DEPARTMENT OF COMMERCE

TECHNOLOGIC PAPERS

OF THE

BUREAU OF STANDARDS

S. W. STRATTON, DIRECTOR

No. 193

DESIGN OF ATMOSPHERIC GAS BURNERS

BY

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Bureau of Standards

SEPTEMBER 6 1921

PRICE, 15 CENTS

Sold only by the Superintendent of Documents, Government Printing Office
Washington, D. C.

WASHINGTON
GOVERNMENT PRINTING OFFICE
1921

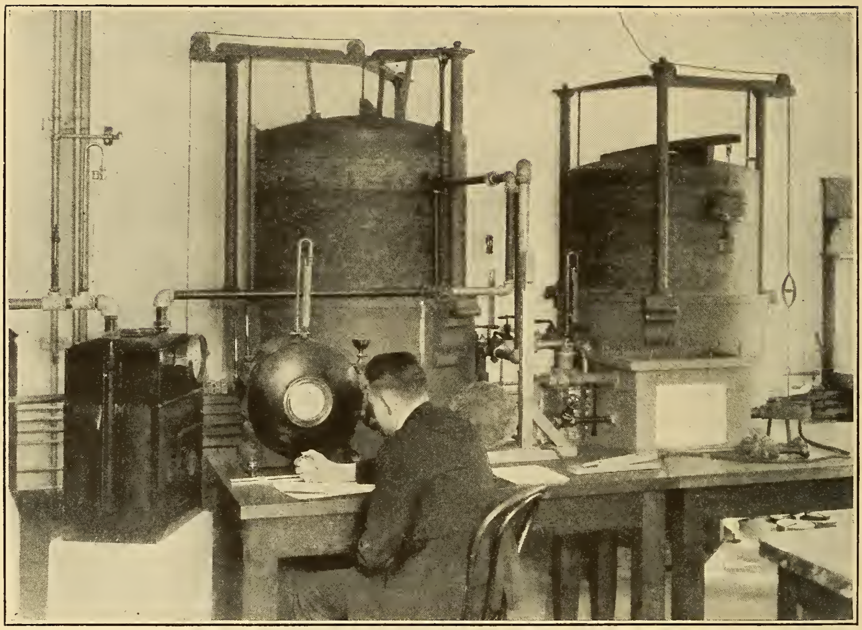

FIG. 1.—*Appliance laboratory, Bureau of Standards, showing apparatus used to test burners*

DESIGN OF ATMOSPHERIC GAS BURNERS

By Walter M. Berry, I. V. Brumbaugh, G. F. Moulton, and G. B. Shawn

ABSTRACT

The first part of an extensive investigation of the design of gas burners is presented in this paper. With the arrangement of apparatus and method of testing that has been developed one can measure quickly and accurately the volume of air injected into any burner under any condition of operation, as well as determine the limits of operation with any quality of gas. Such information is essential in order to enable one to design burners for any predetermined condition of operation.

In order to understand the various factors entering into the design of burners it was found necessary to study the theory of flow of gas through different types of orifices, the principles governing the rate of injection of air into the burner, the design of the injecting tube, the rate of consumption of burners of different port areas, and the effect of adjustment of the air shutter.

The rate of discharge of gas orifices of different types was found to vary greatly with a variation of the angle of approach and with the length of channel or tube of the orifice.

The advantages of having an injector of good design to secure the injection of a larger volume of primary air have been illustrated by tables and a large number of curves. The general design of an injecting tube that produced the greatest injection of primary air is shown.

It has been found that for any given burner the ratio between the momentum of the gas stream and the momentum of the stream of the air-gas mixture entering the burner is always a constant, and this relation enables one to calculate readily the effect on the volume of air entrained when the gas pressure, gas rate, or specific gravity of the gas is changed.

It is very important that there should be a correct relation between the area of the throat of the injecting tube and the area of the burner ports if the energy of the gas is to be efficiently utilized to inject air into a burner. The results of the tests show that the area of the injector throat should be about 43 per cent of the area of the burner ports.

In pipe burners the rate of consumption per square inch of port area increases until the port area is about equal to the cross-sectional area of the pipe. Tables have been tabulated to show the rate of consumption of burners with and without improved injecting tubes.

If the openings in the air shutter are too small there is an appreciable loss of air injected into the burner. The velocity through the air opening should not exceed 4 or 5 feet per second.

The results of tests to show the efficiency of operation of various types of burners with different qualities of gas will be the subject of future reports.

CONTENTS

I. INTRODUCTION

1. PURPOSE OF INVESTIGATION

The large increase in cost of gas-making materials and the difficulty in securing materials of satisfactory quality have resulted in frequent increases in the price of gas to the consumer. This has led to attempts to secure higher efficiencies with the existing appliances, and to devise new and improved methods of utilization.

In many localities the shortage of raw materials has compelled a decrease in standards of quality, and in other localities they have lowered the standards instead of increasing the price. The problem of determining the best standard for any locality would be made materially easier if definite information on the relative utilization efficiencies of different qualities of gas were available. This question can be solved only by a careful study of the performance of appliances and by determining the adjustments which will give the highest efficiency with each quality of gas.

The nation-wide movement for the conservation of natural gas has already brought out rather clearly the fact that a great saving of natural gas can be accomplished by improvement in the design and by readjustment of natural-gas appliances. Many of the natural-gas companies are also finding it necessary to supplement their supply with artificial gas, and, since the characteristics of artificial and natural gas are so different, some investigation of the appliances has been necessary to determine within what limits it is possible to vary the proportions of the two gases.

The question of increasing the efficiency of utilization of gas has been considered so important that a great deal of effort of the Industrial Fuel Sales Committee of the American Gas Association has been directed toward research work on this problem. As a part of this program the gas engineering section of the bureau of standards was requested to make a study of atmospheric burner design and operation. The authors desire to acknowledge the assistance and encouragement which has been received from the members of that committee. Credit is also due J. H. Eiseman for efficient services in this investigation.

A brief report of the first results of this work was read before the commercial section at the 1919 convention of the American Gas Association.

2. SCOPE OF TESTS AND OF THIS PAPER

This paper is based on several thousand observations in which the effects of many of the large number of variables affecting burner operation were studied under a wide range of operating conditions. After a description of the principles of operation of atmospheric burners, the results of investigations on the effect of design of orifice and injecting tube on rate of consumption of burners are given, as well as a discussion of the general principles governing air injection. The questions of burner design, involving the relations between port area and rate of consumption of burners, the limit of velocity of efflux from the ports, and other factors affecting the operation of burners are also considered. Some of the other phases of this problem, such as the completeness of combustion, flame characteristics with different gases, efficiency of operation of burners of various designs and using gases of different quality and with different proportions of primary air, which are not covered in this paper, will be discussed in later reports.

The subject of industrial atmospheric burner design has been covered in a general way in the lesson papers of the educational course edited by the American Gas Association, but these articles do not attempt to show numerically the effects of changes in the gas rate, gas pressure, or the specific gravity of the gas. The advantage of a correctly designed injecting tube as a means of improving the combustion and securing greater rate of consumption has evidently not been fully appreciated, although specially designed injecting tubes have been used to some extent to advantage.

No extensive presentation of theory has been attempted, the aim being rather to show by means of tables and curves representing experimental data the effect of the various factors on the operation of burners. The tables and curves, unless otherwise indicated, will be based on actual results of tests, and can be duplicated by anyone who may care to continue any part of the investigation.

Some of the following discussion may seem elementary, but our experience has shown that the problem is difficult because of the large number of variables involved and the difficulty of keeping in mind the proper relation between the several factors. For that reason an attempt has been made to show clearly by means of illustrations and solutions of simple problems the effect of each variable.

It is hoped that the information in this paper will be of practical assistance to the manufacturers of gas appliances, as well as to the industrial gas-appliance engineers who are frequently required to design and make their own burners for many special purposes.

II. DESCRIPTION OF APPARATUS AND METHODS OF TEST

A photograph of the apparatus used at this Bureau where these tests were made is shown in Fig. 1. The arrangement of the apparatus is shown in Fig. 2.

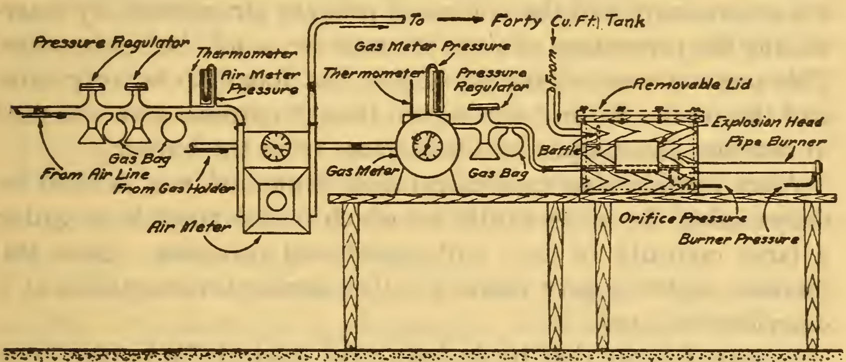

FIG. 2.—*Arrangement of apparatus used for burner tests*

1. METERS AND REGULATORS

In order to obtain higher pressures at the orifice than were available from the gas supply line, it was found necessary to pump the gas from the line into a weighted 40 cubic foot gas holder. The gas was measured with a carefully calibrated wet meter. The temperature and pressure of the gas were read at the time of each meter reading. At the outlet of the meter there were placed in series a dry regulator, and a special gas bag which served as an excellent antifluctuator. This combination gave such uniform pressure at the orifice that the slight kicks from the meter were hardly perceptible in the burner pressure, except at the higher gas rates.

The air supply was taken from a regular laboratory supply line. The pressure was reduced by two dry regulators in series. A gas bag connected at the outlet of each regulator eliminated the remaining fluctuation caused by the compressor. A 30-light dry meter was used for measuring the air. To eliminate the fluctuations in pressure caused by this meter the air was allowed to flow through a 40 cubic foot tank. The temperature and pressure were recorded at each observation.

At the beginning of these tests considerable difficulty was experienced from the fluctuation in pressure in both the gas and air lines, as well as the kicks from the meters. By the use of these regulators and gas bags the air and gas entered the burner at such uniform pressure that the greatest fluctuations of pressure in the burner were usually less than 0.001 inch of water pressure.

2. METHOD USED FOR MEASURING AIR-GAS RATIO

It was realized soon after the experiments were started that the determination of the volume of primary air injected, by determining the percentage of air in the mixture, would be burdensome. This method was attempted at first but found to be very slow, and the results did not agree even though extreme care was used in drawing the samples of the mixture from the burner.

After considerable experimentation a method was devised for determining the air-gas ratio by which it was possible to gather a large quantity of data with speed and precision. Since this method might be very useful in other similar investigations, it is described in detail.

It was thought that by inclosing the inlet of the burner in an air-tight box and metering the air into the box at a rate which would keep the pressure inside at exactly atmospheric pressure, the volume of air entering the burner could be determined directly from the meter reading. This method was described by Mansfield in London Journal of Gas Lighting, July 13, 1909. It was found, however, that minute variations in pressure in the box which could not be determined on the most sensitive gage that was available gave very wide variations in the ratio of the air to gas mixture entering the burner.

The plan tried next, which has been used very successfully, was to leave the top of the box open and operate the burner in a normal manner, observing carefully the pressure in the burner as indicated on a very sensitive slope U gage. The lid was then placed on the box and the air passed through a meter into the box at a rate which exactly duplicated the previous conditions of pressure within the burner. It is evident that if the gas rate in each case was the same, and the pressure in the burner was exactly duplicated, the volume of air injected into the burner in each case was the same. The ratio of the volume of air to the volume of gas that entered the burner was thus readily calculated from the readings of the two meters.

Xylene was used in the slope U-gage. The gage was calibrated against a water gage and was found to give excellent results.

When it was desired to note the flame characteristics, the heights of the blue inner cone and of the flame were also measured.

In some tests made with an open pipe burner, where the static pressure was low and the velocity pressure was high, a very small bent tube was inserted into the center of the pipe burner, with the open end of the tube against the stream. The velocity pressure in the burner was thus secured. Since, in practically all of this work, the absolute pressure values were not essential, it made little difference how or where the pressures were measured so long as satisfactory and comparable relative readings were obtained. It was gratifying to find that with our apparatus and method it proved possible to make the various measurements with considerable precision and reproducibility, as indicated by the tables and curves in this paper.

To eliminate any hazard in case there should be an explosion in the box caused by a combustible mixture, a piece of oiled paper was pasted over a large opening in the side of the box and made a very satisfactory explosion head. It was demonstrated on two occasions during the tests that this was a very wise precaution.

III. THE OPERATION OF AN ATMOSPHERIC BURNER

1. DESCRIPTION OF BURNER

In Fig. 3 is shown a simple type of atmospheric pipe burner used in these tests, and also a sketch of the ordinary domestic range burner, to which much of the following discussion is equally applicable. The different parts of the burners are clearly designated according to the nomenclature commonly used by gas men.

The gas, after passing through the controlling cock, issues with a high velocity through the orifice (about 160 feet per second, with 4-inch pressure and gas of 0.65 specific gravity) and the momentum of the gas stream causes air to be injected into the burner. The velocity pressure of the mixture is converted into static pressure sufficient to force the mixture through the burner ports.

The mixture of gas and air passes through the constriction or injecting throat into the mixing tube, where the gas and air are fairly well mixed before arriving at the first ports in the burner proper, or burner head, as it is called in burners of the domestic or ring burner type.

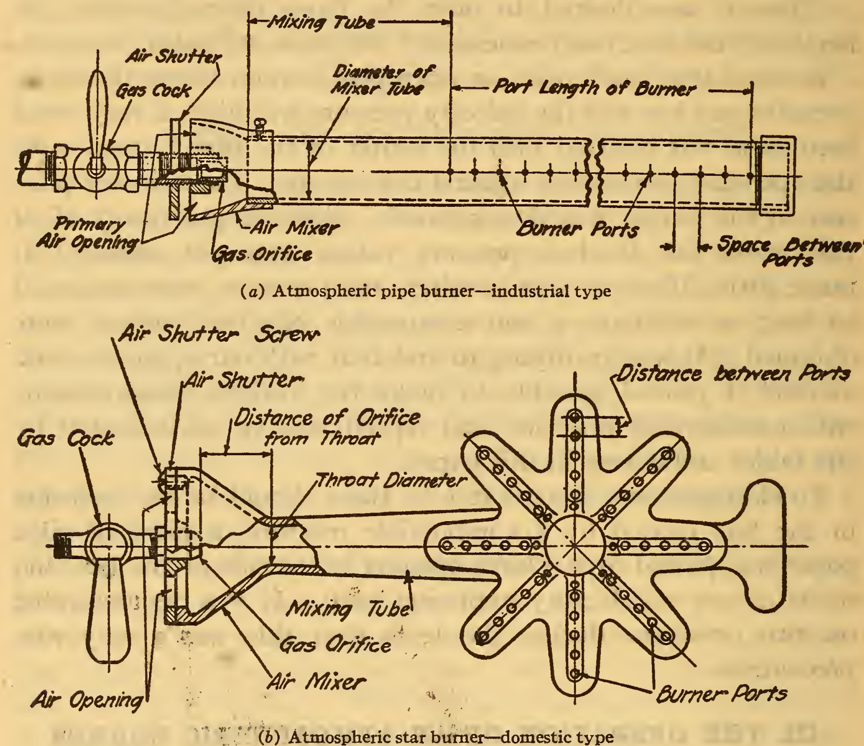

FIG. 3.—*Top view of atmospheric burners of industrial and domestic types, showing nomenclature commonly used*

2. FACTORS AFFECTING AIR ENTRAINMENT

The total volume of air entrained by a stream of gas depends upon the following factors:

1. Gas rate.
2. Gas pressure.
3. Specific gravity of gas.
4. Design of orifice—determining the friction loss.
5. Position of orifice.
6. Area of air opening in shutter.
7. Design of injecting tube.
8. Dimensions and shape of burner head.
9. Total area of flame ports.
10. Temperature of the burner head.

The effect of gas pressure, rate of flow, and specific gravity of the gas on air entrainment will be discussed under the heading "Principles governing air injection"; that of the other factors will appear in later sections under the appropriate headings.

