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TECHNICAL MEMORANDUMS

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

No. 952

STANDARDS FOR DISCHARGE MEASUREMENT WITH
STANDARDIZED NOZZLES AND ORIFICES

German Industrial Standard 1952

Fourth Edition

VDI-Verlag. G.m.b.H., Berlin, 1937

L. L. ...

Washington
September 1940

PREFACE TO THE THIRD EDITION*

The third edition of the Standards has been revised on the basis of new knowledge which has been collected through application of the first two editions to practice.

The construction of the Standards is now clearly divided into definitions of standardized designs of throttling devices and their standardized installations, and guiding information which is to be considered when deviations from the standardized manufacture or the standardized installation occur. Through this division the application of the standards in practice should be simplified and, at the same time, their range of use is extended to those cases that are to be handled with low requirements for accuracy.

For the simple and convenient application of the Standards, new Data Sheets have been compiled, giving clearly the essential diagrams and equations. Handling of the Data Sheets assumes a knowledge of the text of the Standards.

In contrast with the second edition, the third edition contains a series of changes of basic significance. Deference to the 1912 standard nozzle has been dropped, since this form will hardly ever be used in the future.

The discharge coefficient of the standard nozzle has been newly investigated for ratios of areas $m > 0.5$ up to $m = 0.64$. Those investigations have shown that the discharge coefficients for $m > 0.35$ are lower than given in the first and second editions, and that the basic tolerance for m is to be increased for high values of m . Figure 2, Data Sheet 1, shows the design of the standard nozzle for $m > 0.45$, and Data Sheet 5 gives the new discharge coefficients together with additions and tolerances.

New investigations have also been made on the standard orifice. These have confirmed the discharge coefficients

*This translation was made by Mr. Lyman M. Van der Pyl for the American Society of Mechanical Engineers with the permission of the Verein deutscher Ingenieure. The changes and additions made to create the fourth edition appeared as a list at the end of the third edition in the original copy of the translation. In this Technical Memorandum the NACA has incorporated the changes and additions in the text and illustrations.

of the **second edition**. However, for high values of m the **basic tolerance** above the tolerance limit **has been increased to 1 percent; see Data Sheet 5**. The dimensions of the standard orifice **have been determined more accurately** by suitable new **prescriptions**.

The bibliography **lists, besides** new publications of **basic significance**, only those which have appeared since the last edition and **which have direct significance for the Standards**.

The **revision of the third edition** has been undertaken by the Working Committee, **consisting of Messrs.**

Dr.-Ing. H. Hansen, VDI, Göttingen
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Sincere **thanks** are due them. **Special** thanks are due Dr. Witte, who **has again** contributed the valuable **basis** for the third edition. Other collaborators are likewise to be thanked for their valuable cooperation.

The international standards of the I.S.A. (International Federation of National Standardizing **Association**), suggested by the I.E.C. (International Electrotechnical Commission), adopted at the last meeting in Stockholm, **September 1934**, are based principally on the **same** German investigations **as** those **underlying** the VDI Standards. In **this** way, agreement of the VDI Standard with all **essential provisions** of the I.S.A. **Standards** is guaranteed at the start. In particular, the **designs** of the **German standard nozzle** and **standard orifice** and **also** the **discharge coefficients** are the **same as** those of the **I.S.A. nozzle** and the **I.S.A. orifice**.

Blow Meter Committee of
 the **Verein deutscher Ingenieure**

Josse,
 Chairman.

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PREFACE TO THE FOURTH EDITION

In the fourth edition of the Standards, the equations and make-up used in the third edition: are **essentially unchanged**. From experience and on the basis of new tests, a few but quite important changes, as well as several corrections and additions, have been made. These are concerned principally with the dimension of the cylindrical part s' of the standard orifice (see section 18) and the displacement of the tolerance limit in the direction of higher Reynolds numbers. (See section 48.)

A short section on the permanent pressure loss is given again, as in the second edition. Also the section on the conditions when using nonstandard pressure taps is treated more fully as compared with the second edition, in order to enable correlation with foreign investigations, which are frequently conducted with nonstandard pressure taps.

Plow Mater Committee of
the Verein deutscher Ingenieure

Josse,
Chairman.

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NATIONAL ADVISORY COMMITTEE BOB AERONAUTICS

TECHNICAL MEMORANDUM NO. 952

STANDARDS BOB DISCHARGE MEASUREMENT WITH
STANDARDIZED NOZZLES AND ORIFICES*

A. STANDARDS

I - RANGE OF VALIDITY OF THE STANDARDS

a) Purpose of the Standards

1. The measurement of flowing mediums by the pressure loss they undergo in passing through throttling devices (nozzles and orifices) is in many and important technical cases the only usable metering process: in addition, it is an accurate and easy method of general applicability to any flowing liquid, gas, or vapor at any pressure or temperature - with the limitations given in paragraphs 8 to 13.

