveness described is for the limit of effectiveness inherent in the device itself. The limit of effectiveness of a siphon-breaker in combination with a flush valve and water closet may be quite different. If the passage through the flush valve is smaller

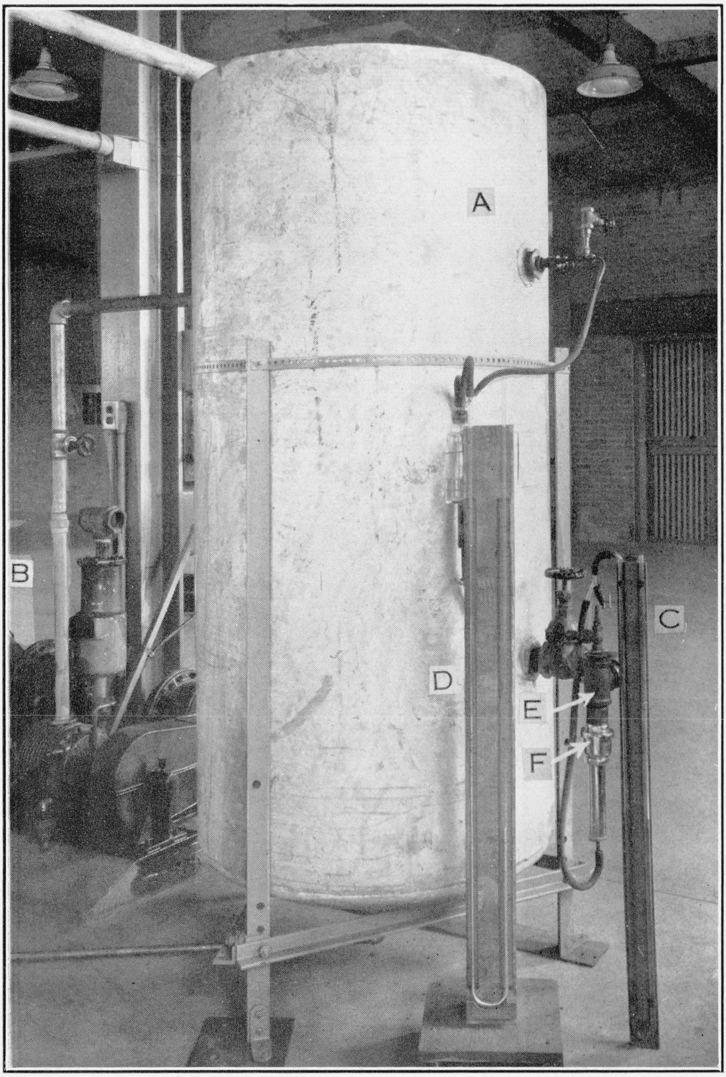

FIGURE 26.—*Apparatus used in air-flow tests.*

than the inner orifice of the siphon-breaker, the flush valve and not the inner orifice controls the rate of flow, and the effectiveness of the siphon-breaker in combination with this valve is increased. Hence the siphon-breaker may be set lower and still be fully effective. If the rim ports of the closet bowl are not submerged, they act as additional orifices, and the over-all effectiveness of the combination is increased. It is impossible to determine the magnitude of the effects of flush valves and closet bowls on the limit of effectiveness except in connection with specific valves and specific bowls, and then only by a test for over-all effectiveness. It follows that the inherent effectiveness of a siphon-breaker is the only basis on which limits can be set for universal application with all flush valves and all water-closet bowls.

(c) CAPACITY OF TEST EQUIPMENT

There has been considerable discussion among those interested in the prevention of backsiphonage as to what equipment is required for making a conclusive test of the effectiveness of a siphon-breaker. The required capacity of the tank depends on the size of opening through the device to be tested, the minimum pressure to which the tank can be evacuated, the types of instruments employed for making the necessary observations, on the adeptness of the testing personnel in manipulating the apparatus, and on the capacity of the pump, in case the test is made with the pump running. It will be found much more economical to install a tank adequate to meet the capacity requirements independently of the capacity of the vacuum pump, as the cost of high-vacuum pumps increases rapidly with increase in capacity. The essential requirement for a test of a siphon-breaker to determine its limit of effectiveness for use in any water-supply system; that is, for use in any or all water-supply sustems, and with any and all types of flush valves, is that critical flow through the device shall be maintained long enough to make the necessary readings or observations for determining the limit of effectiveness.

