heads for an elbow of this type, and the writer feels that it was owing to the method of test where a nozzle preceded the elbow and a uniform velocity was effective over the whole cross-sectional area. This condition does not exist in practice. Besides the tests were made at very high velocities in a 3-in. elbow. The writer's tests check much more closely with those of Busey[5] which were made at the Buffalo Forge Company, although they are slightly higher.

C. E. Peck.[6] The results found by the authors of the paper under discussion are valuable in that they verify the general opinion that the influence of bends or obstructions at the discharge of the fan has little effect on the fan performance. However, it is important to note, as pointed out by the authors, that their conclusions are based on a fan of the centrifugal type with a volute housing so designed that the air being discharged from the housing has a reasonably definite direction and fairly uniform distribution at the outlet.

The axial-flow type of fan or propeller type of fan discharging directly into a duct system may give to the air quite different types of flow characteristics such that bends, sudden enlargements, or obstructions will affect the fan performance appreciably. For instance, the air leaving a high-capacity propeller-type blower has very large rotational-velocity components which maintain themselves at large distances beyond the fan in the duct work. The rotational velocities vary from the center to the wall of the duct. With these complicated directions of flow existing, any obstruction or bend, or change in duct cross-section at the fan discharge or even at some distance from the fan discharge may appreciably affect the fan delivery and efficiency.

The performance of a propeller fan with properly designed guide vanes at the fan discharge is such that the air leaving the vanes is essentially parallel with the axis of the duct. With this condition and uniform velocity distribution the effect of bends or obstructions beyond the guide vanes would probably be small.

In general, when a fan of any type is applied to a duct system and the fan is provided with a volute case or guide vanes or some device which produces uniform flow in a given direction the effect of duct shape and obstructions beyond the fan is negligible. However, when fans are applied in special cases, such as the cooling of electrical machinery where volutes and guide vanes can be rarely used, the effect of obstructions at the fan outlet is more noticeable. The effect of the close proximity of end windings and the rotor of the electrical machine must be considered and usually such obstructions greatly influence the fan performance.

Authors' Closure

The investigations of Mr. Madison show that the pressure drop in an elbow depends on the uniformity of flow of the air approaching it but do not appear to demonstrate any influence of the elbow on the performance of the fan. If the character, location, or orientation of the elbow is such as to change the discharge pressure at the fan outlet, the condition under which the fan is operating will change to some other point on the fan-performance curve. If the elbow actually affects fan operation, the performance curve of the fan will change. Mr. Madison has apparently investigated a combination of fan and elbow for a fan which gives a spiral discharge flow and shows that the performance of this combination varies with the location and orientation of the elbow. The authors believe that this variation results from actions in the elbow alone and that the fan is unaffected as long as it has a complete fan casing.

In the authors' tests the elbow was not tested at some distance from the fan because the air flow at the fan-casing outlet was found to be practically uniform over the whole cross-section.

The authors agree with Mr. Peck that if the fan casing is absent or incomplete, the effect of obstructions in close proximity to the fan may be considerable.

[5] "Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts," by Frank L. Busey, A.S.H.&V.E. Trans., 1913.

[6] Power engineering department, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa. Jun. A.S.M.E.

Calibration of Rounded-Approach Orifices[1]

R. E. Sprenkle.[2] Mr. Smith's data are timely, particularly since some can be connected directly to other pertinent data which greatly extends the scope of usefulness. For instance, the writer's Fig. 1 shows the data from Mr. Smith's large oil-flow nozzle plotted together with data from a Bailey 3.06-in. × 1.836-in. water-flow nozzle. While at the junction point of the two sets of data, or at a Reynolds number of approximately 35,000, a possible separation by approximately 1/2 per cent exists, there is no question but that the data from the Bailey nozzle are a real continuation of the data of Mr. Smith's nozzle, and that a single smooth curve represents the complete data of the two when plotted against Reynolds' number.