2. The metering process consists in using as a measure of the discharge the change in pressure energy of a fluid flowing in a pipe of variable cross-sectional area. The energy law (Bernoulli) furnishes us with an unequivocal relation between the energy of pressure and the energy of flow. In order to use it, a reduction in area is provided in a pipe line by installing a throttling device: a conversion of part of the pressure energy to energy of flow is produced by forcing the fluid to flow through this reduced cross section.

3. The following standards give the standardized forms for two throttling devices. standard nozzles and standard orifices, and enable them to be used in circular pipes without calibration. The definitions of the standards are applicable in principle to the calibration and use of non-standardized throttling devices, such as the venturi tube. The standards are valid, likewise, as a basis for discharge measurements in the German acceptance standards.¹

*"Regeln fuer die Durchflussmessung mit genormten Duesen und Blenden." VDI-Verlag G.m.b.H., Berlin, N.W., 7, 1937.

¹See the standards for acceptance tests of steam turbines, steam boilers, compressors, internal combustion engines, etc., published by the VDI-Verlag, Berlin.

Translator's note. - The American Society of Mechanical Engineers secured permission from the Vereindeutscher Ingenieure for the translation of these standards, on the condition that no translations were to be sold.

Further, they serve as guides for measurements in which high accuracy is not required and for those service measurements in which standard devices are used, though not in accordance with the standard requirements.

4. For the standard nozzle and the standard orifice, the standards give the design and installation standards and the discharge coefficients, which are substituted in the theoretical flow equations to determine the true discharge or flow (discharge per unit time) from the measured pressure loss - the differential pressure. The corresponding tolerance - also called the variance between measurements - refer not to the whole measurement but only to the discharge coefficients. They indicate the limits of our present knowledge of flow processes and, in conducting practical measurements, must be increased by an amount dependent on the inaccuracy of measuring the differential pressure and the density.

5. A complete measuring arrangement consists of the throttling device (nozzle or orifice) installed in a pipe line, a differential pressure meter (discharge manometer)^a, and a means of conducting the pressure from the throttling device to the differential pressure meter. To this may be added, if necessary, the auxiliary arrangements for measuring the density or the values of the conditions, temperature and pressure, from which the density under flowing conditions may be obtained with the aid of the usual laws.

6. The total tolerance of the discharge measurement varies with the type of throttling device and differential pressure meter as well as with the type of application. A total tolerance may be determined only if the individual tolerance for the discharge coefficient, the differential pressure measurement, and the density are known. This is calculated as the "average error" in accordance with the law for combining errors, by taking the square root of the sum of the squares of the individual tolerances (reference 15).

If the individual tolerances (independent of one another) are $x_1, x_2, x_3 \dots$, the total tolerance is

$$x = \sqrt{x_1^2 + x_2^2 + x_3^2 + \dots}$$

^a This may be an indicating or recording device. If necessary, it may operate an electric telemeter or supplementary apparatus for computing the total amount, integrating the discharge against time.

In this, the individual tolerances are to be in percent (see Data Sheet 5) and it may be noted that tolerances for values under the square root in the discharge equation should be halved.

An example of steam and condensate measurement in turbine acceptance tests is given in section C VI.

7. The standards not only discuss the standardized throttling devices, but also bring out details of the pressure piping and present a survey of the commercial differential pressure meters. To set standards and accuracy limits for these is not possible with the present active development of metering apparatus.

b) Requirements of the Substance to Be Measured

8. Discharge measurements may be undertaken only if the density of the flowing substance is known accurately under the conditions upstream from the throttling device, since this always enters the measurement result. On the other hand, it is generally sufficient to know the viscosity approximately in order to calculate an approximate value for the Reynolds number for the discharge being measured.

9. In order to use the standards, the material to be measured must be in a pure phase when it flows through the throttling device. To fulfill this requirement, for example, liquids may contain gases or solids only in the dissolved form, and no separation of gas or vapor should take place. There should be no undissolved substances such as mud.

10. This requirement has Particular significance in the measurement of nearly saturated gases and vapors, and of liquids near the boiling point. It is necessary, in cases of doubt, to be certain that steam remains superheated and does not condense when undergoing the change of pressure in the throttling device.