Assuming that a vacuum pump or other means is available for evacuating the tank to a known degree, for example, to a pressure of 2 inches of mercury (28-in. mercury vacuum), the time given for making observations may be estimated. The time required to change the pressure in the tank from $P_r=0.067$ atmosphere (=2 in. of mercury or 2.25 ft of water), to $P_r=0.527$ atmosphere (=15.8 in. of mercury or 17.9 ft of water), will be roughly (depending on the character of the opening) from 2 to 2.5 seconds per 100 gallons of tank capacity for an opening (orifice) in the device 1 inch in diameter; from 8 to 10 seconds through an opening ½ inch in diameter; and from 32 to 40 seconds through an opening ¼ inch in diameter. Assuming the same initial degree of evacuation, the time available for taking the necessary observations will vary directly as the volume of tank and (roughly) inversely as the square of the diameter of the orifice.

In the test of a nonmoving-part siphon-breaker with a ¾-inch diameter inner passage, the pressure P_c in the 318-gallon tank (fig. 26) increased from 0.26 atmosphere to 0.527 atmosphere in 7.5 seconds, and a simple computation shows that, if the initial tank pressure had been 0.067 atmosphere, it would then have taken about 13 seconds for the

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tank pressure to increase to 0.527 atmosphere. This is equivalent to about 4 seconds per 100 gallons capacity of the tank, a value which is in fair agreement with the estimate made above.

7. STABLE AND UNSTABLE FLUSH VALVES

(a) DEFINITION OF STABLE AND UNSTABLE VALVES

There are certain characteristics in the performance of automatic flush valves, under a reversal of pressure differential, that have a direct bearing on the practical application of the principles of siphon-breakers to these valves. A *stable flush valve* has been defined as one which closes automatically under a pressure in the supply line sufficiently greater than the atmospheric pressure and remains closed under any pressure in the supply line, either greater or less than atmospheric pressure, this last condition constituting a reversal of the pressure differential. An *unstable flush valve* will be defined as one which closes automatically under a pressure in the supply line sufficiently greater than the atmospheric pressure, but which opens under a pressure in the supply line appreciably less than atmospheric pressure. In any case, for either opening or closing, the difference between the line pressure and atmospheric pressure must be sufficient to supply the force necessary to overcome the combined frictional and gravitational forces opposing the movement of the valve.

(b) OPERATION OF FLUSH VALVE UNDER NORMAL PRESSURE CONDITIONS

It will be necessary to discuss the general principles of operation of automatic flush valves in order to explain the principles of stability. However, although there are many types and forms of flush valves in use in plumbing installations, it will be sufficient for the purposes of this paper to describe and discuss two types in detail—the piston and the diaphragm types.

A piston valve is shown diagrammatically in figure 27 in the closed and in the open positions, and a diaphragm valve is shown similarly in figure 28. These valves consist essentially of the valve body *a*; a piston or diaphragm *b*, which constitutes the moving part of the main valve; a pressure chamber *c*; a flush chamber *d*; an auxiliary release valve *e*; a by-pass *f*, connecting the pressure chamber to the line presure; and a handle *g* for setting the valve in operation. When the valve is closed, the service-line pressure P_1 acts on the annular area A_1 on the under side of the piston or diaphragm, giving a force P_1A_1 upward. The pressure P_2 in the pressure chamber *c* acts on the area A_2 on the top of the piston or diaphragm, giving a force P_2A_2 downward. The pressure P_3 in the flush chamber *d* acts on the central area of the piston or diaphragm A_3, giving a force of P_3A_3 upward. The resultant force F on the piston or diaphragm is given then by the relation

$$F=P_2A_2-P_1A_1-P_3A_3+W, \tag{34}$$

in which W is the weight of the piston or diaphragm assembly and forces acting downward are taken as positive. Neglecting friction, the valve will then seat and remain seated, provided F is positive. When the valve is closed under a service pressure P_1 sufficient to operate the valve, $P_2=P_1>P_3$, and eq 34 becomes

$$F=(P_1-P_3)A_3+W, \tag{35}$$

since $A_2=A_1+A_3$ exactly for the piston-type valve and approximately for the diaphragm-type valve. Since P_1-P_3 under any operating service pressure is always positive, F will always be positive under these conditions.