This despite the fact that the Bailey nozzle used pipe-line connections instead of the impact and throat type of connections used by Mr. Smith. Pipe-line taps place the inlet static connection into the pipe wall at a distance of one pipe diameter preceding the nozzle inlet, and the outlet static connection into the pipe line back of the nozzle throat, as shown in Fig. 13 of Mr. Buckland's paper,[3] "Fluid Meter Nozzles."

Further, there is a distinct difference in size between the two nozzles, as well as the use of entirely different flowing fluids in obtaining these calibration data. When to all of these is added the difference in the physical set-up and the fact that the different experimenters involved were working entirely independently of each other, this agreement becomes all the more noteworthy.

Since the water curve obtained from the Bailey nozzle flattens out at about 600,000 Reynolds' number, and continues flat up to the maximum test point of about 900,000 Reynolds' number, there is little reason to doubt the projection of the curve as a perfectly flat line to Reynolds' number of much greater value, possibly to infinity. Such tests, using steam flow, are now scheduled to be made shortly on the Bailey nozzle, better to show the validity of this assumption.

In view of this the curve has been extrapolated to a Reynolds number of over 3,000,000 so as to cover the useful range of steam, air, or other low-viscosity fluids, and it is felt that this same curve could be extrapolated further if desired without being in error more than plus or minus 1/2 per cent. Likewise, the same curve should apply to any flow nozzle of this general structure, provided the diameter ratio of the throat to the inlet-pipe diameter does not exceed 60 per cent.

Mr. Smith's medium- and small-sized nozzles do not line up either with his large one or with the Bailey nozzle, possibly as a result of differences in relative roughness of the throat finish. Then too, recent researches on orifices show the improbability of making orifice throats less than 0.5 in. so as to conform with larger diameter throats. This same observation applies to nozzle throats as well. Thus, such small sizes must be considered in a

[1] Published as paper RP-56-10, by J. F. Downie Smith, in the October, 1934, issue of the A.S.M.E. Transactions.

[2] Bailey Meter Company, Cleveland, Ohio. Assoc-Mem. A.S.M.E.

[3] "Fluid Meter Nozzles," by B. O. Buckland, Trans. A.S.M.E., 1934, paper FSP-56-14.

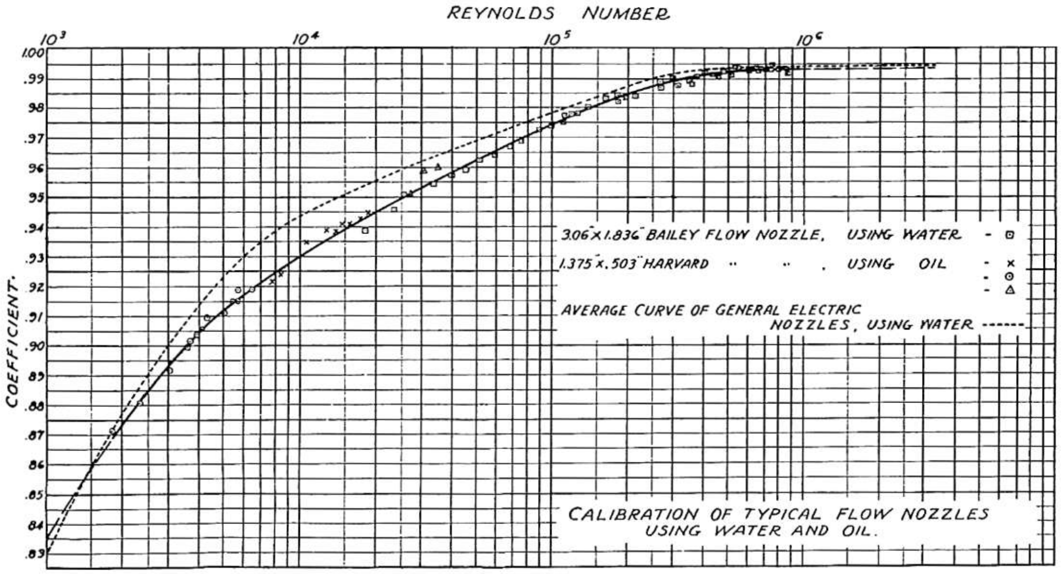

FIG. 1 CALIBRATION OF TYPICAL FLOW NOZZLES USING WATER AND OIL

class by themselves and cannot be grouped with larger ones without the application of some correction factor to compensate for these variations in finish.

