Fluid-Meter Nozzles

By B. O. BUCKLAND,[1] SCHENECTADY, N. Y.

The development of fluid-meter nozzles for the precise measurement of turbine condensate during performance tests in central stations has been stimulated by the fact that in a number of cases the usual weigh-tank installations have not been made. A group of flow nozzles which has been used in such cases is described in this paper. The results of the calibrations of these nozzles are presented and compared with those of some other experimenters.

The calibration results show that the flow coefficients of all the nozzles of the group can be represented as a single function of Reynolds' number with an accuracy of a little better than ±1/2 per cent in flow coefficient.

The comparison of the calibration results with those on other nozzles shows that the nozzles with a gradual approach curve and an appreciable length of cylindrical throat have coefficients which change less abruptly with changing Reynolds' numbers than nozzles with rapidly curving approach and short parallel throat.

THE problem of measuring the flow of water in pipes has been of particular interest to the General Electric Company in making steam-turbine performance tests. The standard method of making such tests has been by means of weigh tanks installed in the generating stations of power companies. As turbine sizes have increased, the flows have, of course, increased also, sometimes to such an extent that the cost of installing sufficient weighing equipment has been considered too high. In some cases where no weigh tanks were installed the company felt the necessity of some form of equipment for the precise measurement of the flow of condensate to determine the performance of some of their turbines. The flow nozzles developed to satisfy this need are described in this paper, together with their calibration.

Between 1930 and 1932 seven of these nozzles were built, all of which have been used to measure the flow of condensate during turbine-performance tests. They range in size from 1 1/4 in. in diameter, which can be used to measure the condensate from a 10,000-kw turbine, to 5 in. in diameter, which can be used to measure the condensate from a 160,000-kw turbine.

These nozzles have been calibrated in the hydraulics laboratory of the University of Pennsylvania by Prof. W. S. Pardoe. Four of the seven have been calibrated in two sizes of pipe so that there are eleven calibration curves on these seven nozzles.

In addition to these calibrations, which were done with great care, there are available less accurate calibrations in oil and water on three very small brass nozzles, similar in shape to the seven larger ones. These three small nozzles were built for some experiments in which they were used to measure the flow of oil.

The purpose of this paper is to describe these nozzles, to present the results of their calibrations, and to compare the results obtained on them with other data that have been published.

Description of Nozzles

Nine of the group of ten nozzles are shown in Figs. 1 and 2.

The seven nozzles built for turbine-performance tests are made of steel. Five nozzles of the seven are made of stainless steel with a smooth machined and polished inside surface. The other two are of machine steel with a smooth machined inside surface which is cadmium plated and polished. The steel nozzles are shown in section in Figs. 3 and 4. The approach curve to the cylindrical throat is a quarter ellipse machined true by the use of a cam.

The three small nozzles are made of brass and range in diameter from 3/8 to 13/16 in. On the brass nozzles the approach curve to the cylindrical throat is also a quarter ellipse but it is approximated by two radii and it is machined to a template.

The proportions of the nozzles are given in Table 1.

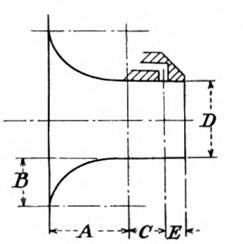


TABLE 1

Nozzle diam., in.	$\frac{A}{D}$	$\frac{B}{D}$	$\frac{C}{D}$	$\frac{E}{D}$
1.253	1.00	0.650	0.500	0.350
1.752	1.00	0.625	0.500	0.250
2.250	1.00	0.638	0.500	0.330
2.999	1.00	0.666	0.583	0.250
4.000	1.00	0.625	0.500	0.250
4.021	1.00	0.625	0.500	0.250
5.016	1.00	0.638	0.500	0.250
0.3757	1.00	0.625	0.354	0.378
0.4995	1.00	0.625	0.425	0.336
0.813	1.00	0.625	0.390	0.350

On the steel nozzles four static-pressure holes for measuring the down-stream pressure are drilled in the throat of the nozzle, a distance of half of a nozzle diameter down stream from the point where the approach curve is tangent to the throat. On one nozzle (1.752 in. diam.) these holes are brought out separately. On the others they are connected to a manifold in the nozzle wall from which a single connection is brought out through the nozzle flange. There are two types of manifold arrangements, illustrated by Figs. 3 and 4.

These static-pressure holes in the throat of the nozzle are made carefully as follows: After the inside surface of the nozzle is machined to its final form and polished, a plug is snugly fitted into the nozzle throat. The static-pressure holes are then drilled through into the plug and reamed. Then the plug is carefully removed and the holes are inspected with a magnifying glass. Generally, they are fairly clean but a wisp of wire edge may be present. If this is the case the spiral reamer is lightly twisted in the hole with the fingers and a hardwood rod is rubbed over the hole where it breaks through into the throat. These two operations are repeated until the wire edge is broken away and the edge of the hole is clean and square.