IV. THE BURNER ORIFICE

1. LAWS OF FLOW OF GAS THROUGH ORIFICES

Many investigators, in their calculations of the velocity of flow of gas through an orifice, start with the assumption that the flow is adiabatic. It is undoubedly true for gases that the flow follows that condition, but for small pressures, such as are used in an ordinary city supply, the difference in absolute pressure is very small, and, therefore, the difference in specific volume due to compressibility is very small, and the hydraulic formula can be applied with an error less than 1 per cent.[1] In this case the "theoretical velocity" of flow in feet per second through an orifice is

$$V=\sqrt{2gh},$$

in which h = the "head" or height in feet of a column of gas (of the same density as the gas at the orifice) required to produce the pressure at the level of the orifice, and g = the acceleration of gravity—32.2 feet per second, per second. The actual velocity at

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[1] See B. S. Sci. Paper No. 359, Efflux of gases through small orifices, by E. Buckingham and J. D. Edwards for the development of the following adiabatic formula of velocity in feet per second.

$$V^2=2g\frac{P_1}{W}\cdot\frac{1-r^x}{x}, \text{ where}$$

P_1= initial absolute pressure of the gas in pounds per square foot;
W= initial density of the gas in pounds per cubic foot;
$x=\frac{k-1}{k}$, where $k=\frac{C_p}{C_v}$ the ratio of specific heats and is very approximately equal to 1.40;
$r=\frac{P}{P_1}$, where P= the back pressure, which in this case is merely the outside barometric pressure in pounds per square foot.
Let $P=P_1-p$, where p is the difference between the initial and barometric pressure in pounds per square foot. Then for small values of p such as would correspond to city supply pressures, we have very approximately

$$r^x=\left(\frac{P}{P_1}\right)^x=\left(\frac{P_1-p}{P_1}\right)^x=\left(1-\frac{p}{P_1}\right)^x=1-x\frac{p}{P_1},$$

in which case

$$V^2=2g\frac{P_1}{W}\cdot\frac{1-\left(1-x\frac{p}{P_1}\right)}{x}=2g\frac{P_1}{W}\cdot\frac{x\frac{p}{P_1}}{x}$$

$=2g\frac{p}{W}$, where $p=Wh$, $\therefore h=\frac{p}{W}$, and the equation

$V=\sqrt{2g\frac{p}{W}}$ is identical with the hydraulic equation

$V=\sqrt{2gh}.$

The error that will result from using $r^x=1-x\frac{p}{P_1}$ is less than 0.1 per cent for a pressure of 4 inches of water and is about 0.4 per cent for a pressure of 8.0 inches of water.

the orifice is always appreciably less than this theoretical velocity, and consequently the quantity of flow is less than the quantity calculated from the theoretical velocity and area of orifice. The ratio of the actual flow to the theoretical is known as the discharge coefficient. The quantity of flow per second is, therefore, equal to the product of the theoretical velocity, the area of the orifice, and the discharge coefficient, which latter takes into account the contraction of the vein or flowing stream, the friction of the orifice, etc.

With a stream of water, using a sharp-edged orifice, such as has been designated as No. 3, Fig. 6, in this paper, it is reported in Marks' Mechanical Engineers' Handbook that the minimum area of the cross section of the jet, which is at a distance out from the orifice about one-half its diameter, is about 0.62 of the area of the orifice. It is stated that the average velocity at this point is about 0.98 to 0.99 of $\sqrt{2gh}$. The discharge coefficient, which is the product of the contraction coefficient and the velocity coefficient, for the above condition is 0.61. With an orifice of this type and using the above formula, our tests have shown the discharge coefficient with a gas stream to be about 0.605, which is identical for the value given for liquids.

A simple equation by which to compute the theoretical velocity of gas from an orifice can be developed as follows:

Let H = orifice pressure of gas in inches of water;

d = specific gravity of gas (air = 1.0); then

$$h=\frac{H}{12}\,\frac{820}{d}=68.33\,\frac{H}{d}\text{ feet, where}$$

$$820=\frac{\text{density of water at }60^\circ\text{ F.}}{\text{density of air at 30 inches Hg and }60^\circ\text{ F}}$$

Therefore, the theoretical velocity is

$$V=\sqrt{2gh}=\sqrt{64.4\times 68.33\,\frac{H}{d}},$$

$$=66.34\sqrt{\frac{H}{d}}\text{ feet per second.}$$

(a) EFFECT OF PRESSURE ON RATE OF FLOW OF GAS

From this simple equation it is at once apparent that the theoretical velocity, therefore, the rate of flow of gas of a given specific gravity issuing from an orifice, varies as the square root of the pressure. Thus, if it is assumed that 1 cubic foot of gas per unit of time flows through an orifice at a pressure of 1 inch the volume of gas of the same specific gravity that would flow through the same orifice per unit of time at different pressures would be with 2 inches pressure 1.415 cubic feet, with 4 inches pressure 2.000 cubic feet and with 5 inches pressure 2.235 cubic feet.

This simple relation, however, applies only to the sharp-edge orifice and does not apply to orifices of the channel type, the reasons for which will appear later. It is also understood that for the above conditions the barometric pressure and gas temperature were the same.

Fig. 4 shows the agreement between the observed values for five different pressures and values obtained by using the relation

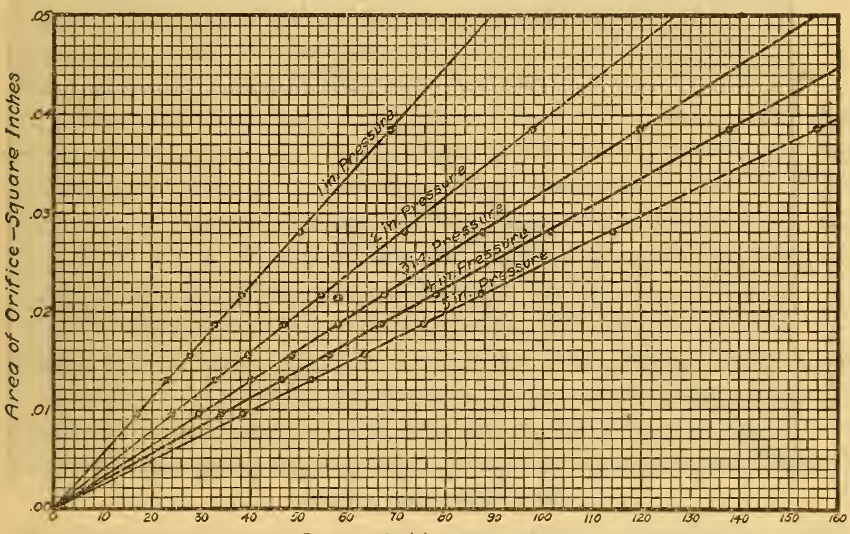


FIG. 4.—*Rates of flow of a 0.65 specific gravity gas through various sizes of orifices of type No. 1 (Fig. 6), with pressures of 1, 2, 3, 4, and 5 inches of water, respectively*

given above. For example, the orifice that passed 69 cubic feet at 1 inch pressure passed 138 cubic feet at 4 inches pressure. The No. 1 type of orifice (see Fig. 6) with gas of 0.65 specific gravity was used in this test.

(*b*) EFFECT OF SPECIFIC GRAVITY ON RATE OF FLOW OF GAS

With the same orifice pressure a light gas, such as coal gas, will flow through an orifice with greater velocity than a heavy gas, such as water gas. For a given pressure the rate of flow will vary inversely as the square root of the density as shown by the simple equation of velocity $V = 66.34\sqrt{\frac{H}{d}}$.

In Fig. 5 are given the results obtained with three different specific gravities of gas. The tests were made with an orifice pressure of 4 inches of water and with orifices of the No. 4 type. These tests were made on different days and the temperature and barometric pressure were slightly different. For this figure and in all the other values which have been reported the gas rates have been corrected to the usual standard conditions of 60° F and 30 inches of mercury pressure.

For example, if an orifice passed 100 cubic feet of gas per hour, of 0.65 specific gravity, the same orifice, with the same pressure,

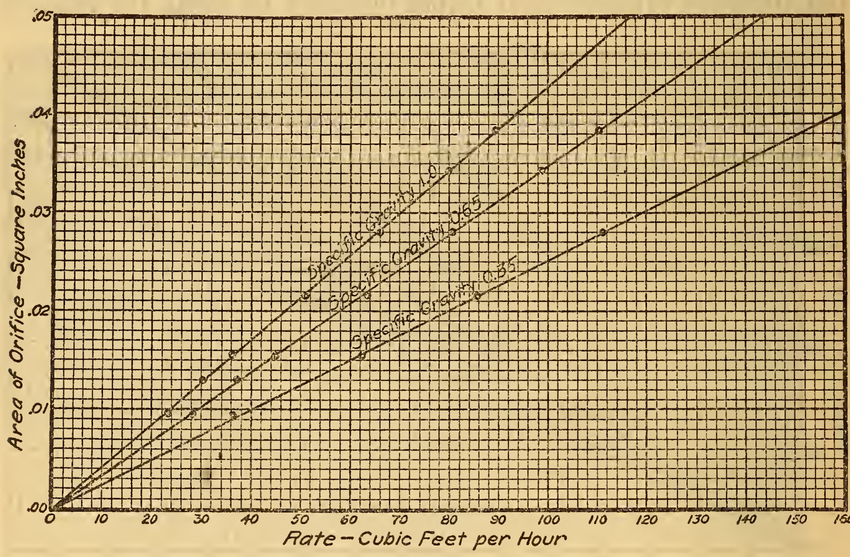


FIG. 5.—*Rates of flow, at 4 inches water pressure, of gases of 1.0, 0.65, and 0.35 specific gravity through orifices of type No. 4, Fig. 6*

would pass 136.3 cubic feet of gas per hour, of 0.35 specific gravity.

$$100 \times \sqrt{\frac{0.65}{0.35}} = 100 \times 1.363 = 136.3 \text{ cubic feet per hour.}$$

The curves in Fig. 5 give 138 cubic feet per hour, of 0.35 specific gravity, which is within about 1 per cent of the calculated value given above and is sufficiently accurate for practical purposes.

Some of the slight discrepancies of the relative gas rates are accounted for by the fact that the corrected value represents the volume of the gas that passed through the orifice, when corrected to the standard condition. The volume that would have passed through the orifice if the temperature and barometric pressure had been actually 60° F and 30 inches of mercury pressure would be slightly different on account of the difference in density.

The change in specific gravity with any one type of gas is rarely sufficient to cause any difficulty in appliances, but where mixtures of two or more gases are being distributed, it becomes of considerable importance. The effect of specific gravity will be discussed further under air entrainment and under Section VII, "The burner tube and burner ports."

2. REASON FOR ORIFICE INVESTIGATION

Most of the experiments on flow of air through orifices, as reported by various investigators, have been made with orifices larger than those commonly used in gas appliances and with pressures which were considerably greater, and for that reason very little of the published data is applicable to this problem.

In the Gas Engineering Journal of December 1, 1917, Riley and Wilson give some values for the flow of gas of 0.550 specific gravity through orifices of various sizes. Calculating a few of these values, the discharge coefficient was found to be about 0.908. In the N. C. G. A. Practical Gas Educational Courses the tables of rates of flow through orifices with water gas of 0.65 specific gravity give a discharge coefficient of 0.624. The reason for the wide difference in these values is not evident from the published data.

In view of the wide difference in these values of the discharge coefficient, as indicated by the above data, it was considered advisable to make up various types of orifices, and to determine their relative rates of flow and their discharge coefficients. It was also thought to be of great importance to determine whether the different types of orifices would give the same relative air entrainment, since it seemed probable that with different types of orifices the shape of the gas stream would be somwehat different, and the loss of energy of the gas stream in passing through the orifice might be different, both of which would affect the air entrainment.

In the first part of this investigation the common type of orifice which has been designated as the "channel" type was used. The approach was made with a large-size drill of standard form, with an angle of approach of about 59°. To obtain orifices of different rates of flow drills of different sizes were used. An effort was made to make the channel of each orifice about one-tenth inch long. It seemed impossible, however, by this method to make two orifices of the same size that would have the same rate of flow.

It was observed shortly after this investigation was started that the gas stream must be directed exactly into the center of the mixing tube, otherwise the air injection would be reduced. One orifice in particular did not inject the air that it should; yet to the eye it appeared to be just like the others in every respect. Its inconsistency was not understood until a stream of water was forced through it. This showed that the stream was directed off center several degrees and explained why there was a large loss in air injection. No trouble of this kind was experienced with orifices cut accurately on a lathe.

Due to the large difference in rates of flow and the difference in air injection of the various types of orifices, it was thought necessary to investigate this phase of the problem quite thoroughly before taking up the investigation of the burner itself. So far

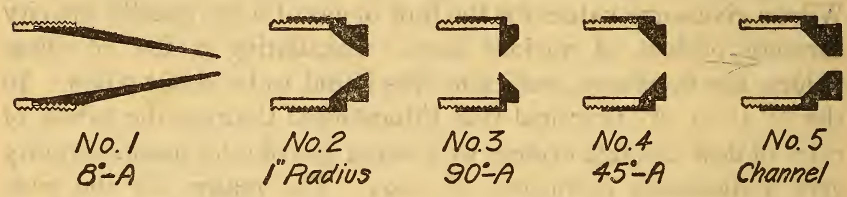


Fig. 6.—*Design of orifices. Orifices Nos. 1, 2, 3, and 4 are designated as "sharp edge" type, orifice No. 5 as "channel" type*

as we know a careful study of the discharge coefficient of various types of gas orifices and the relative air injection of each has never before been made.

3. TYPES OF ORIFICES INVESTIGATED

In Fig. 6 are shown five of the different types of orifices investigated. No. 1 has an 8° angle of approach; No. 2 was cut with a special tool that made the wall of the approach a curve of 1 inch radius; No. 3 is a sharp-edge orifice with a 90° approach and a 45° outlet; No. 4 has a 45° approach. These four types of orifices have been designated as sharp-edge orifices. No. 5 has a 45° approach and a channel of varying length as described in the text. This type of orifice we have called the channel type. Modifications of No. 5 are most commonly used.

(a) THE CHANNEL ORIFICE

The most common type of orifice and one that is used in most gas appliances is the type having a channel or short tube. In this class will be included all gradations from a sharp-edge orifice to one having a channel or a tube, such as would be formed when an ordinary plug is drilled for an orifice.

The channel type of orifice may have a channel of varying length, and it may be made with different lines or angles of approach; changing either will cause a large variation in the rate of flow.

To show the effect of length of channel, orifices with a 45° angle of approach of the type shown by No. 5 in Fig. 6 were used. The orifices were made the size of No. 2 drill which has a diameter of 0.221 inch and an area of 0.0384 square inch.

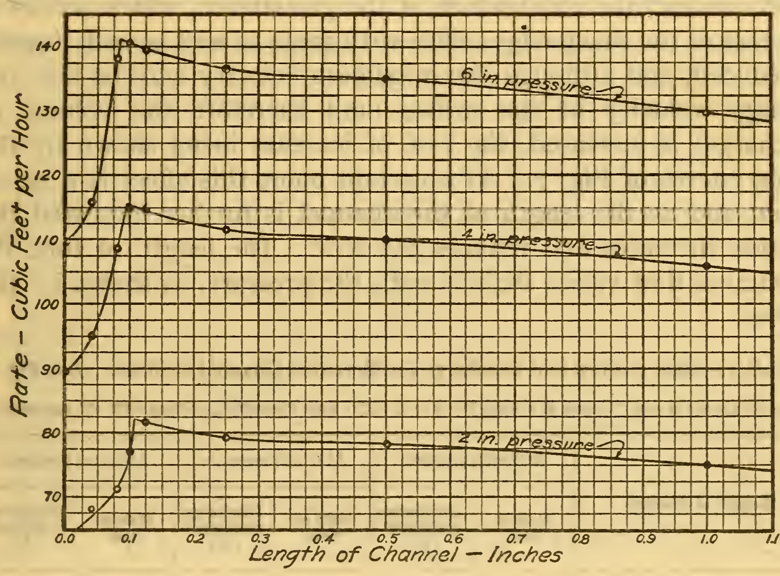


FIG. 7.—*Rates of flow of 1.0 specific gravity gas, at 2, 4, and 6 inches water pressure, through channel orifices of type No. 5, Fig. 6, with a 45° angle of approach and varying length of channel. Size of orifices, No. 2 drill*

Fig. 7 shows the very interesting characteristics of such an orifice and explains why in the preliminary investigation it was impossible to get the gas rates at the different pressures to agree with the simple theory which it seemed should apply. A study of this figure, the values of which were plotted from Table 1, shows that as the length of channel is increased, beginning with a sharp edge, the gas rate rapidly increases and reaches a maximum at a certain length of channel and then gradually decreases. It should be noted, however, that with the higher pressure the gas rate increases more rapidly than it does at the lower pressure, and the length of channel that gives the maximum rate is not quite the same for lower pressures as it is for the higher pressures; for example, with the 0.1 inch length of channel, at 2 inches orifice pressure, the gas rate is below the maximum point, and at 6 inches the same orifice gives a gas rate which is slightly beyond the maximum rate. In other words, the length of channel which will give the maximum rate is slightly different for different pressures.

The reason for the rapid increase in rate of flow when a short-channel orifice replaces the sharp-edge orifice is that with the short channel the constriction of the gas stream, which occurs in the case of the sharp-edge orifice with gases as well as with liquids, is reduced, and although the maximum velocity may be less, the average velocity at the orifice (and therefore the volume of discharge) is increased, the rate of increase being shown by the rising curves of Fig. 7. At a certain point this effect is a maximum, and as the length of the channel is further increased the friction increases and reduces the flow. The length of tube for maximum flow varies slightly with the pressure, as shown by the curves.