An interesting comparison with the average curve of the General Electric nozzles, described by Mr. Buckland,[3] is made possible by placing this average curve on the writer's Fig. 1. The maximum difference between the two curves is about $1^1/_2$ per cent, and it occurs at a Reynolds number of approximately 10,000. At higher and lower Reynolds' numbers, particularly at the higher Reynolds number, the agreement between the two curves is better than $^1/_2$ per cent. Because of the rather spotty data of the General Electric nozzles at Reynolds' numbers less than 50,000, it is believed by the writer that the heavy curve shown in his Fig. 1, as developed by Mr. Smith and the Bailey nozzle, is slightly more favorable. However, if a mean line were drawn between the two, the maximum deviation would not exceed $^3/_4$ per cent, with the average between $^1/_4$ per cent and $^1/_2$ per cent at the more useful ranges which is an accuracy that will conform to most test specifications.

RONALD B. SMITH.[4] While the results of this paper serve as further confirmation of the Reynolds criterion it would seem to me that this is the ideal application of the sharp-edged orifice rather than the nozzle. It is extremely difficult to reproduce accurately the author's approach radii on nozzles $^1/_4$ in. and $^3/_{16}$ in. in diameter so that the coefficients cannot be applied to other nozzles with certainty. For instance, the author's coefficients of the geometrically similar large and medium-size nozzles do not agree. In addition, the use of a $^1/_{16}$-in. throat hole in only a $^1/_8$-in. nozzle must result in some abnormality in the flow.

Apart from the author's research it may be of interest to point out that in the regions of laminar flow the use of a pitot tube at 0.15 diam from the pipe wall offers no advantages over the usual static hole as far as accuracy is concerned. While the author does not mention the straight length upstream of the nozzles, let us assume that it is sufficiently long so that a parabolic profile is nearly developed. Although this would require considerable length it is approached in about 50 diameters.[5] Now the impact tube at 0.15 diam from the wall measures practically the average velocity for the parabolic (or the one-seventh turbulent) profile, yet it is known that the kinetic energy of the parabolic profile is twice the square of the mean velocity. Inasmuch as the development of the flow equation is essentially a balance of energies it would then be more rational to locate the tube at the rms velocity point when there is a laminar flow. This would be at about 0.3 diam.

ED S. SMITH, JR.[6] The author's data cover a range of Reynolds' numbers of present interest for the nozzle having an impact tube in the inlet. The curve in general parallels that for the Herschel Standard venturi tube, falling consistently several per cent below it.

The writer considers an impact tube, spaced only 0.15 diam from the pipe wall as tested by the author, to be a poor pitot on account of the steep velocity gradient so near the wall. It would seem that this tube location would be unduly liable to error at low Reynolds' numbers where the velocity-distribution curve has a parabolic form, i.e., an extended nose in the center. The use of straightening vanes is indicated in this flow régime.

In spite of the foregoing objection, the author's tests show an excellent correlation of coefficient with Reynolds' numbers, thus establishing the relation usefully for the particular nozzle-impact tube forms used.

SANFORD A. MOSS.[7] This paper shows a great deal of precise flow-measurement work, and is a distinct contribution to our knowledge of the properties of rounded-approach nozzles. One of the contributions is evidence in the matter as to whether or not Reynolds' number is a proper criterion for the plotting of nozzle coefficients. The author's curves in Figs. 3 and 4 do not at all coincide as they would if Reynolds' number were a complete criterion. The curves are also a little lower than the Reynolds number curve given by Mr. Buckland.[3] It has been suggested that the coefficient of various nozzles might be brought together if "Head" were used as the abscissas, rather than the Reynolds number, and this is worth investigating. Mr. Smith uses as the

[4] Turbine Engineering Department, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa. Jun. A.S.M.E.