[1] Turbine Engineering Department, General Electric Company, Schenectady, New York. Jun. A.S.M.E. Mr. Buckland was graduated from University of Colorado in 1923 with the degree of B.S. and was immediately employed by the General Electric Company as a student engineer in its testing department for the next two years. From 1923 to 1926 he was a member of the advanced course in engineering given by the General Electric Company. In 1925 he entered the turbine-engineering department where he has since been engaged in development of improved flow passages and refined methods of flow measurement for test purposes. For the past five years he has supervised for the General Electric Company all water-rate tests of their large steam turbines.

Contributed by the Power Division for presentation at the Annual Meeting, New York, N. Y., December 3 to 7, 1934, of The American Society of Mechanical Engineers.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1935, for publication in a later issue of Transactions.

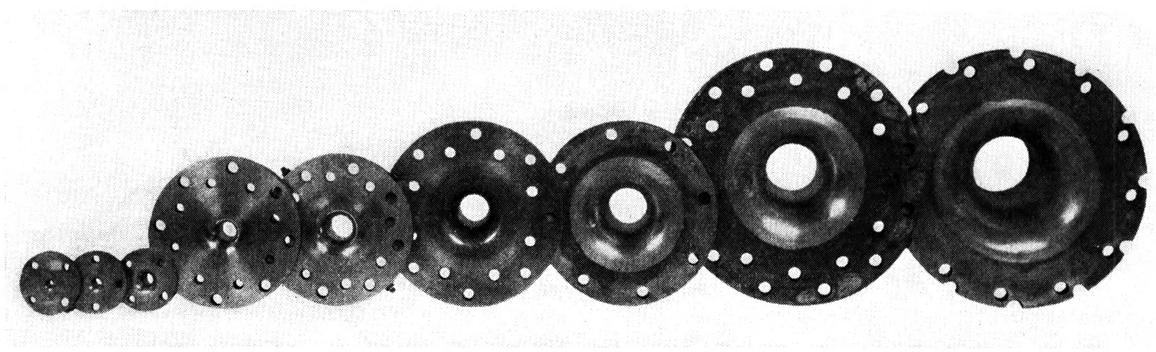


Fig. 1

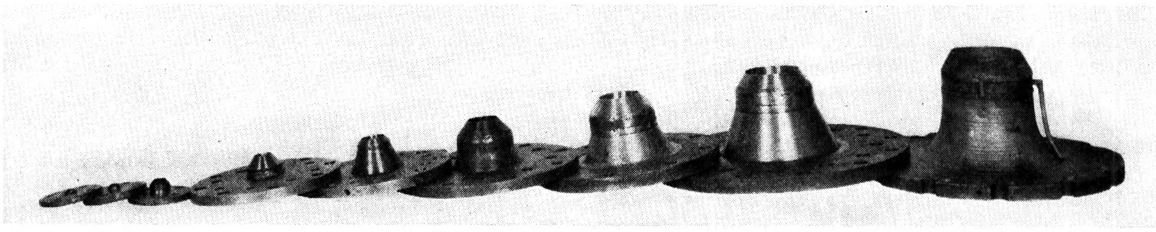


Fig. 2

Figs. 1 and 2 Group of Flow Nozzles With Geometrically Similar Inside Surfaces
(The six large nozzles have been used satisfactorily to measure condensate in turbine-performance tests; the three smaller ones to measure oil flow.)

One static-pressure hole for the upstream pressure measurement is drilled in the pipe wall one pipe diameter ahead of the face of the nozzle flange.

The flanges of the nozzles are made thin enough to allow their insertion into existing station piping simply by springing the pipes apart at a joint.

On the three small nozzles the downstream static pressure is measured by means of but one static-pressure hole placed approximately in the middle of the cylinder throat.

The range of the ratio of the nozzle diameter to the pipe diameter covered by the group is from one-fifth to one-half. No larger ratio of nozzle diameter to pipe diameter than one-half is included because in planning these nozzles it was deemed not advisable to go beyond this ratio in order to insure the accuracy desired in the use of the nozzles. It was felt undesirable to have much more energy in the approach velocity of the stream than the one-sixteenth which results from a ratio of nozzle diameter to pipe diameter of one-half.

The downstream pressure taps were put into the throat of the nozzle for several reasons. First, it was almost necessary in this group of nozzles to have a flange thickness of not much more than 1 in. because they were to be inserted in existing piping. This meant that the external shapes of the nozzles could not be made similar because the 5-in. nozzle would have to have a flange five times as thick as the 1-in. nozzle. The difference in outside form is strikingly shown in Fig. 2. The inside surfaces of the nozzles in Fig. 1 are all similar but the outside surfaces in Fig. 2 are not. It was felt that throat taps would be influenced by this dissimilarity much less than taps in the pipe wall or in the corner. Pipe or corner taps would probably be highly sensitive to this difference in outside form and to the dissimilarity resulting from changes in pipe size. Second, each nozzle before it was used in a water-rate test was to be calibrated. The holes in the throat are an integral part of the nozzle and are calibrated with the nozzle, whereas holes in the pipe would be different for each set-up. Third, the throat holes save making a static-pressure tap in the field.