TABLE 1.—Rate of Flow and Discharge Coefficient of Channel Orifices. (See Fig. 7)

[Specific gravity of gas, 1.0; size of orifice, No. 2 drill (0.221 inch diameter); orifices with 45° approach]

Length of channel	At 2-inch pressure		At 4-inch pressure		At 6-inch pressure	
	Rate of flow	Discharge coefficient (K)	Rate of flow	Discharge coefficient (K)	Rate of flow	Discharge coefficient (K)
Inch	Foot³/hour		Foot³/hour		Foot³/hour	
0.00	63.6	0.705	89.5	0.703	109.6	0.703
.04	68.5	.760	95.1	.747	115.6	.741
.08	71.3	.792	108.6	.853	138.1	.886
.10	77.1	.855	114.9	.903	140.6	.902
.125	81.6	.906	114.6	.901	139.2	.893
.25	79.2	.879	111.4	.875	136.4	.875
.50	78.2	.868	110.0	.865	135.0	.865
1.00	75.0	.832	105.7	.831	129.7	.832

It will be noted in Table 1 that the orifice of 0.08 inch length of channel has a discharge coefficient at 2 inches pressure of 0.792; at 4 inches, 0.853; and at 6 inches, 0.886, although for other lengths the differences due to the variation of pressure are less. It follows, then, that it is impossible to design an orifice of the channel type within the critical length—namely, 0.00 to about 0.125 inch—that will give a perfectly constant discharge coefficient, although the small variations with pressure would probably not be objectionable in practice. No measurements have as yet been made of other sizes of channel orifices. The effect of length of channel on the gas rate will undoubtedly be similar, but the location of the maximum point may vary somewhat with the size.

(*b*) THE SHARP-EDGE ORIFICE

The tests with the channel orifice gave such interesting results that it seemed desirable to investigate various forms of the sharp-edge type. Fig. 8 shows the rates of flow of four types of sharp-edge orifices at 4 inches pressure with gas of 0.65 specific gravity.

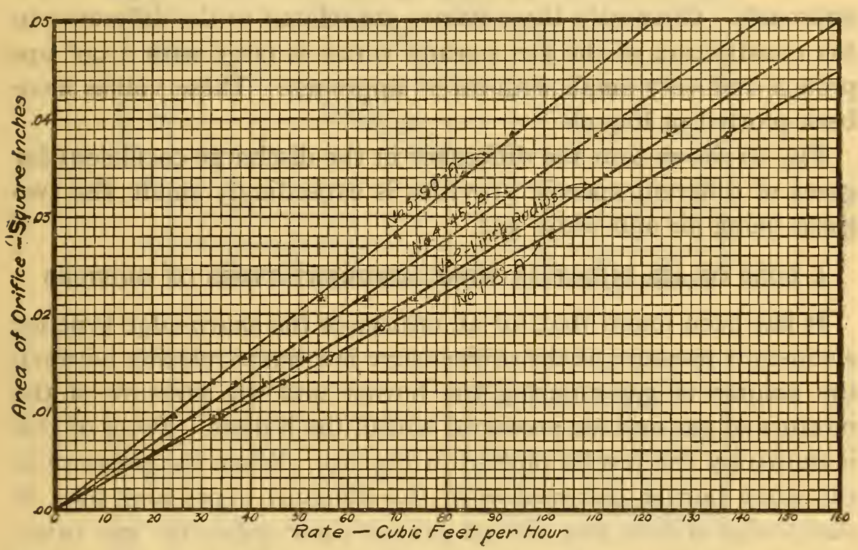


FIG. 8.—*Rates of flow of 0.65 specific gravity gas, at 4 inches water pressure, through various sizes of sharp edge orifices of types Nos. 1, 2, 3, and 4, Fig. 6*

Orifices have been tested over a range of pressure from 1 to 6 inches and with sizes from No. 2 to No. 53 drill, and it has been found that there is a linear relation between the area of the orifice and the gas rate for orifices of types 1, 3, and 4. (See Fig. 6.) From this it follows that the discharge coefficient for each one is constant over this range of sizes, although the discharge coefficient is considerably different for each type.

The "1-inch radius" type shows a slight curve, which is accounted for by the fact that all the orifices of this type were made with the same tool and the angle of approach at the immediate point of discharge was different for each size of orifice. The discharge coefficient for this type of orifice is, therefore, not a constant.

Since it had been found that the length of channel was important, it seemed desirable to investigate the effect of the angle of approach with the sharp-edge type. In Fig. 9*a* the rates of flow of orifices of the size of No. 2 drill have been shown for 2 and 4 inch pressures for seven different angles of approach. Tests are shown with two different specific gravities of gas. It is apparent from this figure that the rate of flow of the orifice varies greatly with different angles of approach.

From this figure the discharge coefficient (K) for the seven different orifices has been computed. The discharge coefficient varies from about 0.605 for a 90° approach to about 0.875 for an 8° approach. Obviously these values are related to the difference in the constriction of the gas stream, which is large with a 90° approach and very small with an 8° approach. These values have been plotted in Fig. 9*b*.

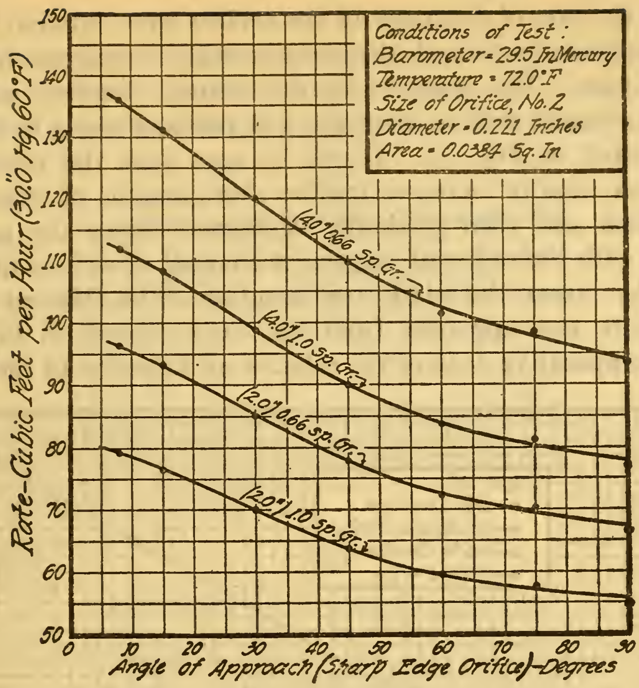
Fig. 9*b* shows that the difference in the discharge coefficient for gases of different specific gravities is exceedingly small, the two gases being air and water gas.

4. LOSS OF AIR INJECTION WITH DIFFERENT TYPES OF ORIFICES

It has been found that for all orifices of the sharp-edge type for a constant pressure at the orifice there is a linear relation between the volume of gas entering the burner and the pressure of the mixture of gas and air produced within the burner. Proof of this is shown by the results plotted in Fig. 10. When the pressure in the same burner obtained with the different types and sizes of sharp-edge orifices was plotted against their respective gas rates, the points all fell on the same straight line. It follows, then, that if the orifices were of such a size that each delivered the same quantity of gas per hour under the same orifice pressure—that is, each had the same rate of flow—the pressure in the burner would be the same with each type of orifice and the volume of air injected with each would consequently be the same.

Orifices of the channel type invariably produced a pressure in the burner which was less than that produced by the sharp-edge type, when the gas rate of the two types was the same and was produced by the same orifice pressure. This shows that the air injected by the channel orifice is always less than by the sharp-edge type, with the same pressure and gas flow. This is because the maximum velocity of the gas, and therefore its momentum, is greater with the sharp-edge orifice than with the channel orifice, and hence its air-entraining power is greater.

A



B

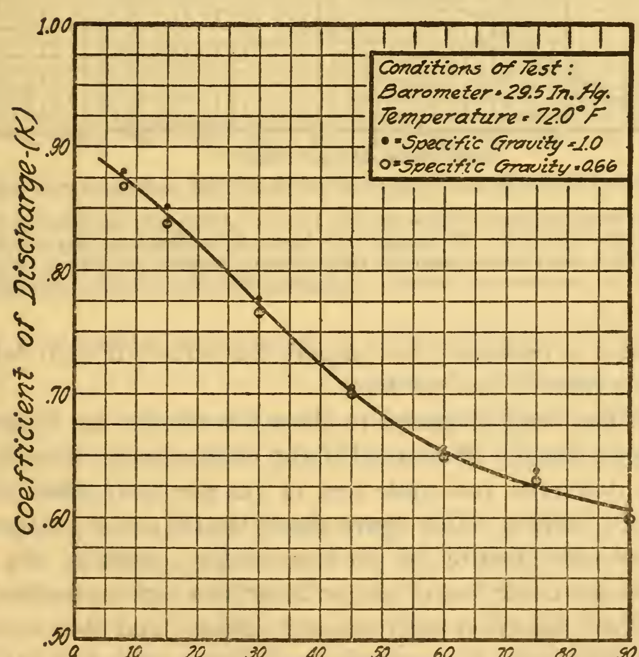


FIG. 9.—*Graphs of discharge rates for sharp edge orifice*

A. Rates of flow of 1.0 and 0.65 specific gravity gas, at 2 and 4 inches water pressure, through sharp edge orifices, No. 2 drill size, with different angles of approach

B. Discharge coefficients of sharp edge orifices with different angles of approach

When the rate of discharge of the orifices with different lengths of channel was determined, the pressure in the burner was recorded in each case. The pressure in the burner obtained with the different channel orifices when the gas pressure was 4 inches has been plotted in Fig. 11*a*. It will be seen that the pressure in the burner rapidly increases for the short lengths of channel up to 0.1 inch and then gradually decreases. Since the gas rate changed with the different lengths of channel as well as the pressure in the burner, the relative air injection of the different orifices is therefore not apparent from the values shown in Fig. 11*a*. To get comparative data of the relative air injection of the differ-

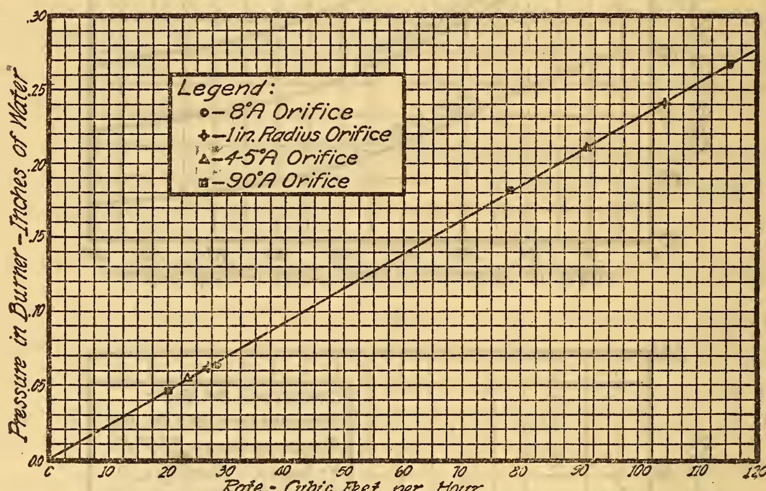


Fig. 10.—*Pressure in the burner for different gas rates with sharp edge orifices*

For these orifices any given gas rate gives the same pressure in the burner, thus indicating that the air injected is the same in each case. The pressure in the burner at the different gas rates plots on the same straight line, which shows that for sharp edge orifices there is a constant ratio between the gas rate and the pressure in the given burner and, therefore, the resistance to the flow of the gas is proportional to the gas rate

ent orifices it is necessary to compare the orifices if each delivered the same volume of gas per hour.

Fig. 11*b* has been prepared to show the relative air injection of the different lengths of channel if the orifices were of such a size that each delivered 100 cubic feet of gas per hour when under a pressure of 4 inches. This figure shows that if a sharp-edge orifice delivers 100 cubic feet of gas per hour under a pressure of 4 inches and injects 400 cubic feet of air per hour into a given burner, there is a loss of air injection with channel orifices, and that a channel orifice 0.1 inch long, for example, will inject only 381 cubic feet of air per hour under the same conditions.

It will be observed from this figure that the loss of air injection increased rapidly for the short lengths up to 0.1 inch and for further increase of length of channel the loss is not so marked.

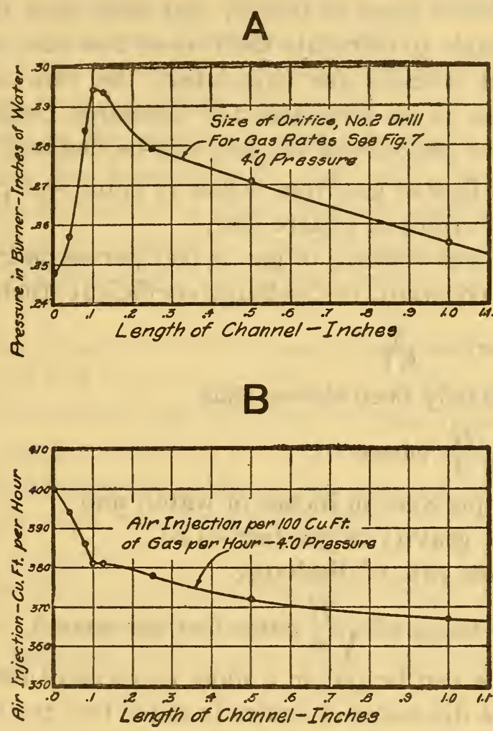


FIG. 11.—*Graphs of pressure and air injection for different lengths of channel*

A. Pressures developed in burner by 1.0 specific gravity gas at 4 inches water pressure flowing through No. 2 drill size orifices with different lengths of channel

B. Air injected by gas at 4 inches water pressure flowing at the rate of 100 cu. ft. per hour from No. 2 drill size orifices with different lengths of channel

5. HOW TO CALCULATE THE RATE OF FLOW AND SIZE OF GAS ORIFICES

If the rate of flow of an orifice is desired, or the size of an orifice is required for any given gas rate, it is necessary to know the gas pressure, the specific gravity of the gas, and the discharge coefficient of the orifice.

The N. C. G. A. educational course on industrial fuel, in lesson paper No. 5, Fig. 12, gives a chart for computing the sizes and rates of flow of orifices; but this chart is not of very great value unless one knows the discharge coefficient for the type of orifice that is to be used. In the same lesson paper there is also given a table of rates of flow for "thin orifices, 0.22 inch thick." We have previously stated that from this table of rates of flow of orifices the discharge coefficient was found to be about 0.624.

From the data and curves given in the preceding pages of this paper it should be possible to estimate very closely the discharge coefficient for any type of orifice, and with that information it should be possible to calculate the rate of flow and size of orifices.

A simplified formula for calculating the rate of flow of an orifice in terms of its area and the discharge coefficient, and of the density and pressure of the gas, can be deduced as follows:

Let q = rate of flow of gas from orifice in cubic feet per second;
a = area of orifice in square feet;
V = theoretical velocity of gas in feet per second;
K = orifice constant, or discharge coefficient; then

$$q = aKV \text{ or } a = \frac{q}{KV}$$

It has previously been shown that

$$V = 66.34\sqrt{\frac{H}{d}}, \text{ where}$$

H = orifice pressure in inches of water, and
d = specific gravity of gas (air = 1.0).

Therefore, the rate of discharge

$$q = aKV = 66.34\, aK\sqrt{\frac{H}{d}} \text{ cubic feet per second.}$$

This formula can be put in a more convenient form as follows:

Let Q = rate of discharge of orifice in cubic feet per hour, and
A = area of orifice in square inches; then

$$Q = 3600 \times 66.34\, \frac{A}{144}\, K\sqrt{\frac{H}{d}}$$

$$= 1658.5\, AK\sqrt{\frac{H}{d}}; \text{ or}$$

$$A = \frac{Q}{1658.5\, K}\sqrt{\frac{d}{H}}$$

In this form the formula will enable one to calculate readily any one unknown value by substituting in the formula the known values.

V. PRINCIPLES GOVERNING AIR INJECTION

1. AIR ENTRAINMENT AND MOMENTUM OF THE GAS STREAM

The momentum of a body is equal to its mass times its velocity. This is true with gases as well as solids and liquids. In an atmospheric burner a part of the momentum of the gas stream is communicated to the air; therefore the quantity of air injected will depend upon the velocity of the gas through the orifice, the rate of flow, and the specific gravity of the gas. There is, of course, always some loss in momentum due to friction and eddy currents, and the total momentum of the mixture is, therefore, less than the momentum of the gas stream at the orifice.

How much of the momentum of the gas stream is converted into the momentum of the mixture will depend upon the design of the injecting tube and the resistance of the burner. In general, for any one burner, the injecting tube that will produce the largest pressure in the burner for a given pressure at the orifice will be the most efficient air injector. The resistance of the burner will depend upon the shape and size of the burner and the area of the ports.

The numerous experiments have conclusively demonstrated that if a given rate of flow of gas issuing from an orifice under a pressure of 2 inches, for example, will inject two volumes of air to one of gas into a given burner, and the gas pressure is increased, thereby increasing the gas rate, the volume of air injected will be increased, but the air-gas ratio will remain the same as it was before the change of gas pressure.