[5] "Aero- and Hydromechanics," by Tietjens-Prandtl, vol. 2, pp. 25–28.

[6] Hydraulic Engineer, Builders Iron Foundry, Providence, R. I. Mem. A.S.M.E.

[7] Research Engineer, General Electric Company, West Lynn, Mass. Mem. A.S.M.E.

ordinate of his curves, on Figs. 3 and 4, the velocity coefficient whereas Mr. Buckland uses flow coefficient as the ordinate for his curves, which is the coefficient occurring in the theoretical formula for weight flow. Might it not have been a little more useful for computations involving use of the nozzle, as well as easier in the computations for finding the coefficient, if Mr. Smith had also done this?

Richard G. Folsom[8] and J. A. Putnam.[9] Rounded-approach orifices with cylindrical downstream sections were developed with a view to obtaining a flowmeter having a constant-discharge coefficient near to unity. The added feature of the impact tube was introduced to simplify the flow equation in that it automatically takes into account the velocity of approach, when placed in the correct position. Such an arrangement has considerable value when metering gaseous fluids. However, in handling liquids the law of continuity is so simple that the additional constructional and experimental difficulties of the impact tube far overshadow its advantages.

Mr. Smith's paper and other recent publications[10,11] clearly illustrate the characteristics of this type of metering device at both low and high values of Reynolds' number. The discharge coefficient drops rapidly at low values of Reynolds' number similar to the corresponding characteristic of the simple diaphragm-orifice.

Fig. 2 of this discussion shows the calibration curve of a small square-edged orifice used for metering in the mechanical laboratories of the University of California and which is comparable with the small orifice used by Mr. Smith. The meter conforms in general with the I.S.A. 1930 orifice, except that the diameter is less and the edge is thicker than the tolerance limits set by the I.S.A. The pressure connections are placed so that accidental burrs can have no effect. A disadvantage of the meter used by Mr. Smith is the position of the static-pressure connection in the high-velocity section where errors due to burrs will be a maximum.

Although the nozzle-impact-tube meter coefficients are higher, they vary as much as the coefficients of the simple orifice in the region of low Reynolds' number. At high Reynolds' number both types have a constant coefficient.

Since there is no choice on the basis of discharge-coefficients, the simpler sharp-edged orifice proves to be the most satisfactory meter under operating conditions. For accurate work, all small meters must be calibrated in place.

R. J. S. Pigott.[12] Mr. Smith is to be congratulated on the excellent consistency of his test work. In comparison with other tests on small nozzles, the scatter of points is quite noticeably less than usual.

8 Instructor, Department of Mechanical Engineering, University of California, Berkeley, Calif. Jun. A.S.M.E.

9 Teaching Assistant, Department of Mechanical Engineering, University of California, Berkeley, Calif.

10 "Regeln für Die Durchflussmessung mit genormten Düsen und Blenden," V.D.I., 1932.

11 Détermination du Coefficient de Débit de Tuyères et Orifices Noyés," by MM. P. Leroux et Deullin, *Annales des Mines*, ser. 13, vol. 4, no. 11, 1933.