It was recognized that because the throat is the point of highest velocity the throat holes would have to be carefully made, but it was believed that with care throat holes could be made which would check each other. This belief was borne out in the nozzle in which the taps were brought out separately. In this case the holes check one another within 0.1 per cent or better in pressure difference.

Results of Calibrations

The results of the calibrations are expressed in the form of

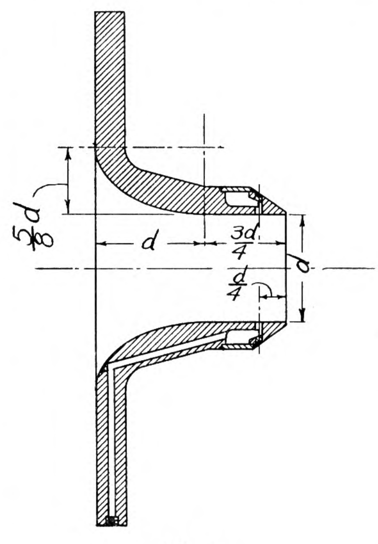


Fig. 3

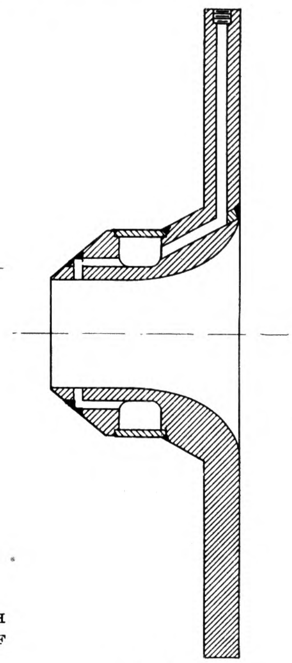
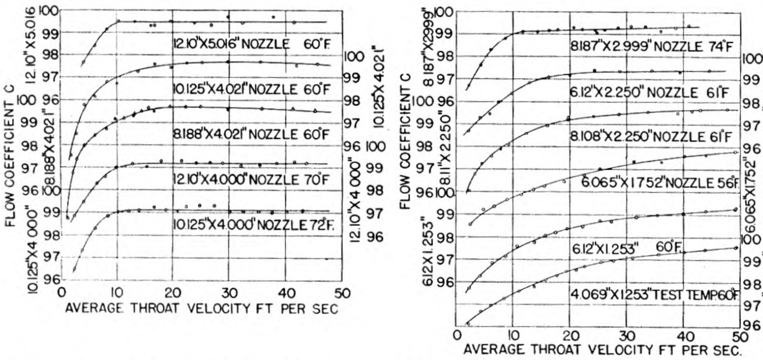


Fig. 4

Figs. 3 and 4 Method by Which Static-Pressure Taps in Throats of General Electric Nozzles Are Connected to Manifolds



FIGS. 5 AND 6 FLOW COEFFICIENTS OBTAINED ON GENERAL ELECTRIC CONDENSATE-MEASURING NOZZLES WITH GEOMETRICALLY SIMILAR INSIDE SURFACES (Data by Prof. W. S. Pardoe, University of Pennsylvania.)

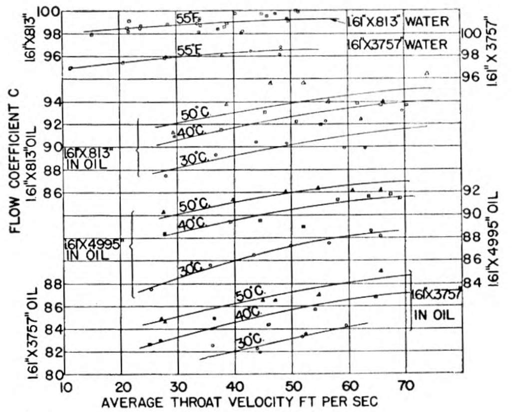


FIG. 7 FLOW COEFFICIENTS OBTAINED ON SMALL BRASS OIL-MEASURING NOZZLES WITH GEOMETRICALLY SIMILAR INSIDE SURFACES

flow coefficients which are defined as the quantity C in the equation

$$Q = C\gamma A_2 \sqrt{\frac{2gh}{1 - R^4}}$$

where

A_2 = nozzle-throat area, sq ft
R = ratio of throat diameter to pipe diameter
γ = fluid density, lb per cu ft
h = pressure differential, feet of flowing fluid
Q = flow, lb per sec
C = nozzle flow coefficient

This definition is used by the A.S.M.E. Special Research Committee on Fluid Meters and it takes account of the approach velocity.