11. The standards apply also for colloidal solutions if the degree of dispersion and the physical nature are such that the solution is only slightly different from a fluid of a single phase (such as milk). There has been no experience with thick, disperse solutions.

a) **Requirements** of the Flow

12. The flowing material must completely fill the cross-sectional area of the metering arrangement.

13. The discharge equations (see Data Sheet **3**) apply only for stationary flow. Consequently, the flow in discharge measurements with throttling devices should be at least quasi-stationary; that is, the velocity should **change** only slowly with time at a **given place**. In particular, there should be no pulsations in the flow as formed, for instance, by a reciprocating piston **engine**. Very little **investigational** work has been done to determine the metering error due to nonstationary flow: this error, therefore, cannot be corrected.

II - DESCRIPTION OF THE STANDARD NOZZLE AND THE STANDARD ORIFICE

a> The **German** 1930 Standard **Nozzle**

14. The **German** 1930 Standard **Nozzle** was first known as the "I.G. Nozzle." **Figure** 1 of Data Sheet 1 shows the standard requirements for the **outlines**³ of the standard nozzle and for the **length** and width of the pressure-tap **openings**, which may be **single** drilled holes or annular channels connected to the interior of the pipe by slits or by openings distributed around the circumference. (See also **figs.** 5 to 13. Data Sheet **1.**)

The standard nozzle is **applicable** to all pipe **diameters** \geq 50 mm. Tests **have** been conducted with pipe **diameters** between 50 and 500 mm and with ratios of area m^4 between 0.05 and 0.64.

With $m > 9.45$ the profile of the nozzle is to be turned out, as shown in **figure** 2, Data Sheet 1, until the **diameter** is **equal** to the pipe **diameter** D . At that point the upstream face of the nozzle is turned flat.

The downstream face is to be shaped **in** such a way that the downstream **pressure** tap **is** not disturbed.

³These correspond essentially to the standard forms in fig. 3 of the second edition.

⁴ $m = \frac{d^2}{D^2}$, d = nozzle diameter. D = pipe diameter.

The arcs with the radii $r = d/3$ and $r = 0.2 d$ are tangential to the cylindrical part of the nozzle and to the upstream face of the nozzle, respectively. Under these conditions and with the given principal dimensions, the middle point of the arcs defining the nozzle profile is located. The two arcs join practically without a break.

The small turned-out part (protective rim at the discharge of the nozzle) serves to protect the discharge edge and consequently may be omitted if no damage is to be feared.

15. The nozzle diameter d must be gaged with an accuracy of $\pm 0.001 d$. The cylindrical part of the nozzle should be turned with special care; subsequent polishing by hand is to be avoided since the discharge coefficient is changed considerably if the cylindrical part expands only slightly conically toward the discharge end or is wavy. A slight conical reduction exerts a lesser effect.

The nozzle profile must be tested with templates. However, the radii of curvature may differ from the theoretical by as much as 10 percent for $m < 0.3$ and by as much as 3 percent for $0.3 < m < 0.5$. However, there should be no break in the profile.

The surface of the nozzle should be smooth. The smaller the nozzle, the more carefully must it be made.

b) The German 1930 Standard Orifloe

16. The German 1930 Standard Orifice (fig. 3 of Data Sheet 1)⁵ is similar to the VDI standard orifice proposed in 1912. As with the nozzle, the differential pressure may be obtained either with single taps or with annular passages. (See figs. 5 to 13, Data Sheet 1.)

The standard nozzle⁶ may be used with all pipe diameters ≥ 50 mm. Tests have been made with pipe diameters of from 50 to 1000 mm and with ratios of areas m of from 0.05 to 0.7.⁶

⁵ Compared with the standard form in fig. 3 of the third edition, the limiting dimension of the cylindrical upstream part a ($\bar{I} 0.02 D$) is changed.

⁶ For measurements with $m > 0.7$, see reference 24.

17. The production of the **rectangular** edge on the upstream side of the orifice opening of diameter d - designated by arrows in figure 4, Data Sheet 1 - affects the discharge coefficient. If $d < 150$ mm, a sufficiently sharp **edge** may be obtained with certainty if a final **hair-fine** cut is taken off the upstream face, working outward from the orifice; if the upstream surface is turned first and then the **orifice**, the **edge** will not be absolutely sharp. The **larger** the orifice diameter, the more **allowable** is a **breaking** of the upstream **edge**, **visible** to the naked eye. With an orifice diameter of $d \geq 150$ mm, it is therefore **permissible** to **break** the **edge** with emery paper.

An edge must be considered as non-sharp, whether due to poor workmanship or to use in service, if the assumptions mentioned are not fulfilled - thus, if with $d < 150$ mm a ray of **light falling** on the edge is distinctly noticeable visually, or if with $d > 150$ mm, the **edge** is distinctly broken. For such non-sharp **edges**, the discharge coefficient must be increased (Data Sheet 5, fig. 24b), and increases in the tolerance (Data Sheet 5, fig. 24f) must be made. If there is any doubt of the **perfection** of the **edge** sharpness.