12 Staff Engineer, in charge of engineering, Gulf Research & Development Corporation, Pittsburgh, Pa. Mem. A.S.M.E.

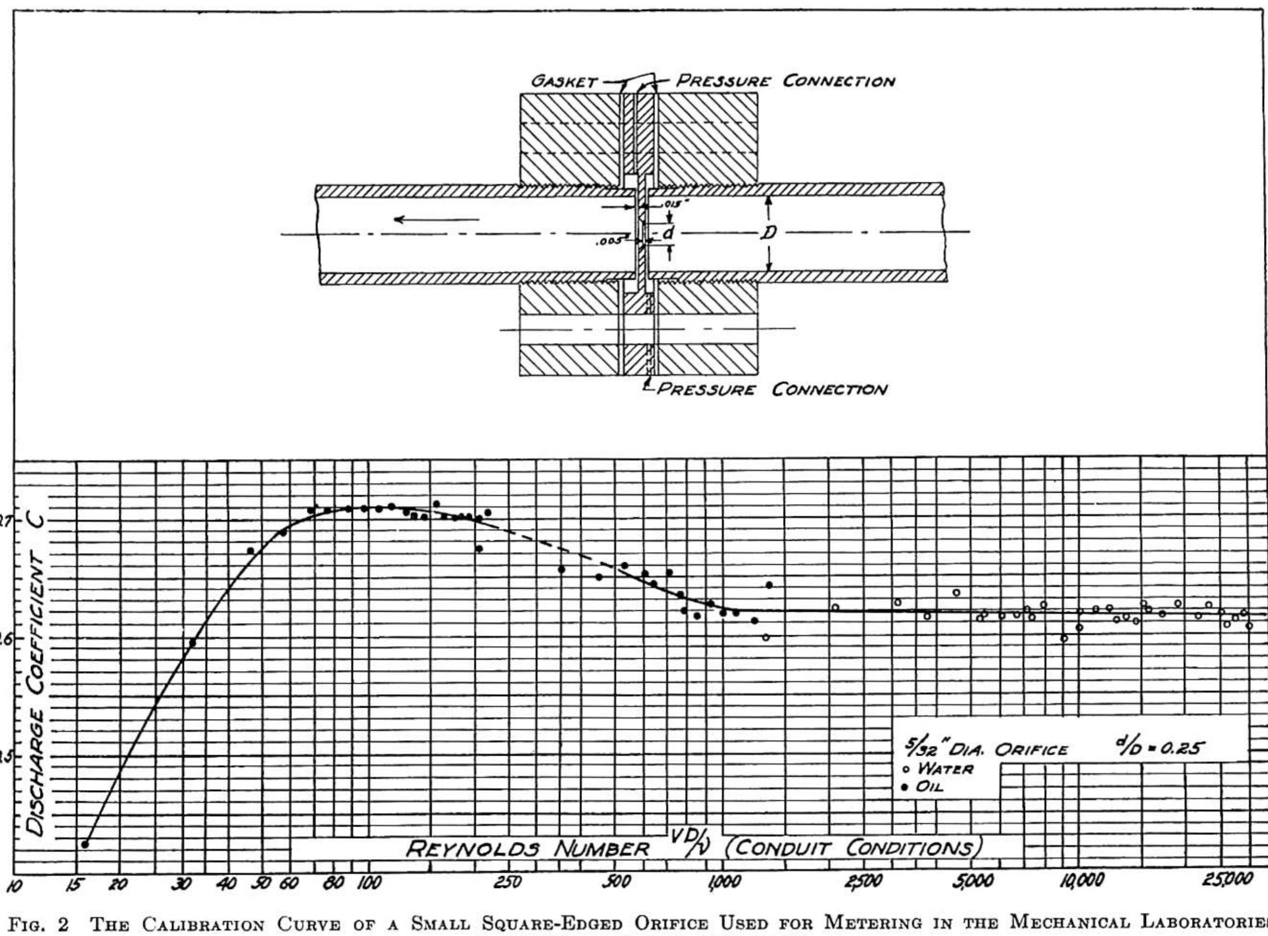

Fig. 2 The Calibration Curve of a Small Square-Edged Orifice Used for Metering in the Mechanical Laboratories of the University of California

It is assumed that the pipe used with these nozzles was drawn brass, although it is not so stated in the paper. The point is of importance, as the roughness of the preceding pipe has a definite effect upon coefficients.

Regarding the use of the impact tube located at 0.15 diam, the writer used this location in 48-in. pipe in 1910, for steam sampling. It gives a fairly good average velocity reading for fairly large Reynolds numbers, but it is certainly not rigorous. Such an impact tube does have an influence on the coefficient, since it disturbs the flow into the nozzle.