When the auxiliary valve is tripped to open the main valve, the pressure P_2 becomes sensibly equal to P_3 temporarily, and eq 34 becomes

$$F=(P_3-P_1)A_1+W, \tag{36}$$

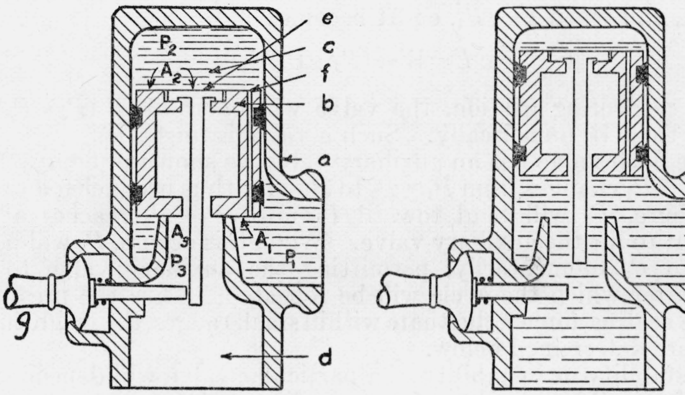

FIGURE 27.—*Piston-type flush valve.*

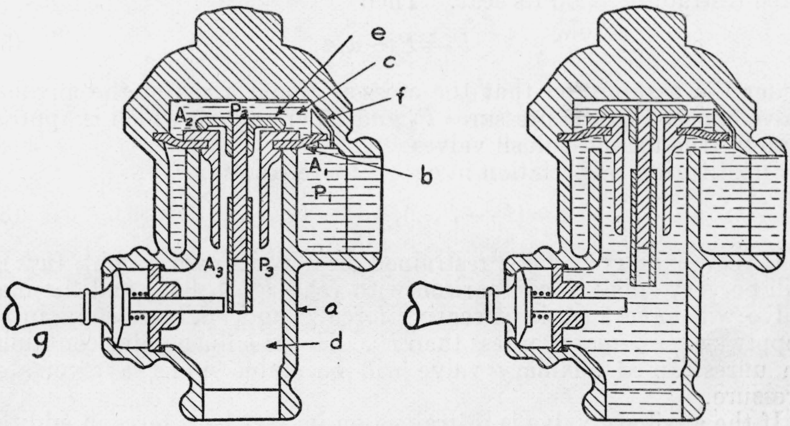

FIGURE 28.—*Diaphragm-type flush valve.*

and since P_3-P_1 is negative, the valve will open, provided $(P_3-P_1)A_1$ is numerically greater than W, as it is in any actual valve.

Once the pressures P_2 and P_3 have equalized, the auxiliary valve seats once more, the pressure P_2 starts to equalize with the line pressure P_1, and the main valve seats as soon as the value of F, as given by eq 34, becomes positive. The timing of the valve depends on the size of the by-pass and the volume of the pressure chamber, either or both of which may be adjustable.

(c) CONDITIONS FOR STABILITY OF FLUSH VALVES

Whether a flush valve will remain closed or will open under a subatmospheric pressure in the water-supply system, i. e., when $P_1 < P_3 = P_a$, depends on whether the value of F in eq 34 remains positive or becomes negative under these pressure conditions. If the by-pass is the only opening into the pressure chamber, P_2 will gradually equalize with P_1, and when $P_2 = P_1$, eq 34 becomes

$$F = W - (P_3 - P_1)A_3. \tag{35}$$

Hence, neglecting friction, the valve will open when $(P_3 - P_1)A_3$ is greater than W numerically. Such a valve is unstable.

If the flush valve has an auxiliary valve, as shown in figures 27 and 28, and if P_1 changes from $P_1 > P_3$ to $P_1 < P_3$, thus producing a reversal of pressure, P_2 will tend toward P_1 until $P_3 - P_2$ reaches a value sufficient to lift the auxiliary valve. When this occurs, P_2 will become equal to P_3 momentarily, permitting the auxiliary valve to close again, after which the cycle will be repeated. Thus the pressure P_2 and the seating force F fluctuate within small ranges, the magnitudes of which are determined below.

The stability or instability of a particular valve will depend on the value of $P_3 - P_2$ required to lift the auxiliary valve. The approximate quantitative relations are as follows:

Let a = the cross-sectional area of the auxiliary valve and w = the force restraining it on its seat. Then

$$P_2 = P_3 - w/a, \tag{37}$$

under the assumption that the cross-sectional areas of the auxiliary valve exposed to the pressures P_2 and P_3 are equal, which is approximately true for most flush valves.