The results of the tests on the carefully calibrated steel nozzles are plotted against average throat velocity in Figs. 5 and 6.

The results of the tests on the small brass nozzles are plotted against average throat velocity in Fig. 7.

The accuracy of the calibrations on the small nozzles is indicated by the scatter of the points. Less weight should be given to these data than to the data on the large steel nozzles because the calibrations on the brass nozzle were less carefully made. The calibrations with water on the small nozzles were made with more care than those with oil.

The results of all of the tests on all of the nozzles are plotted against Reynolds' numbers in Fig. 8. The Reynolds number in this case is defined in terms of the throat diameter, as

$$\frac{Vd\rho}{\eta}$$

where

V = average throat velocity
d = throat diameter
ρ = fluid density
η = fluid viscosity

The points in Fig. 8 fall very nearly on one curve except for the data obtained on the three small nozzles. These nozzles are included because they show the trend of the curve at the lower Reynolds' numbers.

The points in Fig. 8 which represent the aggregate of all the results on the steel nozzles scatter about twice as widely as the

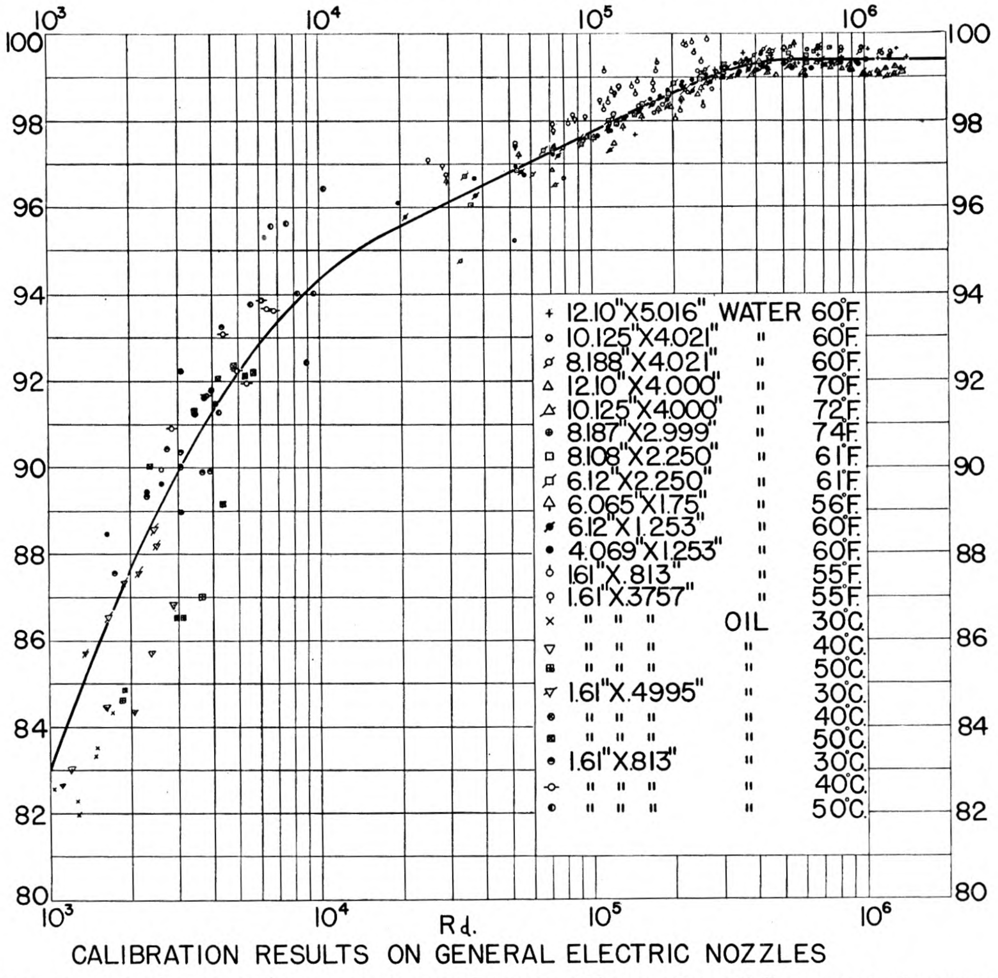


FIG. 8 FLOW COEFFICIENTS PLOTTED AGAINST REYNOLDS' NUMBER FROM 22 CALIBRATION CURVES ON TEN GENERAL ELECTRIC NOZZLES WITH GEOMETRICALLY SIMILAR INSIDE SURFACES

points on any individual steel-nozzle calibration curve in Figs. 5 and 6. This scatter may be due either to a lack of similarity in the nozzles and their calibration arrangements, or to a change in the calibration instruments between one nozzle calibration and another.

As far as similarity is concerned the nozzles are almost exactly similar except for the diameters of the pressure taps in the throat, and even these are approximately similar.