To show the effect of changing the gas pressure, specific gravity of the gas, and the gas rate on air entrainment, there are given below several problems with values taken from the experimental data.

2. RELATION BETWEEN THE MOMENTUM OF THE GAS STREAM AND THE MOMENTUM OF THE STREAM OF THE MIXTURE

The tests of the different pipe burners have shown that for any given burner operated over the range of gas pressure from 2 to 6 inches the ratio of the momentum of the gas stream to the momentum of the stream of the air-gas mixture is practically a constant, irrespective of (1) a change of gas pressure, (2) a change of gas rate, and (3) a change of specific gravity of gas. Then if

m = mass of gas per second that issues from an orifice,

M = mass of air per second that is injected into a given burner,

V = theoretical velocity of gas stream in feet per second $= \sqrt{2gh}$,

v = velocity at first port of the air and gas mixture in feet per second,

mV = the momentum of gas stream, and

$(M+m)v$ = the momentum of mixture of air and gas at first port, therefore, as stated above,

$$\frac{mV}{(M+m)v} = \text{a constant ratio for any given burner.}$$

A simplified formula can be developed as follows, which will greatly aid the reader to apply this useful information in calculating practical problems: Let

q = gas rate in cubic feet per second = cubic feet per hour ÷ 3600;
W = mass of 1 cubic foot of air at 30 inches Hg and 60° F;
d = specific gravity of gas (air = 1.0);
r = air to gas ratio of mixture passing through burner;
x = cross-sectional area of pipe in square feet (0.01038 square foot for 1¼ inch pipe burner);
R = ratio of momentum of gas stream to momentum of stream of mixture;
V = theoretical velocity of gas stream in feet per second;
m = mass of gas stream per second = qdW;
mV = momentum of gas stream = $qdWV$ foot pounds per second;
rq = volume of air injected into the burner per second;
M = mass of air per second = rqW;
v = velocity of the mixture at first port, which is the volume of air per second in cubic feet + volume of gas per second in cubic feet ÷ cross-sectional area of pipe in square feet,

$$= \frac{rq+q}{x} = q\,\frac{(r+1)}{x} \text{ feet per second; and}$$

$(M+m)\,v$ = the momentum of the mixture

$$= (rqW+qdW)\,\frac{q\,(r+1)}{x} = \frac{qW\,(r+d)\,q\,(r+1)}{x} \text{ foot pounds per second:}$$

Therefore the ratio of momentum of gas stream to momentum of stream of mixture

$$R = \frac{mV}{(M+m)\,v} = \frac{qdWV}{\dfrac{qW\,(r+d)\,q\,(r+1)}{x}} = \frac{dVx}{q\,(r+d)\,(r+1)}$$

It is perhaps more convenient to use the gas rate in terms of cubic feet per hour, and the cross-sectional area of the pipe in square inches; then

$$R = \frac{3600}{Q}\,\frac{X}{144}\,\frac{dV}{(r+d)\,(r+1)} = \frac{25XdV}{Q\,(r+d)\,(r+1)}, \text{ where}$$

Q = the gas rate in cubic feet per hour, and
X = cross-sectional area of pipe in square inches = 1.496 square inches for 1¼ inch pipe burner.

Also the equation is much more readily applied if, in place of V, its value, $66.34\sqrt{\frac{H}{d}}$, is substituted, where H = the pressure of the gas in inches of water. Then

$$R=\frac{25Xd\times 66.34\sqrt{\frac{H}{d}}}{Q\ (r+d)\ (r+1)},$$
$$=\frac{1658.5\ X\ \sqrt{Hd}}{Q\ (r+d)\ (r+1)}.$$

The following examples show the ratio between the momentum of the gas stream and the momentum of the stream of the mixture in which pipe burners were used. The burners were equipped with fixed orifices. It should be evident that changing either the design of the injecting tube or the port area of the burner head, as well as a change of adjustment of the air shutter if one is used, will cause a change in the volume of air injected into the burner, therefore, a change of the ratio (R). Also a change of the type of orifice may cause a change of the momentum of the gas stream which in turn will cause a change of the ratio R. It follows that each particular burner has a constant ratio R, if no change or adjustment of the burner parts is made.

EXAMPLE No. 1.—To illustrate that for a given burner the ratio R between the momentum of the gas stream and the momentum of the stream of the mixture is a constant when the gas rate is changed but the pressure remains constant. The gas rate and the air-gas ratio values for both of the following conditions were taken from "(4 inch) no tube ratio" curve of Fig. 26. The gas was 0.65 specific gravity. The burner was a 1¼ inch pipe burner with 1.05 square inch port area.

	Case I	Case II
Gas rate (cubic feet per hour) ....(Q)	47.5	67.0
Pressure of gas (inches of water).....(H)	4.0	4.0
Specific gravity of gas..................(d)	.65	.65
Air-gas ratio of mixture...................(r)	3.72	3.00
Cross-sectional area of pipe (sq. in.)...(X)	1.496	1.496
Ratio $R=\frac{1658.5X\sqrt{Hd}}{Q\ (r+d)\ (r+1)}$	$\frac{1658.5\times 1.496\sqrt{4.0\times 0.65}}{47.5\ (3.72+0.65)\ (3.72+1)}=4.08$	$\frac{1658.5\times 1.496\sqrt{4.0\times 0.65}}{67.0\ (3.0+0.65)\ (3.0+1)}=4.09$

EXAMPLE No. 2.—To illustrate that for a given burner the ratio R between the momentum of the gas stream and the momentum of the stream of the mixture is a constant when the gas pressure is changed, but the gas rate is kept constant. These values were also taken from the "no tube ratio" curves of Fig. 26.

	Case I	Case II
Gas rate (cubic feet per hour)..........(Q)	47.5	47.5
Pressure of gas (inches of water).....(H)	2.0	4.0
Specific gravity of gas....................(d)	.65	.65
Air-gas ratio of mixture..................(r)	3.00	3.72
Cross-sectional area of pipe (sq. in.)...(X)	1.496	1.496
Ratio $R=\frac{1658.5X\sqrt{Hd}}{Q(r+d)(r+1)}$	$\frac{1658.5\times1.496\sqrt{2.0\times0.65}}{47.5\ (3.0+0.65)\ (3.0+1)}=4.08$	$\frac{1658.5\times1.496\sqrt{4.0\times0.65}}{47.5\ (3.72+0.65)\ (3.72+1)}=4.08$

EXAMPLE NO. 3.—To illustrate that for a given burner the ratio R between the momentum of the gas stream and the momentum of the stream of the mixture is a constant when the specific gravity

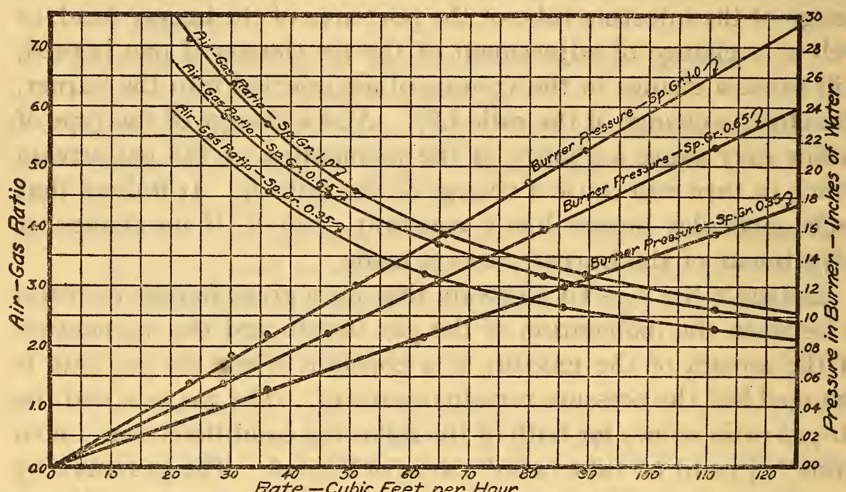


FIG. 12.—*Air injection and pressure in burner obtained with 1.0, 0.65, and 0.35 specific gravity gases flowing through sharp edge orifices at 4 inches water pressure*

Burner used is a 1¼-inch pipe burner of 0.75 square inch port area fitted to improved injector

of the gas is changed. The values are taken from the 0.35 and 0.65 specific gravity of gas curves of Fig. 12.

	Case I	Case II
Gas rate (cubic feet per hour)..........(Q)	45	45
Pressure of gas (inches of water).....(H)	4.0	4.0
Specific gravity of gas....................(d)	.35	.65
Air-gas ratio of mixture..................(r)	4.0	4.6
Cross-sectional area of pipe (sq. in.)...(X)	1.496	1.496
Ratio $R=\frac{1658.5X\sqrt{Hd}}{Q\ (r+d)\ (r+1)}$	$\frac{1658.5\times1.496\sqrt{4.0\times0.35}}{45.0\ (4.0+0.35)\ (4.0+1)}=3.00$	$\frac{1658.5\times1.496\sqrt{4.0\times0.65}}{45.0\ (4.6+0.65)\ (4.6+1)}=3.02$

3. HOW TO CALCULATE THE AIR-GAS RATIO (r) OF A GIVEN BURNER

It should be evident that if the ratio R of the momentum of the gas stream to the momentum of the stream of the mixture is known for a given burner, it is possible from the equation which has been developed to solve for any one unknown value. In the equation $R=\frac{1658.5\,X\sqrt{Hd}}{Q(r+d)(r+1)}$, probably the most important unknown value to solve would be the air-gas ratio (r). Solving the above equation for air-gas ratio

$$r=\sqrt{\frac{1658.5X\sqrt{Hd}}{RQ}+0.25d^2-0.5d+0.25}-0.5d-0.5$$

It has been shown that R for the burner used to determine the "no tube ratio" curves of Fig. 26 has a value of 4.08. Suppose it is desired to know air-gas ratio (r) where the gas pressure $H=2.0$ inches, and the rate of flow $Q=50.0$ cubic feet per hour. The specific gravity of the gas $d=0.65$, and the cross-sectional area of the pipe burner $X=1.496$ square inches. Then,

$$r=\sqrt{\frac{1658.5\times 1.496\sqrt{2.0\times 0.65}}{4.08\times 50.0}+0.25\,(0.65)^2-(0.5\times 0.65)+0.25}$$
$$-(0.5\times 0.65)-0.5=2.91$$

This agrees with the air-gas ratio shown by the "(2-inch) no tube ratio" curve of Fig. 26, which was obtained with 2 inches pressure and the above burner.

4. RELATION BETWEEN GAS PRESSURE AND MOMENTUM OF GAS STREAM

From the equation $V=66.34\sqrt{\frac{H}{d}}$ it is seen that the velocity of a gas stream varies as the square root of the gas pressure, and since the mass issuing from the orifice varies as the velocity of the gas stream, the momentum of the gas stream is directly proportional to the pressure.

For example, let us assume that at 1-inch pressure the volume of gas of d specific gravity issuing from an orifice is q cubic feet per second, and the velocity of the stream is V feet per second. The momentum of the gas stream will be mass $\times$ velocity $=(qdW)\,V$ foot pounds per second, where $W=$ the mass of a cubic foot of air at 30.0 inches Hg and 60° F.

If the pressure is now increased to 4 inches the velocity will be doubled, therefore the volume going through per second will be doubled, which doubles the mass per second, and the momentum will be

$$(2qdW)2V = 4(qdW)V \text{ foot pounds per second.}$$

5. EFFECT OF CHANGE OF SPECIFIC GRAVITY ON THE MOMENTUM OF THE GAS STREAM

If two orifices are of such a size that when under the same pressure each delivers the same volume of gas per hour and the gases are of different specific gravity, the momenta of the two gas streams are proportional to the square roots of their gravities; for example, compare gases of 0.35 and 0.65 specific gravities issuing from different orifices under the same pressure. If q cubic feet per second issues from each under 1-inch pressure, the momentum of the 0.65 specific gravity gas will be $qdWV = q \times 0.65 \times W \times 82.3 = 53.5\ qW$ foot pounds per second, where $V = 82.3$ feet per second.

The velocity of the 0.35 specific gravity gas will be $82.3 \times \sqrt{\frac{0.65}{0.35}} = 112.1$ feet per second.

The momentum of the 0.35 specific gravity gas will be $q \times 0.35 \times W \times 112.1 = 39.25\ qW$ foot pounds per second and $\sqrt{\frac{0.65}{0.35}}$ is proportional to $\frac{53.5\ qW}{39.25\ qW}$.

6. RELATION BETWEEN MOMENTUM OF MIXTURE AND PRESSURE IN THE BURNER

It has been proved that there is a definite ratio between the momentum of the gas stream and the momentum of the stream of the mixture. For any increase in the gas rate there is a corresponding increase in the momentum of the stream of the mixture, resulting in an increase in the pressure in the burner. Fig. 12 shows the pressures in the burner with three specific gravities of gas, and Figs. 21 to 27, inclusive, show the pressures in different burners with different gas rates and gas pressures.

7. THE VARIOUS RELATIONS SUMMARIZED

From the fundamental theory and the relations which have been illustrated in the preceding examples it is possible to summarize the most important relations as follows:

1. The ratio between the momentum of the gas stream and the momentum of the stream of the mixture is always the same for a burner of a given design irrespective of orifice pressure, specific gravity of gas, or the volume of the air and gas mixture going through the burner.

2. Where the gas rate is increased by change of pressure, the momenta of the gas streams are directly proportional to the pressures.

3. When the same volumes of gases of different specific gravities issue from different orifices under the same orifice pressure the momenta of the gas streams are proportional to the square roots of the specific gravities of the gases.

4. When the pressure is changed to give the same gas rate for gases of different specific gravity, the air entrainment is proportional to the specific gravities.

5. The pressure at any one point in the burner increases in direct proportion to the increase in the momentum of the stream of the mixture.

If a change in gas pressure or gas rate should cause a considerable change in the temperature of the burner, the volume of air injected would be slightly different from the calculated values for the new condition, and there would not be quite the agreement between the above stated relations.

VI. THE INJECTING TUBE

1. THE ADVANTAGE OF AN INJECTING TUBE

When a gas under pressure issues from an orifice, the momentum of the gas causes the surrounding air to be set in motion in the direction of the gas stream. Because of the great velocity of the gas stream eddy currents are set up in the surrounding air. This eddy-current motion consumes much of the energy of the gas.

The function of an injector is to eliminate, as much as possible, this eddy-current movement, and to give a ratio of air to gas suitable for the given purpose.

The idea of a proper constriction in the mixing tube of the burner in order to secure better air injection is not new. In fact nearly all domestic range burners are so designed, although exact information as to the best form of injector for gas burners has been wanting.

In atmospheric burners the flame characteristics of combustible gases vary with the change of ratio of primary air to gas. When the air-gas ratio of the mixture entering the burner is low, the combustion at the ports is slow, the flame is long, and the heat liberated per unit of flame area is low, relative to that of a mixture of a higher air-gas ratio. When the air-gas ratio of the mixture entering the burner is increased, the rate of combustion at the ports is increased, the flame height is decreased, the flame area is decreased, and the heat liberated per unit of flame area is increased. The higher air-gas ratio condition permits of greater concentration of heat, and, therefore, greater rate of consumption per unit of burner surface.

By the use of a correctly designed injecting tube it is possible to secure—(1) increased efficiency, resulting from higher flame temperature, and (2) increased rate of consumption, resulting from the injection of a greater quantity of air.

There is no better demonstration of the practical value of the use of an injector than that made by Thompson King, of the Consolidated Gas & Electric Co., Baltimore, Md., during the extremely cold winter of 1917–18. Many installations of gas-burning house-heating boilers in that city were entirely inadequate to supply the required heat during the severe weather, and in order to secure more heat from the boilers it was necessary to increase the gas consumption of the burners. This was accomplished by inserting injecting tubes made out of tin into the mixing tubes of the burners, whereby the rate of consumption of the boilers was increased sufficiently to give entirely satisfactory results.

2. INJECTOR DESIGN AND POSITION OF ORIFICE

Bearing in mind the advantages of an injector, it was decided to determine, if possible, what design of injector would cause the maximum air injection.

To eliminate the effect of as many variables as possible, all of the preliminary investigations were made with one burner, and the design of the injector was varied to determine the most efficient injector for that burner. The burner selected was a 1¼-inch pipe burner with a port area of 0.75 square inch. Since the specific gravity of the gas does not affect the design of the injector, air instead of combustible gas was used for the sake of convenience.

By using a constant orifice pressure the gas rate was kept constant. Observing the pressure obtained in the burner by the use of different injectors, the relative merits of different designs of injecting tubes could be determined. It is evident that with a constant gas rate the injector which gives the maximum pressure in the burner must necessarily inject the most air.