Substituting this relation in eq 34 there results

$$F = (P_3 - P_1)A_1 + W - (w/a)A_2. \tag{38}$$

If the auxiliary valve is restrained only by its own weight, $(w/a)A_2$ will be very small in comparison with $(P_3 - P_1)A_1 + W$, and the main valve will have a positive seating force F under all pressures in the supply system greater or less than P_3. Hence a flush valve containing an unrestrained auxiliary valve will be stable under a reversal of pressure.

If the auxiliary valve is restrained on its seat by a force in addition to its weight, for example by a spring, the valve will be stable or unstable under a reversal of pressure according to whether $(w/a)A_2$ is less or greater than $(P_3 - P_1)A_1 + W$. If the total restraining force w is small compared to W, the valve will be stable. If w is large compared to W, it will be unstable.

Figure 29 illustrates the relation of the seating force F to the pressure P_1 for a stable piston-type flush valve for reversal in pressure from $P_1 > P_3$ to $P_1 < P_3$.

Referring to figure 29, if we assume that the valve is closed and has been closed long enough for P_2 to equalize with P_1, then eq 35 represents the relation between the seating force F and the line pressure P_1 for all values of P_1 greater than P_3-w/a, which is the pressure at which the auxiliary valve will unseat. (See eq 37.) The value of the seating force corresponding to this value of P_1 is obtained by substituting in eq 34 the conditions

$$P_1=P_2=P_3-w/a, \text{ and } A_2=A_1+A_3,$$

and proves to be

$$F=W-\frac{w}{a}A_3.$$

When the auxiliary valve unseats and the pressure P_2 becomes equal to P_3, the seating force suddenly increases to

$$F=W+\frac{w}{a}A_1,$$

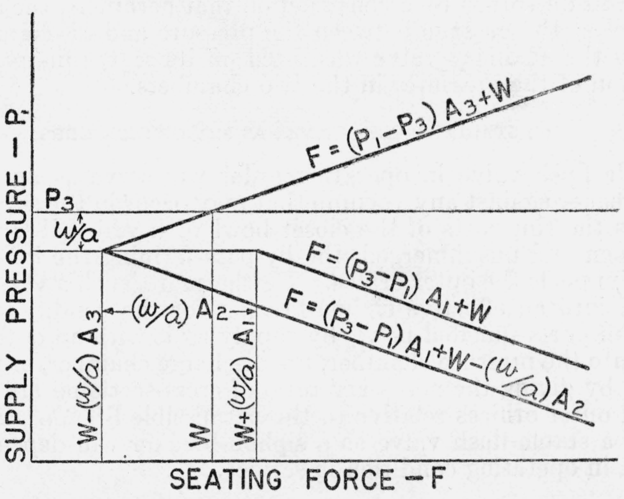

Figure 29.—*Relation between seating force and supply pressure for a stable flush valve.*

obtained by substituting in eq 34 the conditions

$$P_1=P_3-w/a, \; P_2=P_3, \text{ and } A_2=A_1+A_3.$$

Thus the range of fluctuation in the seating force is $(w/a)A_2$.

As the line pressure P_1 decreases still further, the pressure P_2 in the pressure chamber continues to fluctuate between P_3-w/a and P_3 because of the repeated unseating and seating of the auxiliary valve, and the seating force continues to increase, fluctuating over a range $(w/a)A_2$ continuously. Thus, for values of P_1 less than P_3-w/a, the seating force is represented by a band lying between two parallel lines, in which the seating force is always positive. The equations of these two bounding lines can be obtained by substituting the proper values of P_2 in eq 34.

It may be concluded from the preceding analysis that the stability of flush valves when subjected to subatmospheric pressures in the supply system depends on the pressure in the pressure chamber being held to slight variations below atmospheric pressure by an unrestrained or slightly restrained auxiliary valve or by a vent from the pressure chamber to the outer air, the latter combined with a check valve, if used.

Numerical values for the pressures and the areas used in the preceding discussion can be assigned only for particular valves. However, it is obvious that many flush valves would still remain stable if some slight restraint is applied to the auxiliary valve.

A number of different flush valves were examined to investigate their stability under a reversal of pressure gradient by measuring the areas A_1, A_2, and A_3 and by testing the valves under various vacua. With one exception, all the valves of the fully automatic type, in which the motion of the auxiliary valve was unrestrained by a spring, were stable, as the term is defined in this paper. In this one exception, the stability was destroyed by a construction that permitted the auxiliary valve to close the passage between the pressure and discharge chambers when the auxiliary valve was lifted off its seat, thus preventing equalization of the pressures in the two chambers.