In the calibration arrangements, however, no particular attempt was made to obtain similarity regarding the pipe roughness and the length of the approach pipe. Standard pipe was used for the approach with no change except the filing necessary to obtain a good upstream static-pressure tap in the pipe. The length of the approach pipe used in all cases was approximately 12 ft.

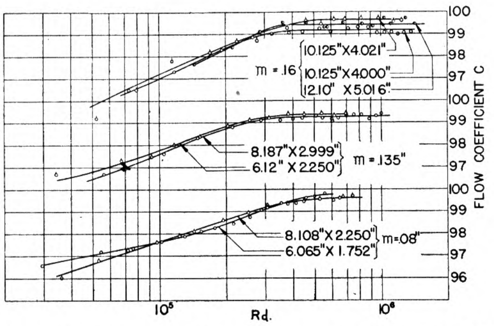


FIG. 9 FLOW COEFFICIENTS PLOTTED AGAINST REYNOLDS' NUMBER FROM SEVEN CALIBRATION CURVES ON SIX GENERAL ELECTRIC NOZZLES WITH GEOMETRICALLY SIMILAR INSIDE SURFACES

(Nozzles grouped according to ratio of nozzle area to pipe area.)

Of course, in plotting all of the points together in Fig. 8 it is assumed that the variation in the ratio of nozzle to pipe diameter, within the range of the tests, has no influence on the results. When there are no whirls in the approach, and the approach-velocity profile is not extremely ragged, it is improbable that the variations in the data, due to the variations of pipe size, would be more than $^1/_5$ per cent in flow coefficient.

This figure of $^1/_5$ per cent is merely an estimate. It is arrived at by the following considerations: With no upstream disturbances the flow approaching a nozzle must have a velocity breast either uniform, fully rounded, or something in between. If the velocity is uniform, the approach-velocity energy is exactly that accounted for in the definition of the flow coefficient. The maximum difference between this amount of energy and the actual velocity energy in the stream occurs when the approach stream has a fully developed and rounded profile. If the approach pipe is very large, this difference is negligible because the velocity energy in the stream is negligible. As the approach pipe becomes smaller, however, the velocity energy in the pipe becomes a larger proportion of the nozzle energy and this discrepancy between the actual and assumed velocity energy would exert a greater influence on the coefficient. Since the actual velocity energy in the pipe is greater than the energy assumed in the definition, the flow coefficients for the smaller pipes might be expected to have the higher values.

The velocity energy of the flow in a pipe with a fully developed turbulent profile is known to be approximately 6 per cent greater than the velocity energy of the same flow with a uniform velocity profile. This means that for a ratio of nozzle to pipe diameter of $^1/_2$, there is approximately $^6/_{16}$ per cent in nozzle energy more actual velocity energy in the approach stream than is accounted for by the flow-coefficient definition. The nozzle coefficient for a diameter ratio of $^1/_2$, therefore, might be expected to be $^1/_5$ per cent higher than that for a zero ratio.

By similar reasoning, it might be concluded also that rough pipes would give higher coefficients than smooth ones.

In Fig. 8 the results of the calibrations in water on the 0.813-in. and on the 0.376-in. brass nozzles are definitely higher than the average of the rest of the data. This difference might conceivably be assigned to pipe roughness or to nozzle-pipe-diameter ratio. However, a comparison of the curves on Fig. 9, on which the data from the more carefully calibrated steel nozzles are plotted in groups having the same nozzle-pipe area-ratio m, shows no definite trend with area ratio. Further, in this group of nozzle coefficients in Fig. 9, two 4-in. nozzles calibrated in 10-in. pipes show results as much as $^1/_2$ per cent difference in coefficient. When these points are considered there does not seem to be sufficient evidence to draw conclusions regarding the variation of coefficient with pipe size or pipe roughness. The author has, therefore, chosen to put a single curve through the data in Fig. 8 and to explain the high coefficients obtained with water on the brass nozzles and the scatter of the data on the steel nozzles as being due chiefly to differences in the nozzles themselves, or differences in calibration instruments.

To establish the influence of pipe size, pipe roughness, or length of approach pipe on the nozzle coefficient, careful tests would have to be made covering a much larger range of ratios of nozzle to pipe diameter than was the case in these tests.

The two 4-in. nozzles in Fig. 9 are alike except for 0.021-in. difference in throat diameter and the difference in inside surfaces. The larger nozzle has a cadmium-plated and polished surface and the smaller has a very smooth machined and polished stainless-steel surface. However, no difference in the polish and smoothness of these two nozzle surfaces can be detected either by sight or touch.