With regard to the dip in the coefficient curve, for the medium orifice it is possible that the impact tube for this particular set-up was at a critical position. There is also another cause for disturbances in this region; the plots are usually based on Reynolds' number in the throat. Turbulence usually begins at $dv\rho/\mu = 1200$ to perhaps 2500. But at this time, the upstream section is in viscous flow, and full turbulence for the whole nozzle does not occur until $dv\rho/\mu$ in the throat is greater than 3300 for the large nozzle, 6600 for the medium, and 13,200 for the small nozzle. Indeed, if the flow is very smooth in the upstream pipe, turbulence may not be fully established until about twice the foregoing values. Consequently, there is some instability in the orifice in this range; it may be detected by a tendency of the head gages to oscillate, since there are at least two possible extreme values for the differential. Very likely the dip is due to a combination of the impact tube and the unstable condition.

The writer would assign even the small variations in coefficient to three factors: (*a*) non-similarity; (*b*) difference in relative roughness; (*c*) the impact tube.

For the past ten years, it has become customary to plot venturi-tube, disk-orifice, and nozzle coefficient against number, assuming geometrical similarity, because the venturi was the same general shape, the orifice was flat and had a sharp edge, or the nozzle was the same contour. This practice is unfortunate, as it is not fair to the Reynolds number comparison. It is a fact that almost no completely similar nozzles have been tested. If the same 12-in. steel pipe is used for a five-, four-, and three-inch nozzle, although the nozzles themselves are similar in contour, the assembly is *not* similar. Further, if a 12 by 3 and a 4 by 1 nozzle are compared, using steel pipe in both cases, they are dissimilar; the four-inch pipe is *relatively* much rougher; so is the one-inch nozzle, with the same polish.

Mr. Smith's nozzles are approximately similar only. In the fluid-meters-nozzle research at present proposed, we intend to establish full geometrical similarity, varying roughness and diameter ratio. In this way, those deviations hitherto all charged to test variation will certainly be reduced to test errors purely, without the additional scatter resulting from dissimilarity.

These small nozzle tests are very much needed to fill out the lower Reynolds number region, and are particularly timely for the writer.

Author's Closure

Mr. Moss has pointed out that the three curves given in the paper do not coincide and on this basis concludes that Reynolds' number R is not a complete criterion to use as abscissas. It would appear to the author, however, that such a conclusion from the present data is unwarranted. In the first place, the orifices are not quite geometrically similar, with the result that a dimensional-analysis treatment of the problem would not give us C as a function of R only. Other dimensionless groups involving lengths would unquestionably enter.

The fact that these curves are slightly lower than those of B. O. Buckland can be explained perhaps by the lack of similarity in the two pieces of apparatus, and by his use of static pressure upstream, whereas the author used impact pressure upstream.

Mr. Moss's suggestion to plot C against head rather than R has been tried. It yields points with the present data that are scattered considerably more than where R is used.

Mr. Moss points out that the author uses the simple equation $v_{\text{theor}} = \sqrt{2\,gh}$ rather than $v_{\text{theor}} = \sqrt{\dfrac{2\,gh}{1 - r^4}}$ used by B. O. Buckland where r is the ratio of throat diameter to upstream diameter, and states that the latter is the coefficient occurring in the theoretical formula for weight flow. He has overlooked the fact that there is a difference in set-up in the two cases. Using static pressure upstream, the latter of the two formulas is correct, but if an impact tube is to be used, the former equation is the correct one, provided that the impact tube is placed at the proper place to get the desired velocity head.

Ronald B. Smith, Ed S. Smith, Jr., and R. J. S. Pigott state that the use of an impact tube at 0.15 diam from the pipe wall offers no theoretical advantages over the usual static hole. The author agrees with this. R. B. Smith's logic in discussing the kinetic-energy relations cannot be questioned. Strangely enough, the author used reasoning quite similar in discussing a paper presented by Prof. L. S. Marks on air flow in fan ducts a few days before these present discussions were presented.