(d) STABLE FLUSH VALVES AS SIPHON-BREAKERS

A stable flush valve in operating order will serve as an effective siphon-breaker against any vacuum that may occur in the supply lines as long as the rim ports of the closet bowl with which it is installed remain open and unsubmerged, the by-pass forming the inner orifice and the rim ports the outer orifice. Furthermore, such a valve can be converted into an effective siphon-breaker for the condition of submerged rim ports (flooded bowl) by supplying it with an outer orifice opening into the pressure chamber, the discharge chamber, or the flush pipe, and by giving the necessary ratio of cross-sectional areas of the inner and outer orifices relative to the permissible lift Δh. The reliability of a stable flush valve as a siphon-breaker will depend on its remaining in operating condition in service.

V. PREVENTION OF BACKFLOW IN WATER-SUPPLY SYSTEMS

Two possible methods of preventing backflow through potential cross-connections in water-supply systems are suggested by the preceding analysis of conditions (section III) and by the effects of vacua in plumbing systems (section IV): (1) The prevention of a pressure drop in the water-supply system sufficient to produce a pressure differential in any supply branch in the direction of the supply system, and (2) the prevention of siphon action in any supply branch, assuming that any possible vacuum may exist in the water-supply system.

Protection by the first method would have to be accomplished by certain forms of construction in respect to sizes and arrangement of supply pipes and by venting to control the pressure in the system to a definite minimum limit, which limit in each case will depend on the shortest down-feed supply branch of the system. Obviously, the capacity of the vents required to hold the pressure in an evacuating water-supply system above a definite minimum limit will depend on the volume rate of evacuation and not directly on the total or final volume evacuated. Since no definite limit can be set for the rate of evacuation that will occur in particular water-supply systems, in order to be on the safe side, it will be necessary to assume that the rate in a particular system may be the maximum that can occur in any vertical system, thus depending on the terminal velocity of flow in a vertical pipe of the diameter of the main supply pipe in the particular system (see section IV–1).

The second of these methods will be applied through the form of construction in the supply branch to the fixture (safe air gap) and by the use of special devices (siphon-breakers) in supply branches. Since there are no definite limits that can be set for the evacuated volume and minimum pressure that will occur in particular unvented water-supply systems short of the total internal volume of the system and the limiting vacuum pressure h_e, and since the maximum effects of a vacuum in producing backflow occur when the flow of air through the supply branch is a maximum, it will be necessary to proceed on the assumption that the evacuated volume and the pressure in any system may be sufficient to produce these maximum effects. The two methods of control are discussed in the following paragraphs with reference to specific application.

1. PREVENTION OF BACKFLOW BY FORMS OF CONSTRUCTION AND GENERAL VENTING

(a) CAPACITY OF VENTS

Assuming that down-feed branches are employed exclusively in a system and that the vertical projection of the minimum down-feed supply branch is x inches, in order to insure against backflow under any service condition that might possibly occur, the vents must have a total capacity sufficient to give a flow of air through the vents and piping equal to Q with a pressure loss of not more than x inches of water, where Q is the terminal volume rate of evacuation (see table 1).

(b) PERFORMANCE REQUIREMENT OF VENT VALVES

To be wholly satisfactory as a means of controlling the pressure in up-feed systems within a definite minimum limit, a vent or relief valve should have the following performance characteristics: (1) It should open fully and positively under a small drop in pressure (not more than x inches of water) below atmospheric pressure in order to prevent an initially low pressure from developing in the system; (2) it should close positively and remain closed at or slightly below atmospheric pressure in order to prevent outflow of water through the vent valve; and (3) these operations should be dependable after long periods of inactivity under service pressure. Until a valve having these performance characteristics has been developed and its dependability under all conditions has been determined, prevention of backflow in up-feed water-supply systems will not be in general a feasible method. However, the method of control by general venting can be applied to simple down-feed water-supply systems.

The essentials of construction for complete protection against backflow into a simple down-feed water-supply system (see fig. 2) are: (1) An overhead supply tank open to atmospheric pressure; (2) a continuous down-feed system through the main supply pipe (riser) from the tank and all branches in which there is any possibility of cross-connection with fixtures; (3) a separate supply branch from the riser to each fixture or an arrangement within any multiple branch, such that a siphon cannot be created within the branch; (4) a proportioning of the riser and branches in such a manner that the friction loss in head under maximum service demands will not reduce the pressure at the connection of any branch to the riser to less than (h_a-x) inches of water, where h_a is the atmospheric pressure in inches of water, and x is the vertical projection of the branch in inches; and (5) the use of open-top fixtures where any cross-connection with the water-supply system is possible. If the main supply riser from the tank has a shut-off valve as shown in figure 2, then in addition the system must have a vent pipe connected below the shut-off valve and extending above the water level in the supply tank as indicated. This method is feasible for a simple down-feed system, and can be made effective for any down-feed system. However, in a complicated down-feed system with a supply header and several risers, the method, although possible, would probably prove impractical on account of the extensive venting costs necessary to meet the requirements. It would be difficult also for building officials to determine whether the construction in particular buildings meets these requirements or not. It would seem from these considerations that, for the present at least, a general and practical control of conditions will be restricted to the second method of attack.