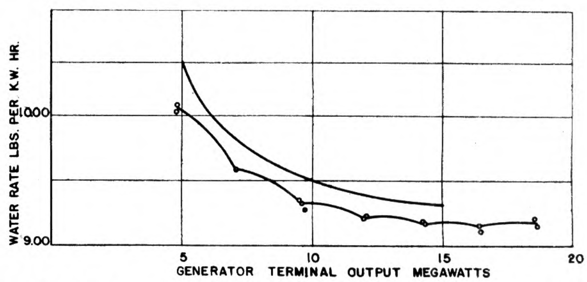


FIG. 10 SAMPLE TURBINE-PERFORMANCE TEST TO ILLUSTRATE TYPE OF DATA OBTAINED USING THE GENERAL ELECTRIC NOZZLES

Considering the data as a whole, none of the condensate-measuring nozzles shows flow more than $^1/_2$ per cent away from the mean curve drawn through the data in Fig. 8. If $\pm ^1/_2$ per cent is acceptable accuracy, the mean curve in Fig. 8 might be used for the entire series instead of the specific calibrations for each nozzle. Of course, the nozzles are not simple. They are made with care, and in their use certain precautions must be taken to realize this accuracy. If the highest accuracy is desired, any nozzle should be calibrated under test conditions which duplicate as nearly as possible the conditions of use.

A fair example of the data obtained measuring the condensate with one of these nozzles is shown in Fig. 10.

COMPARISON WITH OTHER DATA

A comparison of the flow coefficients obtained on the nozzles described with the coefficients obtained on several other types by other experimenters is shown in Fig. 11. The curve from Fig. 8 is labeled "G.E. Nozzle" in Fig. 11. The curves labeled "B. of Std. A_2, B_2, and D_1" are results published by Bean, Buckingham, and Murphy (1)[2] of the Bureau of Standards.

[2] The numbers in parentheses apply to the references at the end of the paper.

The General Electric nozzles and those of the Bureau of Standards are almost exactly the same shape. The approach curve to the cylindrical throat of the latter is a quarter ellipse, one diameter long and two-thirds of a diameter wide. The length of the cylindrical throat on nozzles A_2 and B_2 is one-half the throat diameter and on D_1 it is one throat diameter. The Bureau of Standards nozzles, however, do not have throat taps.

The Bureau of Standards nozzles were calibrated in air while discharging from a 36-in. pipe into the atmosphere. The upstream pressure was taken as the impact pressure in the large pipe and the downstream pressure was taken as that of the atmosphere.

The coefficients of a 5-in. nozzle C_2 of the same proportions as A_2 and B_2 were given in the paper previously mentioned (1), but the results are not shown here because the curve fell almost exactly on the curve B_2. These four Bureau of Standards nozzles range from $1^3/_4$ to 5 in. in diameter and they were calibrated over a range of pressure drop of from 0.25 to more than 26 in. of water although no results are given in the paper above the 26-in. point. The results from the four nozzles cover about the same range of Reynolds' numbers, from 60,000 to 600,000.

The Bureau of Standards coefficients check our curve closely. However, below Reynolds' numbers of 100,000 the curve for nozzle D_1 begins to be higher than ours. This agreement indicates that the pressure measured in the throat of our nozzles would be atmospheric if the nozzles were discharging into the atmosphere, as was the case with Bureau of Standards tests.

The curve marked "M. and J. curve 1" is the result of calibrations made with steam on a 2.2-in. nozzle by Moss and Johnson at the Lynn Works of the General Electric Company (2).

Our nozzles are somewhat the same shape as this Moss and Johnson nozzle. Like our nozzles this M. and J. nozzle No. 1 measures the downstream pressure in the throat but the static

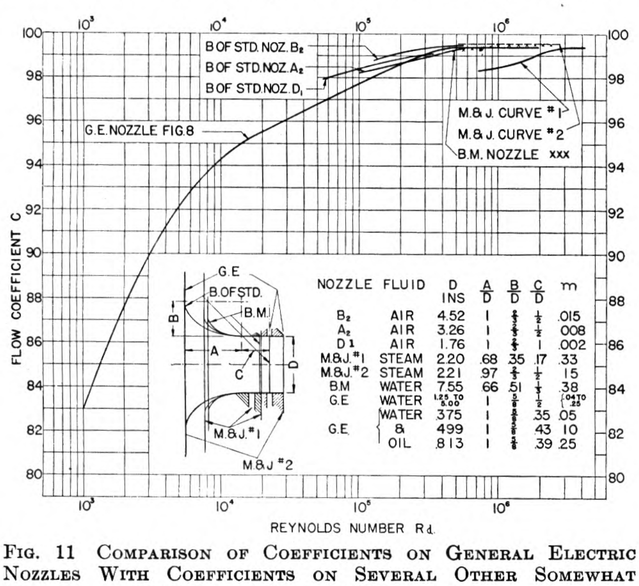


FIG. 11 COMPARISON OF COEFFICIENTS ON GENERAL ELECTRIC NOZZLES WITH COEFFICIENTS ON SEVERAL OTHER SOMEWHAT SIMILARLY SHAPED NOZZLES

hole is much closer to the point where the approach curve is tangent to the throat than is the case with our nozzles. The proportions of this M. and J. nozzle No. 1 are shown in the table in Fig. 11. The dimension C is the distance from the point where the approach curve is tangent to the throat to the point of measurement of the static pressure.