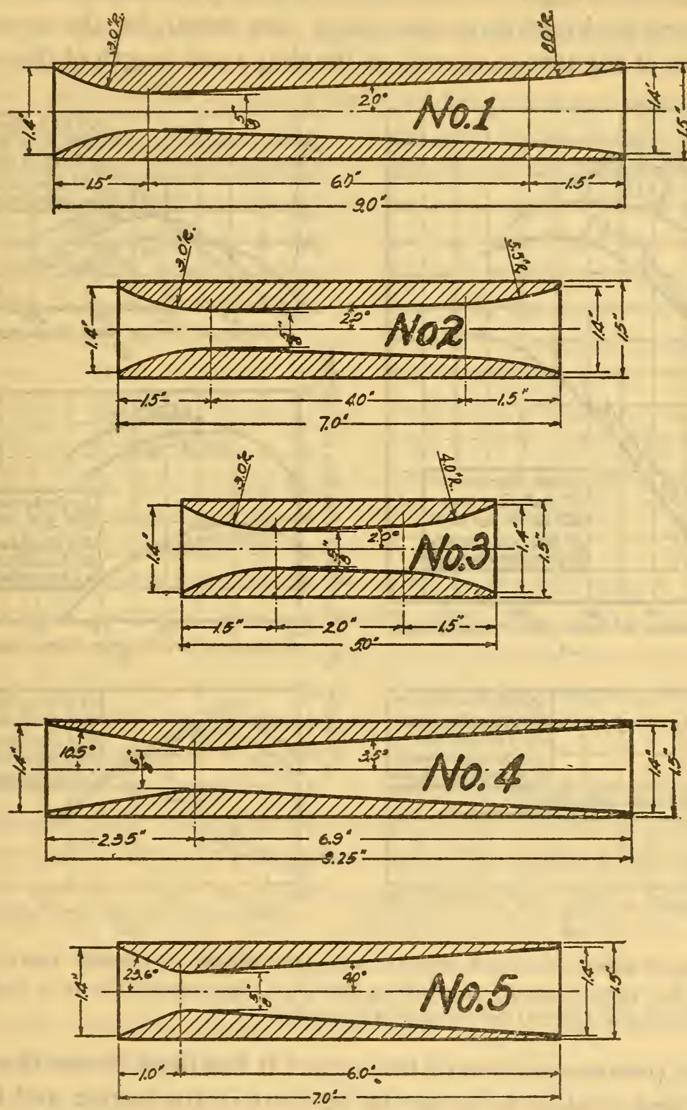


FIG. 13.—*Section views showing dimensions of five designs of injecting tubes*

In Fig. 13 and Table 2 are shown the dimensions of some of the injecting tubes that were made and tested. These injectors were made of brass and cut on a lathe. The end of the pipe burner, designated as the mixing tube, was reamed out, so that the injecting tube made a tight fit when inserted in the burner. Tube

No. 5 is the type of injector which was used in the first part of the investigation. It was soon realized that, in order to draw definite conclusions in regard to the design of the optimum injector, it was necessary to investigate the shape and length of the approach, the area of the throat, as well as the shape and length of the outlet.

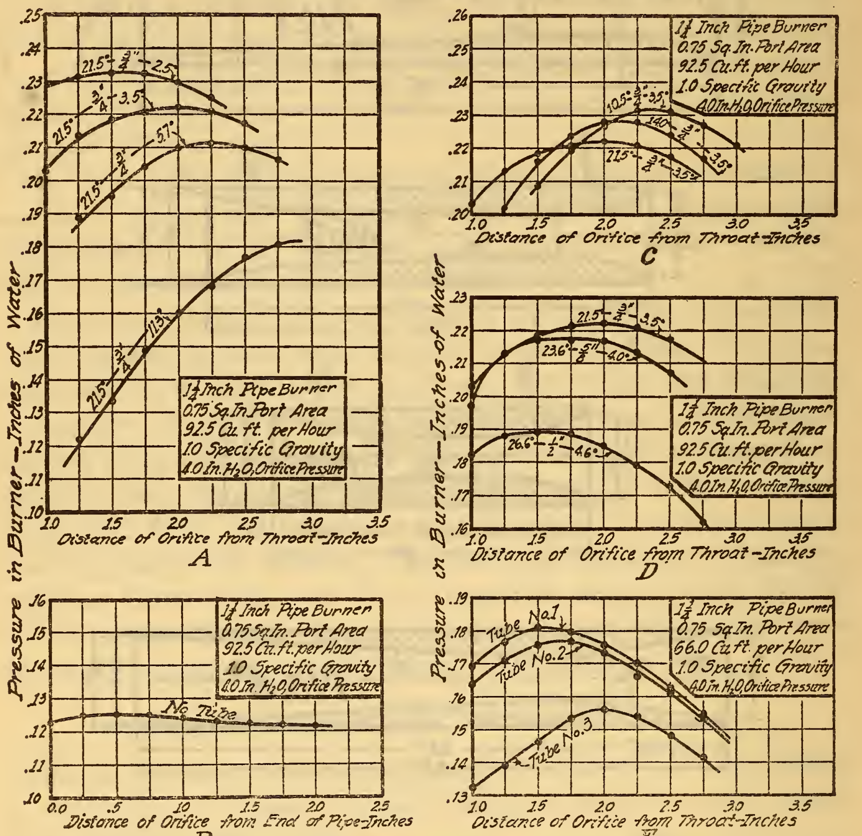


Fig. 14.—*Pressures developed in a 1¼-inch pipe burner, 0.75 square inch port area with 1.0 specific gravity gas flowing through a sharp edge orifice at 4 inches water pressure using different designs of injecting tubes*

In a previous section of this report it has been shown that there is a linear relation between the pressure in the burner and the gas rate in cubic feet per hour when the orifice pressure is constant. This pressure in the burner is known as the maximum pressure and was obtained by changing the distance of the orifice from the throat of the injector. The distance of the orifice from the throat is extremely important, and it seems that the best position of the orifice for any given injector can be determined only by experiment.

The curves of Fig. 14 will show fairly definitely the relation between the position of the orifice and the design of the injector. Once this position of maximum injection is obtained, it will hold good for that injector with any pressure at the orifice and with any gas rate.

The charts of Fig. 14 show the development of the investigation of the injector design. Chart B of this figure shows that the position of the orifice relative to the end of the burner is not so important when an unimproved injector is used. If the gas rate is kept constant at 92.5 cubic feet per hour and the different injecting tubes used in determining the curves of chart A are inserted into the burner, the advantage of an improved injector is clearly shown. These curves show the pressures obtained by the use of four injectors where the throat diameter and inlet angle were the same in each case, each injector having, however, a different outlet angle. The injector used to obtain the upper curve of this chart had an outlet length of 8 inches, while the outlet length of the injector producing the lower curve was 2 inches. It is interesting to note the positions of the orifice at which the maximum pressures for these two injectors were obtained. In the case of the former it was 1½ inches, while that of the latter was at least 3 inches. These curves contrast sharply the comparative values of long and short outlets obtained by changing the outlet angle, the longer outlet, which has the smaller slope, giving much better air injection.

The curves in chart C, Fig. 14, were obtained by using three injectors of the same throat diameter and outlet angle but of different inlet angle. The outlet length for these three injectors was 6 inches. The 10.5° inlet represents an inlet length of 2 inches and the 21.5° inlet represents an inlet length of 1 inch. This chart shows that the angle of approach is not so important as the outlet angle, and also that the position of the orifice for maximum injection changes as the inlet angle varies.

The curves of chart D, Fig. 14, are those of pressures obtained with three injectors, all of 1 inch length of inlet and 6-inch outlet length, in which the throat diameters were different. These curves are not of great value, as it was later found that in order to obtain the maximum injection of air in any burner there must be a definite relation between the area of the injector throat and the port area of the burner. The development of this relation is discussed under the flow capacity of an injector.

TABLE 2.—Dimensions and Designations of Various Designs of Injecting Tubes Tested in Pipe Burners of 1¼-Inch Diameter

[S=slope; R=radius]

Inlet dimension		Throat diameter	Outlet dimension		Injecting tube designation	Total length
Degrees slope or inches radius	Length		Slope angle	Length		
	Inches	Inch	Degrees	Inches		Inches
3″ R	1.5	5/8	2.0	6.0	3″ R-5/8-6″ (No. 1)	9.0
3″ R	1.5	5/8	2.0	4.0	3″ R-5/8-4″ (No. 2)	7.0
3″ R	1.5	5/8	2.0	2.0	3″ R-5/8-2″ (No. 3)	5.0
10°.5	2.40	5/8	3.5	6.85	10°.5-5/8-3°.5 (No. 4)	9.25
23°.6	1.0	5/8	4.0	6.0	23°.6-5/8-4°.0 (No. 5)	7.0
2″ R	1.30	5/8	3.5	6.85	2″ R-5/8-3°.5	8.15
2″ R	1.35	1/2	3.5	7.9	2″ R-1/2-3°.5	9.25
10°.5	2.7	1/2	3.5	7.9	10°.5-1/2-3°.5	10.6
26°.6	1.0	1/2	4.6	6.0	26°.6-1/2-4°.6	7.0
2″ R	1.25	3/4	3.5	6.0	2″ R-3/4-3°.5	7.25
10°.5	2.0	3/4	3.5	6.0	10°.5-3/4-3°.5	8.0
14°.0	1.5	3/4	3.5	6.0	14°.0-3/4-3°.5	7.5
21°.5	1.0	3/4	3.5	6.0	21°.5-3/4-3°.5	7.0
21°.5	1.0	3/4	2.5	8.0	21°.5-3/4-2°.5	9.0
21°.5	1.0	3/4	5.7	4.0	21°.5-3/4-5°.7	5.0
21°.5	1.0	3/4	11.3	2.0	21°.5-3/4-11°.3	3.0
6″ R	1.75	7/8	2.5	6.5	6″ R-7/8-2°.5	8.25
3″ R	1.4	7/8	3.5	4.85	3″ R-7/8-3°.5	6.25

3. THE OPTIMUM INJECTOR

From a study of the preceding charts, and from observations made on other injectors which it did not seem necessary to report, it is possible to draw some definite conclusions concerning the design of the optimum injector:

1. The change of the lines of the approach of the inlet to the lines of the outlet should be gradual.

2. The approach should follow approximately a curvature which should be not less than 3 inches radius for a ⅝-inch throat. Other sizes should be proportioned about the same.

3. The outlet angle should be about 2°.

It is, of course, true that in some installations the length of the injector is limited, and for that reason injectors Nos. 1, 2, and 3, shown in Fig. 13, were designed. A comparison of the results obtained with these three injectors is shown in chart E, Fig. 14. This chart shows that little is to be gained by making the 2° outlet slope longer than 4 inches, but that there is considerable dropping off in pressure in the burner if the length of the outlet slope is only 2 inches. Note that the position of the orifice giving maximum pressure in the burner changed with the variation in outlet length. The injecting powers of these injectors are easily compared from a study of Fig. 15. Any injector of different design that for a given gas rate would give a higher pressure in the burner than that shown by No. 1 in Fig. 13 would, of course, be a better injector.

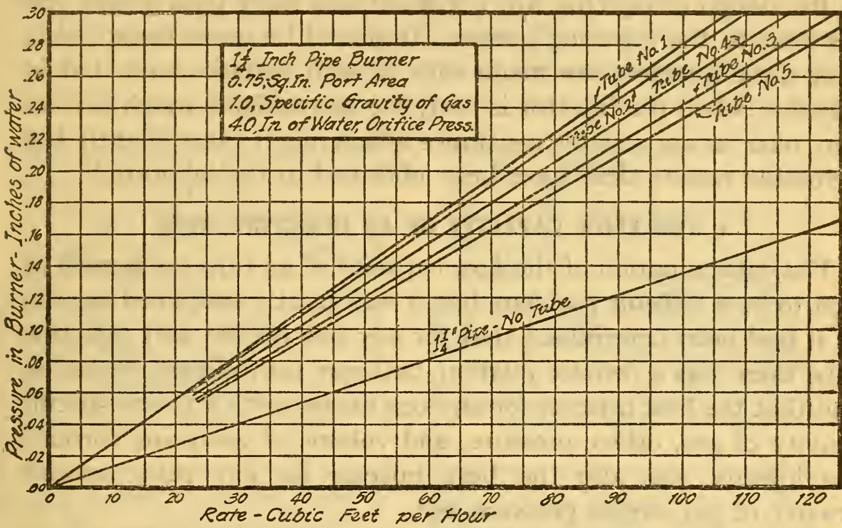


FIG. 15.—*Relative air injection with injecting tubes of different designs as indicated by the pressure in the burner*

The injecting tubes shown in Fig. 13 were used and are compared with the same burner when a 1¼-inch pipe served as an injecting tube.

In order to show the injecting powers represented by the pressure curves in Fig. 15 the following table has been prepared:

TABLE 3.—Relative Injecting Power of Different Injectors as Indicated by the Resulting Air-Gas Ratios

Type of injecting tube	Air-gas ratio		Type of injecting tube	Air-gas ratio	
	At gas rate 50.0 cubic feet per hour	At gas rate 75.0 cubic feet per hour		At gas rate 50.0 cubic feet per hour	At gas rate 75.0 cubic feet per hour
No injecting tube	3.25	2.45	No. 3	4.55	3.50
No. 1	5.00	3.85	No. 4	4.75	3.65
No. 2	4.90	3.80	No. 5	4.40	3.40

From a study of this table we note that in the case of this particular burner with a rate of 50 cubic feet per hour it is possible to inject 54 per cent more air by the use of injecting tube No. 1 than without the use of an injector. Injecting tube No. 1 is only about 2 per cent better as an injector than No. 2, about 9 per cent better than No. 3, 5 per cent better than No. 4, and 12 per cent better than No. 5.

From conclusions based on our study of many injectors with different designs we believe that any improvement over the type of the design of injector No. 1 will not add more than a very few per cent to the injecting power. It should be remembered, also, that these injectors are made with smooth surfaces, and that in practice, when the injector is cast, the unavoidable rough surface may offer an appreciable resistance which might cause slightly less favorable results than have been obtained in the laboratory.

4. THE FLOW CAPACITY OF AN INJECTING TUBE

The determination of the flow capacity of an injector seemed at first to be a difficult problem, but it was greatly simplified as soon as it had been determined that for any one burner and injecting tube there was a definite relation between the different variables, and that the best injector for any one burner with any one specific gravity of gas, orifice pressure, and volume of gas going through the injector was also the best injector for any other specific gravity of gas, orifice pressure, etc.

The problem resolved itself into a determination of the proper relation between the total port area of the burner and the area of the injector throat. Injectors of definite throat areas and burners with definite port areas were necessary. Three injectors of the following dimensions were made: Each had an angle of approach of 10.5°, an outlet angle of 3.5° with throat diameters of one-half, five-eighths, and three-quarters inch, respectively. This design is shown by injector No. 4, Fig. 13. Since it is essential that the change from the lines of the inlet to those of the outlet should be gradual, the injectors were cut with the throat diameters smaller than desired and then carefully rounded off so that their lines were, as nearly as possible, exactly alike.

Four pipe burners of 0.45, 0.75, 1.05, and 1.35 square inches port area, respectively, were made. The ports were made with a No. 30 drill, arranged in two rows, staggered and spaced five-eighths inch from center to center.

In this study air was again used for the sake of convenience, and an orifice pressure of 4 inches of water was used.

Since each injector has its own distance of the orifice from the throat which produces the maximum pressure in the burner, it was necessary to determine that position for each injector. To be absolutely sure that this distance of the orifice held good for any given injector when the port area had been changed, the tests shown in Fig. 16 were made. Again, to be sure that the maximum pressure obtainable in the burner for any gas pressure bore a linear relation to the gas rate with a change of port area, numerous tests represented by the curves shown in Fig. 16 were made. Invariably when the gas rates were changed the maximum pressures fell on the same straight line for a given injector and a given port area. These maximum pressures were plotted and are shown in

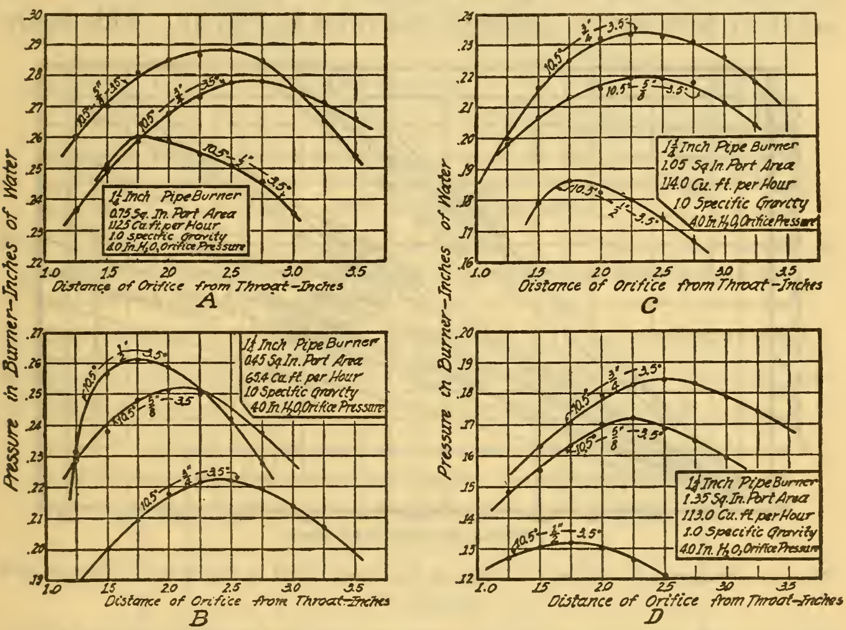


FIG. 16.—*Pressures developed in 1¼-inch pipe burners of different port areas using different injecting tubes with 1.0 specific gravity gas flowing through a sharp edge orifice at 4 inches water pressure*

Fig. 17. The curves of Fig. 17 represent the best injection that could be obtained with the three different injectors and the four burners having different port areas. It should be noted how the pressure in the burner changes with the injecting tubes of different throat diameters for a given burner and also how it varies with the port area of the burner. Of the three pressure lines shown for the burners with different port areas the injector with the throat diameter that gave the greatest pressure was, of course, the best injector. For the 0.45 square inch port area, the one-half-inch diameter throat proved to be the best injector, yet for the other burners it was the poorest. The five-eighths-inch

diameter showed up best with the 0.75 square inch port area burner, while the three-fourths-inch diameter was best for both the 1.05 and 1.35 square inch port area burners.