2. PREVENTION OF BACKFLOW IN SUPPLY BRANCHES

Assuming no provision for controlling the minimum pressure that may occur in water-supply systems, there are three ways in which backflow into the system can be restricted or prevented: (1) By safe air gaps; (2) by siphon-breakers; and (3) by check valves.

(a) SAFE AIR GAP

In open-top plumbing fixtures that do not require a direct connection from the water-supply system, the obvious and economical means of preventing backflow is to provide a safe air gap,

$$x=2.45\,d\left(1-0.26\,\frac{d}{D}\right)\left(1-0.28\,\frac{d}{h_a}\right)\text{ (see eq 32),}$$

between the supply opening (faucet spout) and the highest possible level to which water can rise in the fixture or receptacle. Also a safe air gap slightly larger than the minimum may be specified as a concrete number applicable to a limited range of sizes of faucets or supply outlets, for example, $x=a$ inches (see eq 32a and discussion). This method may be used effectively in any cross-connection where a direct connection is not essential for the proper functioning of the fixture.

(b) SIPHON-BREAKERS

Where a clear air gap between the fixture and the water-supply branch is impractical, for example, in water-closet bowls with flush-valve supply, a siphon-breaker can be installed to prevent backflow. The effectiveness of such a device is measured by $x/\Delta h$, where x is the vertical projection from the lowest orifice in the siphon-breaker to the highest possible water level in the fixture, and Δh is its effective limit. A value of $x/\Delta h=1$ signifies full effectiveness inherent in the device itself, and the margin of safety increases with the value of this ratio.

There has been considerable rivalry in the past between the advocates of the two general types of siphon-breakers (moving-part and nonmoving-part) and some controversy regarding their relative merits. Questions of this nature can be answered best by stating the general characteristics of the two types for consideration in regard to their effectiveness in preventing backflow, dependability, and general serviceability. For each pattern or design of either type there is a definite minimum limit to the distance x which the device may be set above the top of a water-closet bowl and give 100 percent effectiveness in preventing backflow.

The nonmoving-part siphon-breaker must be suited in design to the flushing characteristics of the closet bowl with which it is installed, and it is impossible to design a single model, without an adjustment feature, suitable for use with water-closet bowls distinctly different in their flushing characteristics or to design a single model, whether adjustable or not, that will be completely effective in preventing backflow when x is small. There is no assurance that a siphon-breaker suitable for the closet bowl at the time of installation will be suitable later if the flushing characteristics of the closet bowl change after a period of service. Also the nonmoving-part siphon-breaker is inherently noisy in operation.

The design of a moving-part siphon-breaker is not dependent on the characteristics of the water-closet bowl, so it can be designed to make it completely effective for very low values of x and quiet in operation.

Objections to the moving-part siphon-breaker have been made on the grounds that the moving part may get out of order in service and cease to function and that it does not permit a rapid relief of vacua in water-supply systems. Whether the former objection is a valid one or not can probably be most convincingly answered by the history of these devices in actual service. In reference to the latter objection, it may be pointed out that there is no logical reason for relieving the vacuum in a water-supply system if all supply outlets from the system are protected against backflow. Also it will be a distinct service disadvantage if the water-supply system is permitted to fill with air every time a subatmospheric pressure occurs, since the air has to be let out before the system can refill.

(c) PREVENTION OF BACKFLOW BY CHECK VALVES

A check valve alone can not be regarded as positive protection against backflow through a cross-connection, because of possible failure to operate. However, in any existing direct cross-connection, it may be less hazardous to permit the use of a check valve or combination of check valves than to remove the cross-connection, if its removal would introduce a different hazard or increase one already existing, as, for example, the fire hazard. Control of such conditions is more a matter of judgment, based on a knowledge of the local conditions, such as pressure variations, nature of the cross-connection, character of the waters in the two systems, etc., than of facts of general analysis. The only certain way to prevent flow through a direct cross-connection is to remove the cross-connection or convert it into one that can be controlled positively.