The Moss and Johnson calibrations on this nozzle No. 1 were in 215 lb per sq in. absolute pressure, 75 deg superheated steam with pressure drops ranging from $^1/_4$ to 6 per cent of the initial pressure. These conditions brought the tests into a rather high range of Reynolds' numbers (from 700,000 to 4,000,000). The upper values of the Moss and Johnson coefficients check our curve exactly. The lower values, however, are about 1 per cent lower than our curve. The experience with incompressible flow in nozzles generally has indicated that in the range of Reynolds' numbers covered by the Moss and Johnson tests the nozzle coefficient is independent of the Reynolds

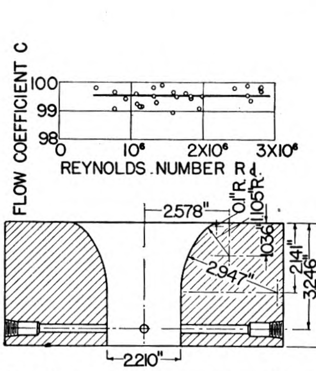


FIG. 12 FLOW COEFFICIENTS OBTAINED FROM CALIBRATIONS IN STEAM BY MOSS AND JOHNSON ON THEIR NOZZLE NO. 2 AND A SKETCH OF THE NOZZLE

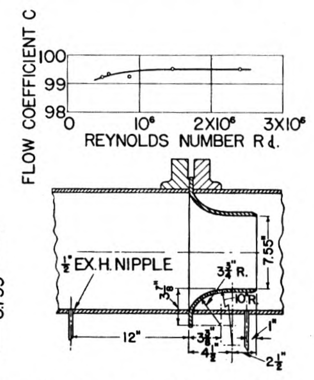


FIG. 13 SKETCH OF BAILEY METER COMPANY 12 BY 7.554 IN. NOZZLE AND FLOW COEFFICIENTS OBTAINED ON IT FROM CALIBRATION IN WATER AT THE OHIO STATE UNIVERSITY

number. The drop in coefficient indicated by the low end of the Moss and Johnson curve is extremely unusual at these high Reynolds' numbers.

The curve labeled "M. and J. curve No. 2" is the result of calibrations in steam by Moss and Johnson on a 5.745 by 2.2086 in. nozzle. The data on this nozzle have not been published before, and, therefore, it seems advisable to show the test points and a sketch of the nozzle. These are shown in Fig. 12.

As described to him by Mr. Johnson to whom the author is indebted for the data, this nozzle was very carefully calibrated using two steam condensers. The second condenser was used to catch the steam carried in the air going to the ejectors.

This Moss and Johnson nozzle No. 2 is almost exactly the same shape as the G.E. nozzles. The approach curve is a quarter ellipse which is approximated by two radii. It is two-thirds of a nozzle diameter wide and 0.97 of a nozzle diameter long. The cylindrical throat is three-quarters of a diameter long and the downstream pressure is taken in the throat at the same point as it is in our nozzles. The only essential differences of this nozzle from our nozzles are its flat exit face and its manner of bringing out the throat-pressure taps.

The Moss and Johnson curve No. 2 agrees very well with the results on the G.E. nozzles. This agreement indicates that the two main differences of our nozzles from this nozzle have very little effect on the flow coefficient.

The curve marked "B.M. nozzle" is the result of a calibration in water of a 12 by 7.554 in. nozzle built by the Bailey Meter Company and calibrated at Ohio State University. For these data and the permission to show them the author is indebted to R. E. Sprenkle, of the Bailey Meter Company. The calibration points and a sketch of the nozzle are shown in Fig. 13.

The B.M. nozzle is approximately the same shape as our nozzles. The approach curve is a quarter ellipse, which is approximated by two radii. It is 0.6 of a nozzle diameter long and 0.5 of a diameter wide. The cylindrical throat is one-third of a diameter long. The downstream pressure, however, is taken in the pipe wall at a point shown in the sketch in Fig. 13.

This B.M. nozzle curve falls directly on top of the M. and J.

curve No. 2, and it checks the curve representing the data on our nozzles very closely. The B.M. nozzle has the largest ratio of nozzle to pipe area of any of the nozzles shown in Fig. 11 (m = 0.394).