When these orifices were installed the question of possible errors arising in coefficient due to erroneous readings of velocity upstream was investigated. Perhaps a résumé of the conclusions reached at the time would be illuminating. If an error of 50 per cent in velocity upstream were made, the effect on the velocity at the throat of the big orifice was 1.6 per cent, for the medium orifice it was 0.1 per cent, and for the small orifice it was entirely negligible. These conclusions are based on the assumption that an error of 50 per cent in the velocity upstream would be shown up in the flow through the orifice, but even this assumption is unjustified, since an error in the velocity upstream obtained for each orifice during calibration would be mainly counterbalanced in the use of the orifice during tests on flow measurement. Thus the small errors mentioned are considerably larger than any mistakes which would occur in the use of such an instrument, and the conclusion that the introduction of an impact tube would lead to negligible errors was borne out by the results obtained.

Now there is no single point upstream which would give the proper impact pressure over a wide range of Reynolds' number, particularly if the flow may change from viscous to turbulent, but the movement of the position of the impact tube at every reading would have introduced many complications and, in view of the small errors introduced, as previously mentioned, the distance 0.15 diam was adopted, as recommended in the Power Test Codes Tentative Draft, series 1933, Instruments and Apparatus, part 2, p. 13, and previously recommended by Sanford A. Moss in a verbal communication.

R. E. Sprenkle has presented a most remarkable verification of the author's data in his curve, especially as the apparatus used and liquid flowing in each case were different. Such close agreement is very gratifying.

Several discussers, including R. J. S. Pigott and R. B. Smith, have asked about the relative roughness of the orifices and approach pipes. That question the author cannot answer quantitatively. All orifices were made of composition metal (a brass) and were machined as smoothly as our shops could make them, templates being used in turning and polishing. The finish was bright and apparently glassy-like in smoothness. The pipes on each side of the nozzles were standard iron pipes, of ordinary roughness.

Mr. Pigott's comments on the dip noticed with the medium orifice are interesting, and obviously true; but they do not explain fully why this dip was found with only the one orifice.

Messrs. Folsom and Putnam state that the static-pressure connection in the author's apparatus is at the high-velocity section where burrs would have a maximum effect. The removal of burrs formed was not a serious matter, and the data obtained would seem to demonstrate that any irregularity left had little effect on the coefficients of the orifices.

The determination of the static pressure at this point has the definite advantage that the pressure is obtained under relatively stable conditions. A static-pressure connection immediately after the orifice, as used by Messrs. Folsom and Putnam, is not desirable, as this is the position of unstable turbulence. The eddies formed by the fluid immediately after passage through the orifice are very troublesome, although it is possible that in the extreme corner they would have little effect. The graph shown by the discussers has points departing by as much as 3 per cent from the mean line drawn, in the region of ordinary operation. In many tests this deviation is not allowable. It is true that over a range of Reynolds' number (conduit conditions) of from 1000 to 30,000, as shown in the discussers' graph, the coefficient is relatively steady, and this has definite advantages if a rough automatic measuring device is to be used. Otherwise, however, it is not a difficult matter to calculate the Reynolds number and pick the coefficient from the proper graph.

The V-Notch Weir for Hot Water[1]

H. N. Eaton.[2] This paper illustrates, in an interesting way, the fact that frequently, by varying one of the physical quantities involved in a physical phenomenon, we can determine what would be the effect of varying a different physical quantity which is also involved in the phenomenon. In the present instance, the effect on the coefficient of the V-notch of varying the head acting on the notch is used to show what would be the effect, over a limited range of the coefficient curve, of varying the kinematic viscosity of the water flowing through the notch. The advantage of this procedure lies in the fact that it is much more difficult to vary the kinematic viscosity of the water than to vary the head on the notch, at the same time controlling the conditions carefully enough to obtain accurate measurements. This expedient has been utilized to advantage in other branches of engineering and physics, particularly in aerodynamics, and the writer is interested to see an application of it made to hydraulics.

The process of reasoning by which the author arrives at his plot of C against $h/\nu^{2/3}$ appears to be correct, but the following treatment is suggested as a more direct and logical one.