(d) PROTECTION OF CLOSED FIXTURES

Since there is no means of determining the maximum back-pressure which a closed fixture may exert on a direct supply connection, there is no apparent means of positively preventing backflow through a cross-connection of this type. If the direct supply connection is essential, the only certain way of preventing pollution of the main water supply of the building is by means of a separate or auxiliary water-distributing system to fixtures of this type. If the direct-supply connection is not essential for satisfactory service, then the obvious way of preventing backflow is to convert the cross-connection to one in which a safe air gap may be employed.

(e) AUXILIARY SUPPLY SYSTEMS

An auxiliary water-distributing system for water closets and urinals has been advocated and used to some extent as a means of preventing contamination of the main water supply. If such an auxiliary system has no connection or branch through which water might be drawn for drinking or domestic purposes, and if the main system is fully protected against backflow from the auxiliary system by a safe air gap between the two, the method is perfectly safe, so long as the system remains unaltered by structural changes. Objections to this method of protecting the main water supply are principally on the basis of cost, and on the assumption that the system may be altered without the knowledge or consent of the health authorities. However, in some cases, the cost of installing an auxiliary supply system may not exceed the cost of installing siphon-breakers for individual fixtures, and it is impossible to estimate or weigh the chance of unauthorized alterations.

3. COMBINATION OF METHODS OF PREVENTING BACKFLOW

There is, of course, the possibility of combining the method of general venting with the use of siphon-breakers on the supply branches for a limited number of fixtures. It has been suggested that siphon-breakers or vacuum-breakers may be constructed to serve the functions of venting, thereby preventing low-pressure vacua from occurring in the water-supply system. There are, however, difficulties in applying this method effectively that are practically insurmountable at the present stage of development. First, the siphon-breakers of large venting capacity so far developed are suitable for use only on the fixture side of the supply control valve, and therefore the rate of inflow of air into the supply system (venting) will depend principally on the valve and not on the siphon-breaker or vacuum-breaker. In the case of compression faucets and stable flush valves, the venting through the valve will in general be negligible in its effects, since these valves ordinarily are closed. In the case of unstable flush valves, the maximum inflow of air into the supply system, though much greater than through the by-pass of a stable valve under the same pressure conditions, will still be a function of the valve rather than of the siphon-breaker. Furthermore, if a siphon-breaker wholly satisfactory for use on the supply side of the flush valve has been or can be developed, it might just as readily be installed on the top of risers and principal supply branches, where it would be most effective in controlling the pressure in the system. Finally, if all supply outlets are fully protected by air gaps or by siphon-breakers, there is no need for controlling the pressure in the water-supply system.

VI. PRACTICAL CONSIDERATIONS IN THE APPLICATION OF RESULTS

The ultimate purpose of this investigation is to supply information on which to base effective health regulations pertaining to the installation of plumbing. Presumably these regulations, when they become law, will be mandatory and will affect many different groups—the owners or managers of properties, the public officials charged with the enforcement of the regulations, architects, sanitary engineers, manufacturers of plumbing materials, contractors (master plumbers), workmen (journeymen plumbers), and the public who occupy or use the buildings and who use the public water supply.

There have been apparently two widely different or extreme attitudes toward the hazard created by cross-connections in plumbing systems: (1) The attitude or viewpoint that a distinct hazard exists wherever there is any possibility of backflow into the water-supply system, however remote; and (2) the attitude that the hazard offered by an indirect (potential) cross-connection in a plumbing system is negligible because the combination of conditions necessary to complete the cross-connection is not likely to occur in most plumbing installations. Between these two extreme views, any number of opinions as to the seriousness of the hazard and as to specific necessary corrective measures, as well as noncommittal attitudes, may be encountered among those who have given the problem some thought. Therefore, it is to be expected that these differences in opinion and attitudes cannot be wholly reconciled without further and possibly prolonged discussion of the subject.

The analysis of the problem in the preceding sections of this paper, as has been stated frequently, has been developed from the standpoint of complete protection against backflow under the most extreme conditions, not because this is necessarily the correct basis from the standpoint of the most satisfactory or most effective health regulation, but because it is the only basis on which definite limits and definite performance specifications for corrective measures can be established. In fact, in spite of the extreme attitude that may appear to be supported by this discussion as a whole, the authors are of the opinion that a conservative attitude on the part of those who may be called upon to formulate either recommended or mandatory regulations for the prevention of backflow in plumbing systems will result in a more effective control than an intolerant attitude which demands immediate perfection.