From the curves of Fig. 17 it is possible to draw the curves in Fig. 18; and from these latter curves we can arrive at the relation between area of the injector throat and port area of the burner. To obtain curves that would show the comparative value of the injectors with different throat diameters, it was necessary to select a constant gas rate and plot the pressure in the burner for the different diameters against the port area of the burner. This has been done for two different gas rates in Fig. 18. This figure

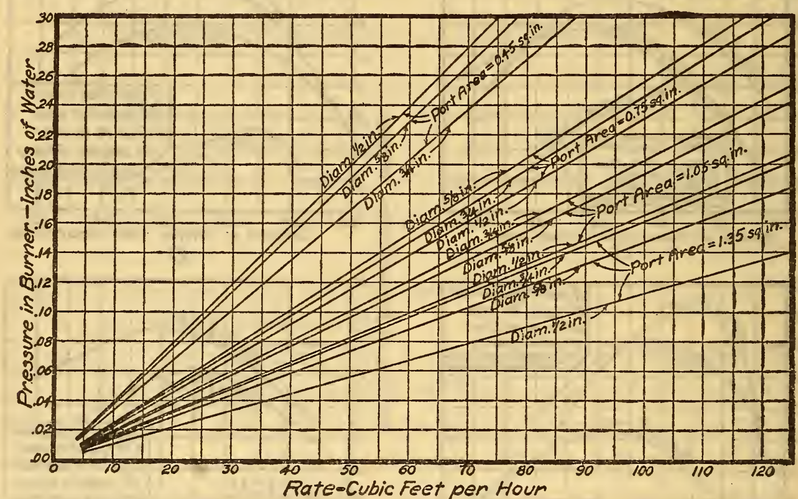


Fig. 17.—*Pressures in burners of different port areas fitted to injectors of different flow capacities*

These curves were determined from the maximum values taken from curves in which those of Fig. 16 are included. Pressures developed by gas flowing through orifice at 4 inches water pressure

indicates at a glance which diameter is the best for the different port areas. It should also be noted that the curves for the two gas rates are exactly alike, which is to be expected. A study of the curves for either gas rate shows, for instance, that the one-half and five-eighths inch throat diameter are equally good injectors for a port area of 0.55 square inch, and that the five-eighths and three-fourths inch throat diameter are equally good at 0.85 square inch port area, because in each case the pressure in the burner would be equal.

It would seem from the intersection of the curves of Fig. 18 that the five-eighths-inch throat would produce a higher pressure for a 0.70 square inch port area than that shown in the figure because the result of the five-eighths-inch diameter for the 0.75 square inch port area is so near the intersection for the three-fourths-inch

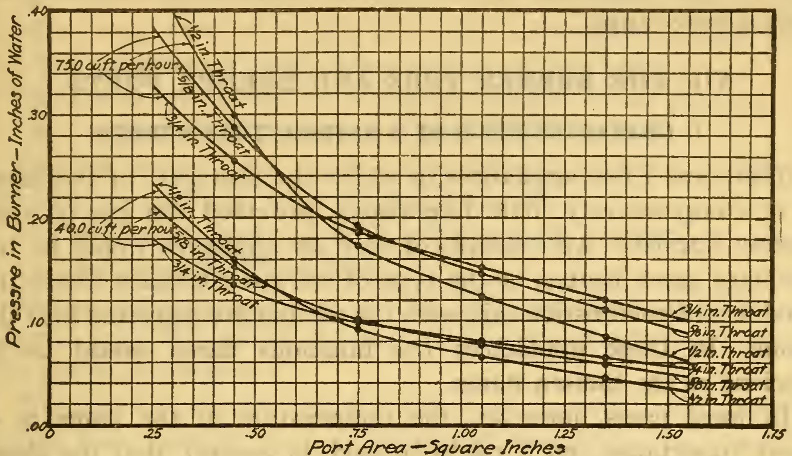


FIG. 18.—*Curves showing pressures developed in burners of different port areas by gas at orifice pressure of 4 inches of water with differently designed injecting tubes (Values taken from Fig. 17)*

throat diameter. If the area of the one-half-inch throat is plotted as the best injector for the 0.45 square inch port area, the area of the five-eighths-inch throat as the best injector for the 0.70

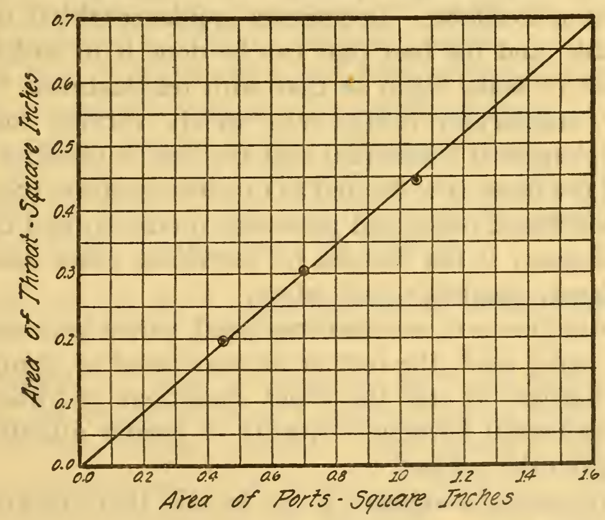


FIG. 19.—*Relation of area of ports to area of throat of injecting tube of good design (Values selected from Fig. 18)*

square inch port area, and the area of the three-fourths-inch throat as the best injector for the 1.05 square inch port area, the points will plot the straight line curve shown in Fig. 19. From this straight line it is observed that the ratio of throat area of injector to port area of burner should be about 0.43. As a result of the study of our numerous data we believe this relation will hold good over a wide range.

VII. THE BURNER TUBE AND BURNER PORTS

1. CHARACTERISTICS OF A SATISFACTORY BURNER

There are a few appliances in which the character of the flame is of comparatively little importance provided the gas is completely burned. An example of such an appliance would be the ordinary space heater, or the type of warm-air furnace where the products of combustion mix with the air and are delivered into the room. In these appliances the luminous flame would be as efficient as the bunsen flame.

In most cases, however, the temperature of the flame is of great importance and it might be said, in general, that the closer the atmospheric burner approaches the performance of the blast burner the higher will be the efficiency obtained and the wider will be the field of application of such burners.

Different processes will require burners of different characteristics, and to get the very best results one should have a burner designed for the particular quality and composition of gas, and for the pressure available. In domestic appliances this is, of course, impracticable, and the best that can be done is to strike a good average and to make them so that with readjustment they will give fairly satisfactory results over widely varying conditions. In such appliances it is essential that the heat is properly distributed; that the flame is so located as to allow complete combustion without objectional odors and poisonous products; and that high thermal efficiency is not secured by sacrificing other items, such as convenience, simplicity, and safety.

In industrial burners, on the other hand, where large quantities of gas are being used, the cost or inconvenience of changing the design of burners to suit the exact conditions is trifling. The design of industrial burners is worthy of greater attention than has been given the subject.

As a very general statement it can be said that burners should have the following characteristics:

(*a*) For a given size they should have a large rate of consumption. Large rate of consumption means reduction in cost of manufacture, and permits concentration of heat which usually produces greatest efficiency. Increase in velocity of mixture going through the burner means less heating of the burner.

(*b*) The burner should be capable of operating with a high air-gas ratio, since an air-gas ratio that approaches a theoretical mixture produces a small flame of high temperature.

(*c*) The flame should be of uniform height in all parts of the burner, so that the distribution of heat will be uniform.

(*d*) The burner must stand a considerable variation in the gas pressure, or gas rate, without giving trouble.

(*e*) The flame must not flash back into the burner.

(*f*) The flame must not blow off.

The relative importance of the different characteristics will depend upon the nature of the process and is somewhat a matter of individual judgment.

To make the tests described in this paper as practical as possible it was first thought best to determine the results that could be obtained with burners just as they were supplied by the manufacturer, and then study the effect of improvements on those burners. We soon observed that unless the gas stream was directed exactly into the center of the mixing tube, the results were uncertain and variable. The following precautions are necessary:

1. The shoulder on the air mixer, as well as the end of the burner, must be machined to obtain correct alignment.

2. The threads of the orifice and those of the air mixer holding the orifice must be true to give proper alignment.

3. The orifice must be drilled with extreme care.

With the more constricted throat formed by the venturi tube, the proper alignment of the orifice is absolutely essential if it is desired to obtain the maximum air injection.

2. LIMITING VALUES FOR VELOCITY OF EFFLUX OF DIFFERENT AIR-GAS MIXTURES

The velocity of flame propagation increases rapidly with increase in the air-gas ratio, but decreases again before the theoretical mixture has been reached. The increase in speed of combustion is accompanied by a decrease in the size of the flame and an increase in the temperature of the flame. Fig. 20 shows roughly the limits, for water gas of about 590 Btu, within which one must operate ordinary pipe burners to secure satisfactory conditions.

As shown by the curves, there are limiting values for the maximum and the minimum rates of gas consumption, and these limits vary with the air-gas ratio. For instance, with a 1.35 square inch port area burner, when the velocity of the mixture through the ports exceeds 15 feet per second with a 3 to 1 ratio, the flames are blown from the ports; when the velocity is about 3.5 feet per second the mixture flashes back into the burner. With a higher air-gas ratio the limits of flash back and blow from ports become closer together. It follows, then, that the port area must not be

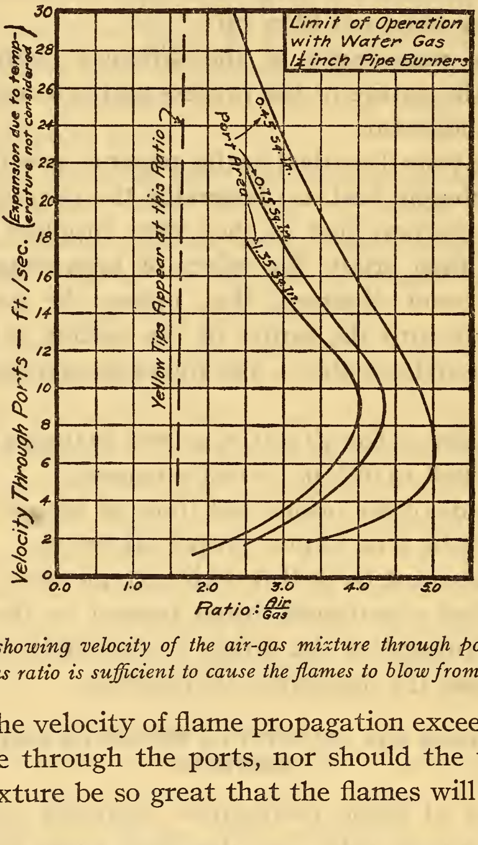


FIG. 20.—*Curves showing velocity of the air-gas mixture through ports of burners when the air-gas ratio is sufficient to cause the flames to blow from the ports*

so large that the velocity of flame propagation exceeds the velocity of the mixture through the ports, nor should the velocity of the outflowing mixture be so great that the flames will blow from the ports.

If a small size port is used it is possible to reduce the velocity of outflowing mixture to a much lower rate than with the larger size ports before a flash back occurs. This is perhaps due to the greater cooling effect of the secondary air. With special burners having small ports, the velocity of outflowing mixture may be reduced very low without a flash back.

3. WHY IT IS EASIER TO CHANGE FROM WATER GAS TO COAL GAS THAN FROM COAL GAS TO WATER GAS

Some discussion of this matter of changing from one gas to another occurred at one of the meetings during the 1919 convention of the American Gas Association. A statement was made that by making water gas of very high heating value it had been found possible to change from coal gas to water gas without adjustment of appliances. Several others related experiences when changes were made from gas of one composition to gas of another composition. No one explained under what conditions these changes were possible. Because it appears that the reason is not clear to everyone and as it is a rather important question, some further explanation may be worth while, although a careful con sideration of the preceding sections should have explained the reason.

Most processes, from the ordinary domestic cooking to the large industrial processes, require approximately a certain number of Btu per hour. If the appliances are adjusted to the condition that gives the best results for a light gas, and if a change is made to a heavier gas, such as changing from coal gas to water gas, a less number of cubic feet of gas per hour and a less number of Btu per hour are delivered if the heating values of the two gases are the same. Even though it may be possible to operate the appliance without readjustments the value of the service is decreased and dissatisfaction results. Changing from the lighter to the heavier gas will also increase the air entrainment, as described in previous sections, and the velocity of flame propagation will be greater. The result will be that in most cases the velocity of efflux through the ports will not be sufficient to prevent the mixture from flashing back.

In the case cited above, where the water gas was made of extremely high heating value, the increase in Btu per cubic foot offset, to some extent, the decrease in volume of gas entering the burner, and the increase in illuminants decreased the velocity of burning sufficiently to prevent the flash back. If, when operating with coal gas, the appliances were adjusted for a very soft flame—that is, a low air-gas ratio—less difficulty would be experienced on changing over to water gas.

In changing from the heavier to the lighter gas less difficulty would ordinarily be experienced. With the same pressure the orifice will pass a greater volume of gas to the burner, and the consumer will merely turn off the excess and rarely have any great trouble, unless the air injection has been so reduced at the same time that the combustion is not complete and the gas burns with a luminous flame which may blacken the utensils.

4. RELATION BETWEEN THE TOTAL PORT AREA AND THE RATE OF CONSUMPTION OF BURNERS

(a) BURNERS WITHOUT INJECTING TUBES

With pipe burners as ordinarily constructed without injecting tubes, the necessity of keeping a correct ratio between the port area and the cross-sectional area of the pipe becomes extremely important when it is desired to secure the maximum rate of consumption from the burner and a good injection of primary air. This is well illustrated in the following table, in which the values are taken from Figs. 21, 22, and 23, for burners operated "cold" with gas of 1.0 specific gravity:

TABLE 4.—Rate of Consumption of Pipe Burners Without Injecting Tube

[Air-gas ratio, 3 to 1; specific gravity of gas, 1.0; burner operated "cold"]

Total port area in square inches	Ratio of port area to area of pipe	Cubic feet per hour at 4.0 inches water pressure	Cubic feet per hour per square inch of port area
0.45	0.3	24	53
.75	.5	57	76
1.05	.7	84	80
[Rates of consumption for a 0.65 specific gravity gas calculated from the foregoing values]			
0.45	0.3	21.3	47.3
.75	.5	50.6	67.4
1.05	.7	74.0	70.5

From the above table it is seen that, for a burner without an injector, the rate of consumption per square inch of port area increases until the port area is about the cross-sectional area of the pipe. These values are shown later in Table 7, and give the relative rates of consumption of burners with and without injecting tubes. See also Fig. 28 in which all of these values are plotted.