We start with the customary formula for the V-notch

$$Q = Ch^{5/2} \ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where Q is the volume rate of flow, h is the head on the notch, measured above the vertex, and C is the coefficient of discharge of the notch.

We wish to determine how C is affected by the different physical and geometrical quantities which are involved in the phenomenon.

The following quantities may affect the flow Q, and hence the coefficient C: The head h, the acceleration of gravity g, the density of the water ρ, the viscosity of the water μ, the surface tension of the water s, the width of the approach channel b, the height of the vertex of the notch above the floor of the approach channel H, the width of the crest of the notch plate w, the angle of the notch α, the factor of a relative roughness k of the upstream surface of the notch plate, and to a lesser extent the roughness of the walls of the approach channel.

We can express this dependence as follows:

$$f_1\ (Q, h, g, \rho, \mu, s, k, \alpha, H, b, w) = 0 \ldots\ldots\ldots [2]$$

where f denotes "function of."

From these eleven significant quantities we can form $n - i$ dimensionless products, where n is the number of significant quantities and i is the number of physical dimensions required to express these quantities (in this case three—mass, length, and time). Hence, eight dimensionless products result, by means of which we can express our relationship as follows:

$$f_2\left(\frac{Q}{g^{1/2}h^{5/2}}, \frac{Q\rho}{h\mu}, \frac{s}{\rho g h^2}, k, \alpha, \frac{h}{H}, \frac{h}{b}, \frac{h}{w}\right) = 0 \ldots\ldots [3]$$

The particular forms of the products we choose depend upon the particular relationships we wish to study, and for different purposes we can express the same relations in many different forms. A concrete illustration of this will be given later in this discussion in passing from Equation [4] to Equation [5].

The first three of the dimensionless products chosen above were designed to separate clearly three different effects: first, the balance between the inertial and gravitational forces expressed by the product $\frac{Q}{g^{1/2}h^{5/2}}$; second, the balance between inertial and viscous forces, expressed by $\frac{Q\rho}{h\mu}$; and third, the balance between surface tension and gravitational forces expressed by $\frac{s}{\rho g h^2}$. The fourth variable, k, is a dimensionless roughness factor, and the last four are purely geometrical factors which express the form, but not the size, of the notch and the approach channel.

The first three dimensionless products, because of the particular force ratios which they represent, correspond, respectively, to the Froude, Reynolds, and Weber numbers. However, the names "Froude number" and "Reynolds number" should not be applied to the first two, because these names are used in a more restricted sense to apply, respectively, to the square of a velocity divided by a length and the acceleration of gravity, and to the product of a length and a velocity divided by a kinematic viscosity. It has been suggested to the writer by Dr. L. B. Tuckerman of the National Bureau of Standards that the names "generalized Froude number" and "generalized Reynolds number" be applied to these two dimensionless products. The name "Weber number" is usually applied to the dimensionless product $v^2 l\rho/s$, which represents the balance between inertial and surface-tension forces, instead of the form given above. This name has not yet become as fixed in its usage as have "Froude" and "Reynolds" numbers, and, since these two forms of the Weber number both take account of the effect of surface tension, no distinction will be made here.

The surface tension of the water affects the coefficient curve only at very low heads and is probably of no significance over the range discussed by the author of the paper. Consequently, the dimensionless product $s/\rho g h^2$, will be omitted from consideration in what follows. It is interesting to note that, if surface tension can be ignored, the density then appears only in combination with the viscosity in the form of the ratio μ/ρ, which we call the kinematic viscosity, ν, and hence, under this condition, the density and viscosity need not be included separately in [2] but can be replaced by the kinematic viscosity.

The width of the approach channel and the depth of the floor below the vertex of the notch will affect the coefficient at high

[1] Published as paper RP-56-9, by Ed S. Smith, Jr., in the October, 1934, issue of the A.S.M.E. Transactions.

[2] Acting Chief, Hydraulic Laboratory Section, National Bureau of Standards, Washington, D. C. Mem. A.S.M.E.