Because of the varied interests affected, it would appear that if possible, the interested parties should be satisfied that any regulations which may be adopted are reasonable in their requirements and are necessary for the protection of the health of the public. It would also appear that these varied groups should have, through competent representatives, a voice in the formulation of the proposed regulations, in order that the viewpoint of each particular group may be presented and considered. For these reasons, this paper deals principally with the technical phases of the problem, and no attempt has been made to draw final conclusions as to the form or scope of the regulations that may be based on the information given. From the standpoint of the practical application of the results, the investigation is incomplete, and the major part of the investigation, in some respects the most difficult part, is yet to be accomplished.

The committee or organization that may be called upon to formulate a satisfactory regulation, and the legislative body to which this proposed regulation may be presented for enactment into law, will be confronted not only with the technical problems dealt with in this paper but also with the social and economic problems of obtaining compliance with the law after its enactment. The effectiveness of any health regulation that may be adopted will be determined by the degree of obedience to it rather than by its literal requirements. It will therefore be advisable in formulating the law to consider very carefully the probable results of the law from the standpoint of enforcement or obedience. In this connection the following questions may well be raised and carefully considered:

1. Will a mandatory health regulation aimed at a complete prevention of backflow into the water-supply system in any building under any possible condition that may occur prove as effective in its results as a regulation less exacting in its scope of application and in the completeness of the protection given?

2. Should the mandatory regulation apply only to new construction, or should it be retroactive and apply to all plumbing, with the attendant difficulties of surveys, condemnation, and enforcement of replacements in existing structures?

3. Should the regulation apply equally to all types of buildings, public and private, or should some limit of mandatory application be set, as, for example, to hospitals, hotels, restaurants, and other buildings of a similarly public character?

4. Should any differentiation be made in the mandatory application of the regulation in reference to topography, zoning, etc.? Or should a differentiation be made in respect to the type of cross-connection?

These are some of the questions that may well be raised in advance of the formulation and enactment of any mandatory regulation. Any regulation that appears unreasonable in its requirements will invite evasion and violation of the law, thus to a large extent defeating its real purpose.

It may be worth while to illustrate some of the difficulties referred to in the preceding paragraph. For example, the minimum air gap that will give complete protection against backflow from an open-top faucet-supplied plumbing fixture under any pressure condition that can occur is roughly 1.8 d, where d is the mean internal diameter of the faucet nozzle. This is based on the simultaneous occurrence of a sustained vacuum pressure of less than 0.527 atmosphere in the supply line, a fixture filled to the rim, and an open faucet. Obviously, this combination of conditions will occur only infrequently in any system and will never occur in some systems. Obviously, too, any clear air gap, however small, will give protection in any system the greater part of the time, regardless of the maximum vacuum and maximum flooding of fixtures that may occur at irregular and infrequent intervals. Again, there is no possible way of determining whether the conditions that are necessary to produce backflow from a particular fixture having an air gap less than 1.8 *d* will or will not occur in an up-feed water-supply system of a particular building, and therefore, there is no possible way of determining what air gap less than 1.8 *d* would suffice to give complete protection to a particular fixture on a particular system.

The situation in regard to other types of cross-connections is similar; and, because of this situation, attempts to classify cross-connections as to the relative danger of backflow become a matter of judgment or opinion and not one of scientific fact. It may become necessary, however, to make a classification of cross-connections on the basis of the menace to health offered by them in order to reduce or to restrict the field of mandatory application to one that can be adequately handled and controlled. Fortunately, when corrective measures are necessary for protection, either by special forms of construction or by the use of special devices, it will usually be as easy and as economical in new construction to obtain complete protection against backflow under any possible condition as to obtain a less complete or partial protection.

The authors acknowledge gratefully the valuable assistance rendered by manufacturers of plumbing supplies in connection with this investigation, particularly by submitting for consideration the problems that they face in the prevention of backsiphonage and by furnishing materials and supplies needed in the experimental studies. The suggestions offered by various members of engineering and trade organizations have been very helpful and are acknowledged with thanks. Special acknowledgement also is due Mr. R. H. Zinkil of the Crane Company laboratories for permission to use his data on safe air gaps and to Dr. G. H. Keulegan of the National Hydraulic Laboratory staff for his contributions to and assistance in the mathematical analysis.

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WASHINGTON, January 4, 1938.