Due to the heating of the burner and the expansion of the mixture within the burner, the rate of consumption is reduced somewhat upon the burner being lighted, and is further reduced when

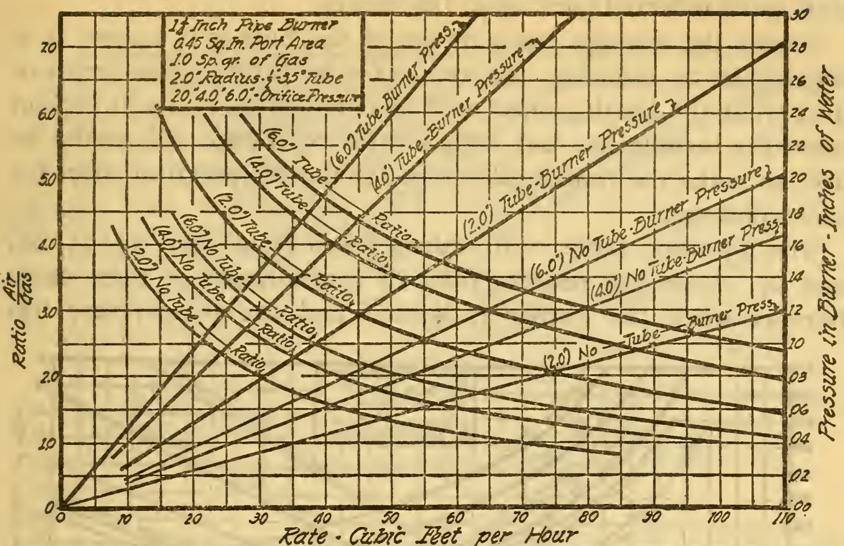


FIG. 21.—*Air injection secured with 1.0 specific gravity gas at orifice pressures of 2, 4, and 6 inches of water in a 0.45 square inch port area burner with and without improved injecting tube*

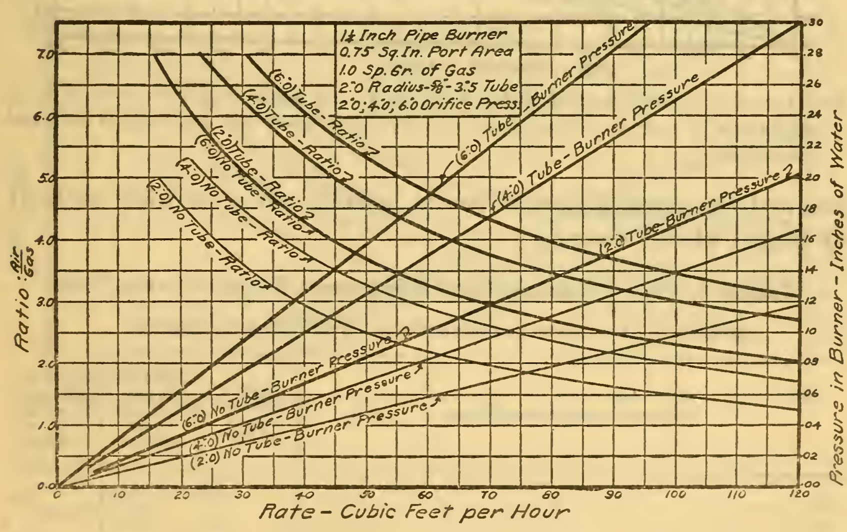


FIG. 22.—*Air injection secured with 1.0 specific gravity gas at orifice pressures of 2, 4, and 6 inches of water in a 0.75 square inch port area burner with and without improved injecting tube*

an object is placed over the burner which will cause some of the heat to be reflected back upon the burner.

Unless the average temperature of the burner is known it is impossible to calculate the reduction in rate of consumption. Just what the heating effect will be is indefinite and will depend upon the installation, but with a little experience one should be able to make the proper allowance for the temperature effect for each condition.

The following table, with values taken from Figs. 24, 25, 26, and 27, shows the relative rates of consumption of these same burners when the burner is lighted and the temperature has

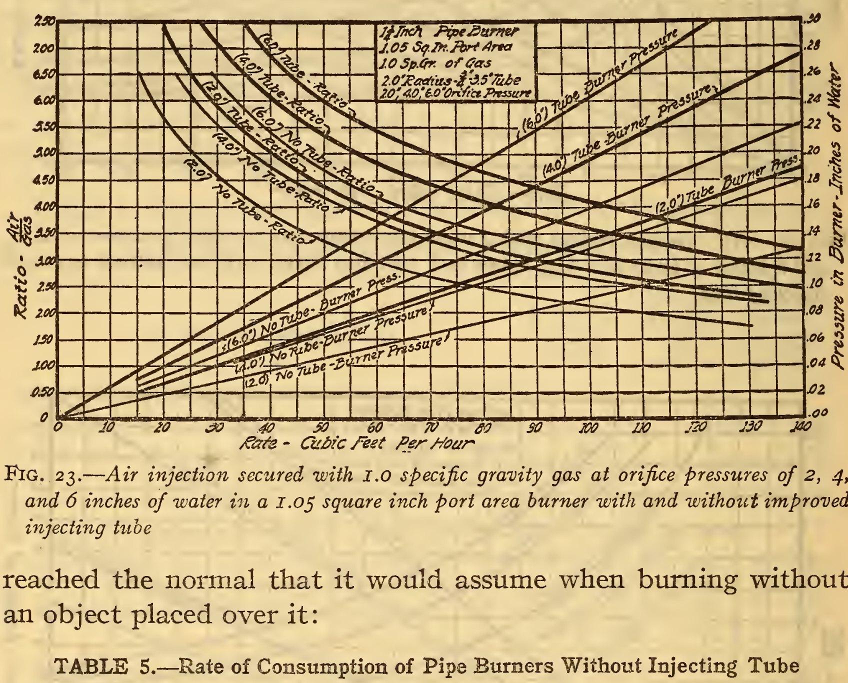


FIG. 23.—*Air injection secured with 1.0 specific gravity gas at orifice pressures of 2, 4, and 6 inches of water in a 1.05 square inch port area burner with and without improved injecting tube*

reached the normal that it would assume when burning without an object placed over it:

TABLE 5.—Rate of Consumption of Pipe Burners Without Injecting Tube

[Air-gas ratio, 3 to 1; specific gravity of gas, 0.65; burner at normal temperature]

Total port area in square inches	Ratio of port area to area of pipe	Cubic feet per hour at 4.0 inches water pressure	Cubic feet per hour per square inch of port area
0.45	0.3	18.0	40.0
.75	.5	41.5	55.3
1.05	.7	66.5	63.3
1.35	.9	99.0	73.2

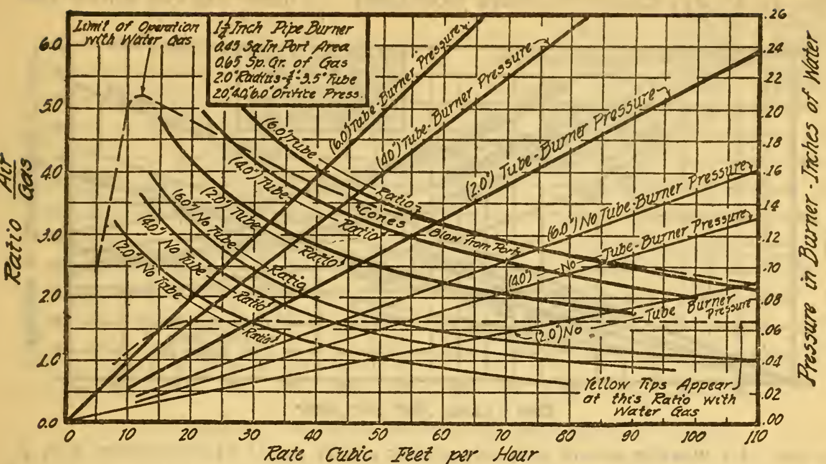


FIG. 24.—*Air injection secured with 0.65 specific gravity gas at orifice perssures of 2, 4, and 6 inches of water in a 0.45 square inch port area burner with and without improved injecting tube*

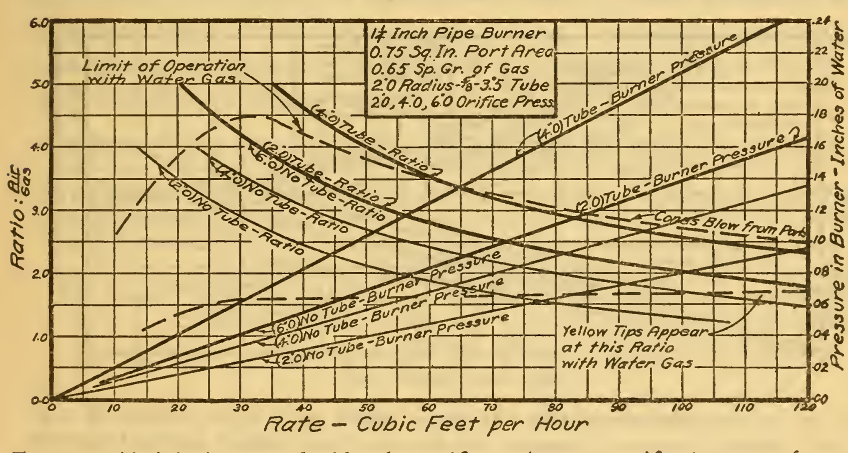


FIG. 25.—*Air injection secured with 0.65 specific gravity gas at orifice pressures of 2, 4, and 6 inches of water in a 0.75 square inch port area burner with and without improved injecting tube*

(b) BURNERS WITH INJECTING TUBES

With injecting tubes of good design the curves as shown in Figs. 21 to 27, inclusive, were obtained. These curves show the

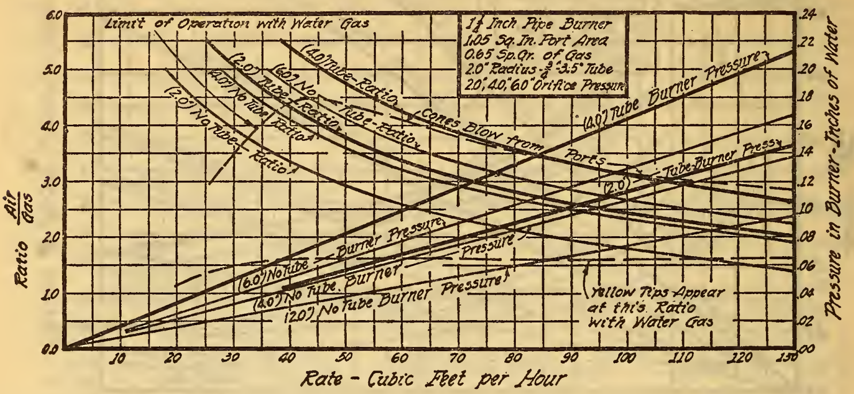


FIG. 26.—*Air injection secured with 0.65 specific gravity gas at orifice pressures of 2, 4, and 6 inches of water in a 1.05 square inch port area burner with and without improved injecting tube*

rates of consumption of burners of 0.45, 0.75, 1.05, and 1.35 square inches port area.

A still more efficient injecting tube, shown by No. 1 in Fig. 13, was designed later. With this design it is possible to secure

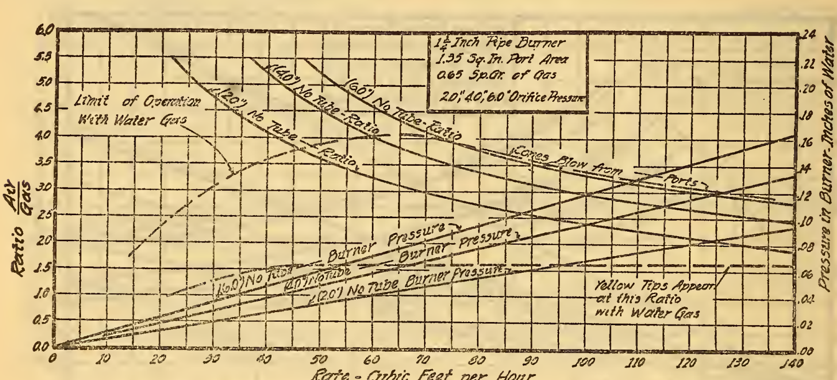


FIG. 27.—*Air injection secured with 0.65 specific gravity gas at orifice pressures of 2, 4, and 6 inches of water in a 1.35 square inch port area burner without improved injecting tube*

slightly higher rates of consumption as indicated from the pressure curves in Fig. 15.

The following table will show the results from the above-mentioned figures for gases of 1.0 and 0.65 specific gravities with burner "cold," and for a 0.65 specific gravity with burner "hot":

TABLE 6.—Rate of Consumption of Pipe Burners With Injecting Tubes

[Air-gas ratio, 3 to 1; specific gravity of gas, 1.0; burners cold]

Total port area in square inches	Ratio of port area to area of pipe	Cubic feet per hour at 4.0 inches water pressure	Cubic feet per hour per square inch of port area
0.45	0.3	65	144.4
.75	.5	101	135.0
1.05	.7	126	120.0
[Values calculated from the foregoing figures, for a gas of 0.65 specific gravity; burners cold]			
0.45	0.3	57.0	126.5
.75	.5	89.0	118.5
1.05	.7	114.0	108.5
[Values for the same burners with gas of 0.65 specific gravity; gas burning]			
0.45	0.3	52.5	116.7
.75	.5	77.5	103.3
1.05	.7	107.5	102.4
1.35	.9	124.0	91.9

Some of the values from the preceding tables have been arranged in the following table, No. 7, to show the relative rates of consumption of burners "cold" and burners "hot," and also to show the percentage of increase in rate of consumption secured with a well-designed injecting tube.

TABLE 7.—Rate of Consumption of Pipe Burners With and Without Injecting Tubes

[Air-gas ratio, 3 to 1; specific gravity of gas, 0.65; values taken from figs. 21 to 27, inclusive; burners cold

Total port area in square inches	Ratio of port area to area of pipe	Cubic feet per hour		Cubic feet per hour per square inch of port area		Per cent increase in capacity
		No injector	Injector	No injector	Injector	
0.45	0.3	21.3	57.0	47.3	126.5	167
.75	.5	50.6	89.0	67.4	118.5	76
1.05	.7	74.0	114.0	70.5	108.5	54
[Rate of consumption with burners lighted]						
0.45	0.3	18.0	52.5	40.0	116.7	192
.75	.5	41.5	77.5	55.3	103.3	87
1.05	.7	66.5	107.5	63.3	102.4	62
1.35	.9	99.0	124.0	73.2	91.9	25

The values from the last table have been plotted in the form of curves in Fig. 28.

These tables and curves show for the "no injecting tube" study that the rate of consumption per square inch of port area increases as the area of ports is made nearer the cross-sectional area of the pipe. They also show that there is a drop in rate of consumption when the burner is lighted.

With the injecting tubes curves were obtained which require some explanation. In this case 1¼-inch pipe burners of different

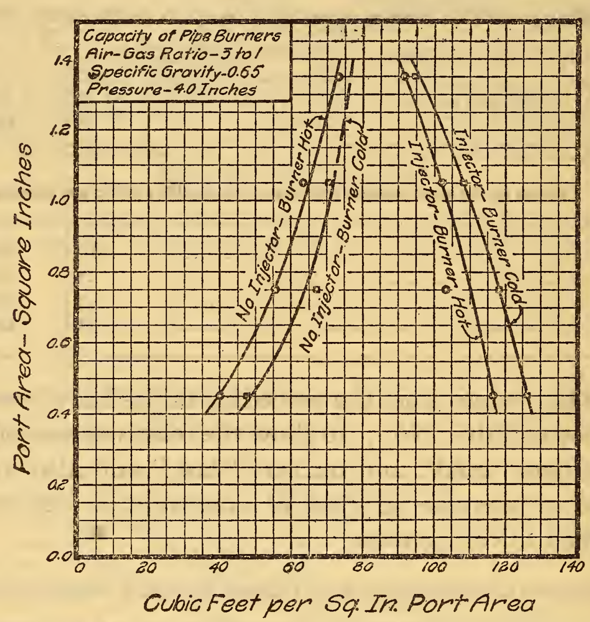


FIG. 28.—*Maximum rate of consumption of 1¼-inch pipe burners with and without improved injecting tubes when using 0.65 specific gravity gas at an air-gas ratio of 3 to 1*

port areas were used. With each change in port area an injector was used that had an area of throat which our previous experiments had shown to be the best for that particular port area. The inlet and outlet angles of all the injectors were the same, but, since the injector with the smallest throat was the longest one, and the cross-sectional area of the pipe was largest, relative to the area of the throat, it allowed the velocity of the mixture to be reduced more gradually and the average static pressure in the burner was greater.

5. EFFECT OF LENGTH OF BURNER ON RATE OF CONSUMPTION

The 0.45 square inch port area burner was less than half as long as the 1.05 square inch port area burner, and it was at first thought that the greater rate of consumption per unit port area secured with the throat was due to increased friction loss with the longer burner. A 1.05 square inch port area burner was made with a double row of ports, and one twice as long with a single row of ports, and it was found that under the same conditions the difference in pressure in the burner was about 1 per cent, thus proving rather conclusively what has already been said in our discussion of Table 7, about the effectiveness of the small throat in a large pipe.

Where the burner is unusually long, or has cross arms, or sharp bends, it is difficult to calculate the friction loss and reduction in rate of consumption, but some allowance will have to be made, depending upon circumstances.

6. EFFECT OF PORT SIZE ON RATE OF CONSUMPTION

With burners having the same port area, but with the size of ports varying from No. 27 to No. 40 drill, no appreciable difference in rate of consumption was found; but with the smaller size port it was possible to turn the gas lower without causing a flash back. This is perhaps due to the secondary air having a greater cooling effect on the small flames, thus reducing the velocity of flame propagation.

7. PRESSURES IN PIPE BURNERS—FIRST AND LAST PORTS

In pipe burners all of the mixture must pass the first port. The velocity of the mixture, therefore, is greatest at this point, while at the last port the velocity is nil. At the first port there are both velocity and static pressures, but at the last port there is, of course, no velocity pressure. The static pressure is the maximum, then, at the last port. The volume of the mixture which issues from a port is dependent upon the static pressure at that port. If the ratio of the port area to the cross-sectional area of the pipe is large, there is a wide difference in static pressures and, consequently, in the volumes of the mixture which issues from the first and last ports. Table 8 contrasts the results of tests made with two ratios of port area to cross-sectional area of pipe.