Fig. 14 is a comparison of our curve, Fig. 8, and some results on the German standard nozzle of 1930 (Deutsche Normdüse 1930) for which under the name of rounded orifice (abgerundete Drosselscheibe), coefficients were given by Witte in 1928 (4). The three curves marked "D.I.N. 1952" are taken from the German rules of 1932 for flow measurement with standard

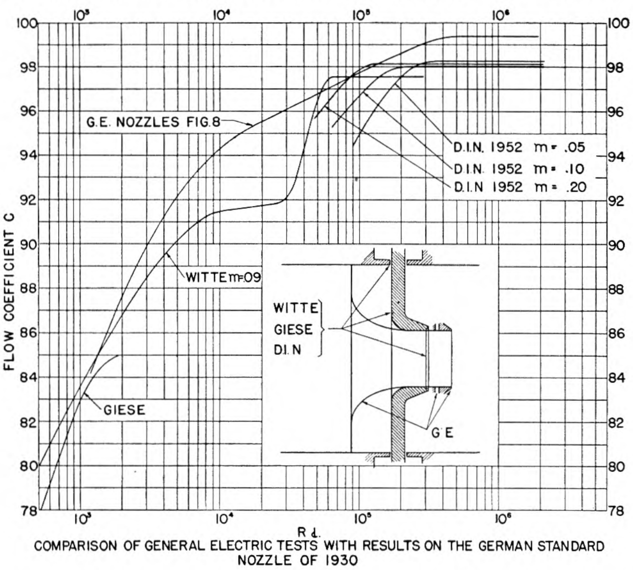


Fig. 14 Comparison of Coefficients on General Electric Nozzles With Coefficients on the German Standard Nozzle (Normdüse)

(The Normdüse coefficients are modified to make the sets of data comparable as regards definition of coefficients and methods of measuring the pressure differential.)

nozzles and orifices. The curve marked "Witte" is the result of calibrations by Witte on a 0.59-in. diameter Normdüse in water, benzine, and four different oils (4). The curve marked "Giese" is the result of calibrations at low Reynolds' numbers in oil on a 0.47-in. diameter Normdüse 1930 (5).

On all the coefficient curves of the Normdüse 1930, in Fig. 14, three corrections have been applied. The first is to take care of the difference in definition of flow coefficient, used by the Germans, from that used in this paper. The second and third are to change the Normdüse coefficients to what they would have been had the upstream pressure been measured one pipe diameter upstream from the nozzle face and the downstream pressure at the point of minimum pressure on the pipe wall on the downstream side of the nozzle. For these two corrections the results of Witte's measurements of pressure difference between the corner pressure taps and points along the pipe wall, both up- and downstream, were used (3) and (4).

The point of minimum pressure on the down-stream pipe wall was used as a basis for comparing these German nozzles with ours because it corresponds more closely than the corner-tap pressure to the throat pressure in our nozzles. The coefficient of the Normdüse based on a downstream pressure tap at the point of minimum pressure on the pipe wall would correspond to the coefficient obtained with Normdüse discharging into an infinite chamber. This is borne out by a comparison of the measurements by E. Stach (6) of the coefficient of the Normdüse, discharging freely into the atmosphere, with the measurement of pressure distribution along the downstream pipe wall by Witte (3) and (4). This difference between the corner-tap pressure and the minimum pressure on the pipe wall is, according to Witte, about one per cent of the nozzle velocity head at an area ratio of 0.09.

The sharp irregularity of the Normdüse coefficients between values of the Reynolds number of 30,000 and 50,000 indicates a rapid and radical change in the character of flow in the nozzle between these points. This sharp irregularity as shown in Fig. 14 is a faithful reproduction of the results given by Witte in 1928 and the V.D.I. Regeln limit the use of the calibrations on the Normdüse to flows above this rather uncertain place. The sharp entrance curve and short cylindrical portion, together with the fact that a maximum value of the coefficient on Fig. 14 is about 0.975, would seem to indicate that the jet separates from the nozzle walls somewhere in its passage through the nozzle.

The original name used by Witte, "rounded orifice," or, literally, "rounded throttle disk," seems to us more appropriate than "nozzle" to describe this measuring device.

References

1 "Discharge Coefficients of Square-Edged Orifices for Measuring the Flow of Air," by H. Bean, E. Buckingham, P. S. Murphy, Bureau of Standards *Research Journal* (Research Paper No. 49, 1929).

2 "A System for the Measurement of Steam With Flow Nozzles for Turbine-Performance Tests," by S. A. Moss and W. W. Johnson, A.S.M.E. Trans., 1933, paper FSP-55-10.

3 "Die Strömung durch Düsen und Blenden," by R. Witte, *Forschung auf dem Gebiete des Ingenieurwesens*, July, 1931, vol. 2, no. 7, pp. 237–272.

4 "Durchfluss Beiwerte für Wasser, Öl, Dampf, und Gas," by R. Witte, *Zeit. V.D.I.*, vol. 72, no. 42, Oct. 20, 1928.

5 "Mengenmessung mit Düsen und Blenden bei Kleinen Reynoldsschen Zahlen," *Forschung auf dem Gebiete des Ingenieurwesens*, January/February, 1933, vol. 4, no. 1, pp. 1–52.

6 "Die Beiwerte von Normdüsen und Normblenden in Einlauf und Auslauf," by E. Stach, *Zeit. V.D.I.*, vol. 78, no. 6, Feb. 10, 1934.