TABLE 8.—Static Pressures in Pipe Burners—First, Middle, and Last Ports

[Percentage on basis of pressure at last port]

Port area in square inches	Ratio of port area to cross-sectional area	First port	Middle port	Last port
		Per cent	Per cent	Per cent
0.75	0.5	90.0	97.5	100.0
1.35	.9	77.0	92.5	100.0

8. CONE HEIGHTS IN PIPE BURNERS—FIRST AND LAST PORTS

For a given air-gas ratio there is a direct relation between cone height and flame height, pressure in the burner, and the size of the port. The tests shown in the following table indicate the manner in which the heat is distributed along pipe burners.

TABLE 9.—Cone Height Study With Water Gas

[1½ inch pipe burner; specific gravity, 0.65; port area, 1.35 square inches; size of ports, No. 30 drill; air-gas ratio, 3 to 1]

Orifice pressure, inches of water	Gas rate, cubic feet per hour	First port		Last port	
		Velocity of mixture in burner, feet per second	Cone height in inches	Velocity of mixture in burner, feet per second	Cone height in inches
2	70.5	7.52	0.55	0	0.61
4	100.0	10.70	.63	0	.80
6	122.5	13.10	.67	0	.95

For the particular conditions stated in Table 9 it is seen that the cone height at the first port is: At 2 inches pressure, 90 per cent; at 4 inches pressure, 79 per cent; and at 6 inches pressure, 70 per cent of the cone height at the last port. This difference in cone height is caused by the difference in static pressure in the burner between the first and last ports.

For this condition the ratio of the port area to the cross-sectional area of the pipe was 0.90. At the same orifice pressures, with an air-gas ratio of 3 to 1, any pipe burner with the same ratio of port area to cross-sectional area should give the same proportional differences in cone heights. Since the air-gas mixture leaving each port is the same, it follows that, when there is a difference in cone heights, more heat is liberated at the last port than at the first. It is evident that, if an even distribution of heat is desired, the burner must be so designed that the difference in velocity of mixture between the first and last ports is not too great.

9. HOW TO CHOOSE A BURNER FOR ANY CONDITION

Much of the foregoing information on the effect of port area and burner size is valuable and is essential for a real understanding of the problem, but it will not serve the everyday requirements of the industrial fuel engineer and appliance man unless the information can be tabulated clearly so that he can select quickly and accurately the proper burner for any given condition.

In view of this fact there has been compiled a series of tables, based on the preceding experimental work and calculations, which show the rate of consumption of various sizes of burners for different pressures and air-gas ratios.

These tables are adapted only to the conditions specified, but if these tables or further modifications of them are found to work in everyday practice, it will be a simple matter to work up such tables for any other given condition.

Since the rates of consumption of burners without injecting tubes increase with increasing port area, it will be difficult to make up tables that will be generally applicable for the various types of burners.

Many of the industrial pipe burners are made at present with a port area within 5 to 10 per cent of the cross-sectional area of the pipe, and this works well in practice for the usual low rate of consumption secured without injecting tubes, and where it is not necessary to have a very uniform distribution of heat. A port area of 20 per cent, or even 40 per cent, less than the cross-sectional area of the pipe might prove to be more satisfactory in many installations.

With an injecting tube the rate of consumption of a burner is increased so much that it is necessary to have the pipe larger in proportion to the port area, otherwise the velocity past the first ports is too great to give good results.

From the curves in Fig. 28, we have taken the value corresponding to the 1.2 square inch port area as a basis for our tables. The value is 70 cubic feet per square inch of port area and may be used with but small error for burners from 0.7 to 0.9 ratio of port area to cross-sectional area. No tables are shown for burners with injecting tubes because the injectors will vary considerably in practice, and it will be necessary to make up the tables after testing the injector.

In designing an installation the first thing to consider is the volume of gas that will be required for the particular operation. The next point to consider is the nature of the operation, since if the appliance is a drying oven or any other installation requiring hot air, it will be sufficient if the gas is completely burned, in which case the characteristics of the flame are not so important. If, on the other hand, a hot flame is required, and especially where a large quantity of heat is required in a small space, it is absolutely essential that much of the air required for combustion is drawn in as primary air. Increasing the primary air, however, makes it necessary to have a more definite relation between the gas rate and the port area, for, as shown in Fig. 20, a small change in rate will either cause the burner to flash back or the flame to blow from ports.

Having decided on the gas rate and the character of the flame that is required, it will be necessary to know what gas pressure is available during the periods of minimum pressure.

Knowing the gas pressure, the volume of gas required, and the flame characteristics required, it is possible to select from such tables as the following the correct burner for any condition:

TABLE 10.—Rate of Consumption[a] of Pipe Burners Without Injecting Tubes

[Specific gravity of gas, 0.65; area of ports, 0.8 of the cross-sectional area of the pipe]

Nominal size of pipe in inches	Air-gas ratio	Rate of consumption in cubic feet per hour at gas pressures of—				
		1 inch	2 inches	3 inches	4 inches	5 inches
3/4	2.5 to 1	20	28	34	40	44
1	2.5 to 1	32	45	55	64	71
1 1/4	2.5 to 1	55	78	96	111	124
1 1/2	2.5 to 1	76	107	131	151	169
2	2.5 to 1	124	176	216	249	278
3/4	3 to 1	15	21	26	30	33
1	3 to 1	24	34	42	48	54
1 1/4	3 to 1	42	59	73	84	94
1 1/2	3 to 1	57	81	99	114	128
2	3 to 1	94	133	163	188	210

[a] Calculated on the basis that the rate of consumption per square inch of port area is 70 cubic feet per hour at 3 to 1 ratio.

Where it is desired to increase or decrease the ratio of the port area to the cross-sectional area of the pipe, it is necessary to increase or decrease the values in the same ratio as the port area is changed. This applies for ratios from 0.7 to 0.9. For other changes in the ratio see curves giving rate of consumption per square inch of port area for different burners.

TABLE 11.—Rate of Consumption [a] of Pipe Burners Without Injecting Tube

[Specific gravity of gas, 0.35; area of ports, 0.8 of the cross-sectional area of the pipe]

Nominal size of pipe in inches	Air-gas ratio	Rate of consumption in cubic feet per hour at gas pressures of—				
		1 inch	2 inches	3 inches	4 inches	5 inches
3/4	2.5 to 1	16	23	28	32	36
1	2.5 to 1	25	36	44	51	57
1 1/4	2.5 to 1	45	64	78	90	101
1 1/2	2.5 to 1	61	86	106	122	137
2	2.5 to 1	101	143	175	202	226
3/4	3 to 1	12	17	21	24	27
1	3 to 1	19	27	33	39	43
1 1/4	3 to 1	33	47	58	67	75
1 1/2	3 to 1	46	65	78	91	102
2	3 to 1	75	106	130	150	168

[a] Calculated on the basis that the rate of consumption per square inch of port area is 56 cubic feet per hour at 3 to 1 ratio.

Where it is desired to increase or decrease the ratio of the port area to the cross-sectional area of the pipe, it is necessary to increase or decrease the values in the same ratio as the port area is changed. This applies for ratios from 0.7 to 0.9. For other changes in the ratio see curves giving rate of consumption per square inch of port area for different burners.

VIII. THE AIR SHUTTER

Domestic appliances are generally operated at different pressures in different localities and must be so designed by the manufacturer that they will give satisfactory service even with extremely low pressures. Such burners, when operated with medium and high pressures, inject too large a volume of primary air and an adjustment of the air shutter is necessary.

Most municipal ordinances require that the minimum gas pressure shall not be less than 2 inches, and it would seem that in designing a burner for the average condition pressures lower than 2 inches should not be considered. If this is conceded, it can be said that good design in a domestic range burner demands that, when the burner is operated with artificial gas at its maximum rate of consumption—from 15 to 18 cubic feet per hour at 2 inches pressure—the burner should operate satisfactorily and give good flame characteristics with the air shutter wide open. If this is not possible, the port area is not correct for the volume of gas, and what has been gained in injecting power through the use of good orifices, injecting tubes, etc., has been lost by restricting the free flow of air into the injector. This is equally true for industrial appliances, but as these are usually designed more nearly for the existing conditions, it is not necessary to make so much allowance in design for varying conditions as in the case of domestic burners.

1. AREA OF AIR-SHUTTER OPENING REQUIRED

The maximum rate at which a burner is to be operated must be known. It is also necessary to know what the minimum size of air-shutter opening should be in order to get the required volume of air into the burner. To form some opinion of the area of air-shutter opening required, a burner was operated with the air shutter in a position where it offered no resistance to the flow of air, and the total volume of air injected was determined. By

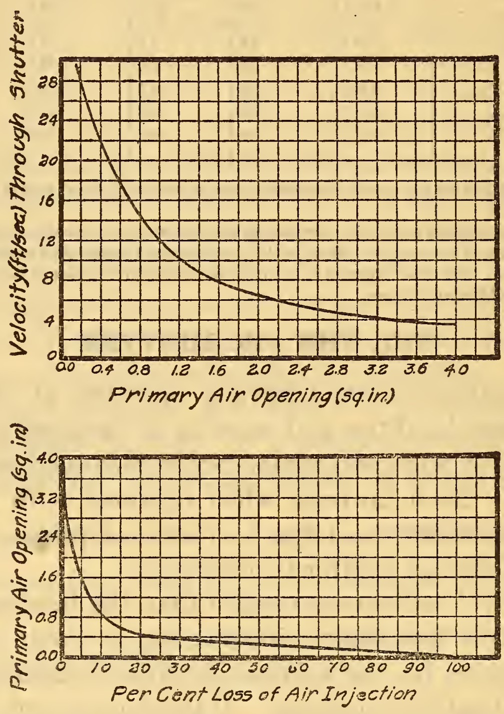


FIG. 29.—*Velocity of air through shutter with different areas of shutter opening and relation of air injection to velocity of air through shutter*

gradually closing the air inlet and observing the effect of the air injection the values shown in Table 12 were obtained.

These values have been plotted in the curves of Fig. 29, and show the loss of injecting power with the decrease of the area of air inlet for the particular conditions stated in Table 12. No attempt to draw any definite conclusions from these curves is made, since there are a number of things to be taken into consideration. The air injection will vary with the momentum of the gas stream, the size of the burner, the design of the injector, and the area of the air-shutter opening. In general, one might say that to keep the loss of air injection down to 1 or 2 per cent, it is necessary to have the shutter opening large enough that the velocity of the air through the opening does not exceed 4 or 5 feet per second.

TABLE 12.—Reduction in the Volume of Air Injected When Closing the Air Shutter

Gas rate, 77.0 cubic feet per hour; specific gravity of gas, 1.0; gas pressure, 4 inches; burner, 1.25-inch diameter, with 1.05 square inch port area, with injector]

Total air injected (cubic feet)	Area of air inlet(square inches)	Velocity through air inlet (feet per second)
318	4.0	3.18
317	3.2	3.91
313	2.0	6.26
305	1.4	8.72
285	.8	14.25
260	.6	17.35
210	.4	19.13
138	.2	27.56

IX. SUMMARY

The investigation of the design of the atmospheric burner has resulted in the development of improved methods for studying burner operation. With the apparatus that has been developed one can determine quickly and accurately, for any burner, under any condition of operation, the volume of air injected. The limits of operation for any burner with any quality of gas can be determined. From these limits one can draw fairly definite conclusions regarding the design of burners for any predetermined condition of operation. In order to thoroughly investigate burner design it was found necessary to study the burner orifice and the theory of flow of gas through different types of orifices, the principles governing air injection, the injecting tube, the burner and burner ports, and the air shutter.

1. THE ORIFICE

The orifices have been divided into two types—the channel type and the sharp-edge type. In Fig. 6 orifices Nos. 1, 2, 3, and 4 are designated as sharp-edge type and No. 5 as channel type.

(*a*) SHARP-EDGE ORIFICE—DISCHARGE COEFFICIENT

The discharge coefficient of a sharp-edge orifice with a given angle of approach is a constant for ordinary sizes of gas orifices and over the usual range of gas pressures.

When the discharge coefficient (K) was determined for sharp-edge orifices with different angles of approach it was found that it varies from about 0.605 for a 90° approach to about 0.875 for an 8° approach. The results have been plotted in Fig. 9*b*.

(*b*) SHARP-EDGE ORIFICE—LOSS OF AIR INJECTION

With the sharp-edge types of orifices the loss of air injection was found to be exactly the same for all designs. The results from which the conclusion is drawn are plotted in Fig. 10.

(*c*) CHANNEL ORIFICE—DISCHARGE COEFFICIENT

It was found that with the orifices of the channel type the coefficient will vary not only with a change in the angle of approach but also with a change in the length of channel. In Table 1 and Fig. 7 are shown the results obtained with orifices of different lengths of channel in which the angle of approach was a constant of 45°.

(*d*) CHANNEL ORIFICE—LOSS OF AIR INJECTION

Orifices of the channel type invariably produced a pressure in the burner which was less than that produced by the sharp-edge type when the gas rate of the two types was the same and was produced by the same orifice pressure. The air injection was, therefore, less with the channel type. Fig. 11*b* shows the loss of air injection for different lengths of channel in which the angle of approach was 45°.

(*e*) RATE OF FLOW OF ORIFICE

If the rate of flow with an orifice is desired, or the size of orifice is required for any given gas rate, it is necessary to know the gas pressure, the specific gravity of the gas, and the discharge coefficient of the orifice. The following formulas will enable one to calculate readily any one unknown value by substituting in the formula the known values:

$$Q = 1658.5\ KA\sqrt{\frac{H}{d}} \text{ or } A = \frac{Q}{1658.5\ K}\sqrt{\frac{d}{H}}$$

Where Q = rate of flow from orifice in cubic feet per hour;
A = area of orifice in square inches;
K = orifice constant, or discharge coefficient;
H = orifice pressure in inches of water;
d = specific gravity of gas (air = 1.0).

2. PRINCIPLES GOVERNING AIR INJECTION

From the fundamental theory and the relations which have been illustrated by examples it is possible to summarize the most important relations as follows:

1. The ratio between the momentum of the gas stream and the momentum of the stream of the mixture is always the same for a burner of a given design irrespective of orifice pressure, specific gravity of gas, or the volume of the air and gas mixture going through the burner.
2. Where the gas rate is increased by change of pressure, the momenta of the gas streams are directly proportional to the pressures.
3. When the same volumes of gases of different specific gravities issue from different orifices under the same orifice pressure the momenta of the gas streams are proportional to the square roots of the specific gravities of the gases.
4. When the pressure is changed to give the same gas rate for gases of different specific gravity, the air entrainment is proportional to the specific gravities.
5. The pressure at any one point in the burner increases in direct proportion to the increase in the momentum of the stream of the mixture.

If the volume of air injected into a given burner with a gas of a given specific gravity at a given pressure and the gas rate is known, it is possible to calculate from the above-stated relations the volume of air injected with a gas of any other specific gravity, gas rate, or gas pressure.

3. THE INJECTING TUBE

The advantages of having an injector of good design to secure a high injection of primary air has been illustrated by tables and a large number of curves. The general design of an injecting tube that produced the greatest injection of primary air was determined and is shown by injector No. 1, Fig. 13. If it is impracticable to use injectors of this design because of limitations in the size of the burners, it is possible to determine from Table 3 the relative injecting power of other designs.

There is a definite relation between the area of the throat of the injecting tube and the port area of the burner that will give a maximum injection of primary air. It has been shown that the area of the injector throat should be about 43 per cent of the area of the burner ports as shown by Fig. 19.

4. BURNER TUBE AND BURNER PORTS

In the section on the burner tube the characteristics required in a satisfactory burner have been discussed, and we have shown for burners without injecting tubes how the rate of consumption per square inch of port area increases with increase in the port area. The rate of consumption of burners with injecting tubes has been shown in the same way. From the tabulations and curves we have taken the rate of consumption per square inch of port area corresponding to an average burner and have made up Tables 10 and 11 showing the rates of consumption of various sizes of burners for different gas pressures.

5. THE AIR SHUTTER

The velocity of the air through the air opening will depend upon the area of the air opening, the momentum of the gas stream issuing from the orifice, the area of the burner ports, and the design of the injector. The opening in the air shutter must be large enough to allow a free and unrestricted flow of air into the burner. From a few curves that have been shown, it seems that the area of the air opening should be of such size that the velocity of the air through the opening does not exceed 4 or 5 feet per second.

6. CONCLUSION

On account of its simplicity, low cost, and reliability the atmospheric burner is well adapted for domestic and most of the smaller industrial purposes. If it is possible to widen the range within which such burners can be operated efficiently and without adjustments, and design them to meet the needs of any particular purpose, it will make gas fuel much more valuable and will broaden its field of application. With this in view, this Bureau has been conducting experiments upon the efficiency and performance of atmospheric burners, both with natural and artificial gas, and the results will be reported in subsequent papers.

Washington, December 16, 1920.