The upstream **face** should be smooth, at least near the upstream **edge** of the **orifice**.

18. The orifice **diameter** is to be measured with a tolerance of $\pm 0.001 d$.

The **plate** thickness should be equal to or less than $0.1D$ and the **length** of the cylindrical upstream part should be equal to or less than $0.02D$. The downstream bevel must be **greater** than 30° (usually 45°), in order to avoid a sucking-back of the **jet** to the **pipe wall**.

If the plate thickness $s \leq 0.02D$, the bevel on the downstream of the orifice may be **eliminated**, i.e., the cylindrical **length** s' of the **orifice** may be equal to s .⁷ In practice, this occurs when using large pipe **sizes**.

⁷ In the first and **second** editions, $s' \leq 0.01D$ was given as a **limiting** dimension. In the third edition, this **value** was raised to $\leq 0.04D$ in order to comply with many wishes of industry. New test results (reference 14a) indicate that, with this value, the **discharge** coefficient is appreciably **changed**. Consequently the value was reduced to $s' \leq 0.02D$.

c) Common Construction Details

19. Figures 1 to 3 of Data Sheet 1 give no construction details; they show only the details for the surfaces lying in the flow space and for the type of pressure tap. With annular chamber pressure taps, the opening of the annular slit should be small compared with the area of the chamber so that an equalization of pressure may take place. The answer to the question as to whether the annular chambers or the single taps are to be preferred, depends on the installation conditions. (See section III.)

20. With single pressure taps, condensation and the force of capillarity may falsify the measurement if the diameter of the tap is too small; for example, in measuring steam with overflow chambers for the condensate, the taps must be at least 8 mm in diameter if the upper chambers of figure XIX are used, though they may be smaller - about 4 mm - if the condensate pots on the side are used. As a result of this requirement and that in figure 1, Data Sheet 1 - that the width of the pressure tap $\leq 0.03D$ - only annular chambers are applicable for standard steam measurement in pipes of small diameter.

With annular chambers, the pressure tap slit should be as narrow as possible in order to equalize the pressure satisfactorily. (Note, however, the effect of surface tension if the slit is too narrow.)

21. In making the throttling devices, they may be split up into parts, as shown in figures 9 to 13 of Data Sheet 1, in order to facilitate manufacture, interchangeability, etc. The materials of the throttling devices must be so chosen that they will not wear rapidly. Particularly with orifices, care should be taken when shipping and installing that the sharp upstream edge is not damaged. Likewise, with nozzles the discharge edge should not be damaged. (See what is said about the protective rim in paragraph 14.)

22. The mounting rings which hold the orifice plate or the nozzle must have rounded corners on the upstream side (figs. 5 to 13, Data Sheet 1) and their inside diameters should not be smaller than the pipe diameter. It is permissible, however, to make the mounting ring 5 percent larger than the pipe diameter with standard orifices and 3 percent larger with standard nozzles, as long as the axial

distance of the ring from the front of the plate or nozzle dose not exceed **O.I.D.** The **ratio of areas** is not to be referred to the nominal diameter but **always** to the true measured pipe diameter on the upetrom side.

23. For pipes of large diameter, the narrow nozzle or orifice of figure 6, Data Sheet 1, is ordinarily used because it may be installed more easily and because of weight coneidoratione. The arrangement of figure 7 may be substituted for the annular chambers; in this case the pressure tap openings must be small compared, to the cross-sectional area of the equalizing line.

For pipes of small diameter it is recommended that the designs of figures 14 or 15, Data Sheet 1, (reference 9), be used in pipelines of which the characteristics (roughness, roundness, diameter, etc.) are not known accurately.

III - REQUIREMENTS FOR STANDARD INSTALLATIONS

(See also section C I)

24. The applicability of the tolerances given for the discharge coefficients assumes that the throttling device is installed very carefully. If this is not assured, metering errors considerably in excess of the tolerance will occur. At present there is no reliable knowledge by which these errors may be estimated or even corrected mathematically. However, some tests indicate that the metering error usually does not become larger, as frequently assumed, but smaller as the velocity of the substance being metered increases. In all cases, measurements with high ratios of areas are intrinsically more sensitive to all irregularities in installation than measurements with small ratio of areas, which is reflected in the tolerances. (See Data Sheet 2 and reference 24.)

The following instructions are to be observed if the tolerances of the discharge coefficients are not to be exceeded. In the plans for the pipe lines sufficiently long straight pipe lengths must be provided.

25. The faultlessly made throttling device is properly installed between two flanges of the pipe line. For pipe diameters below 7.0 mm it is recommended that the arrangements of figures 14 and 15 on Data Sheet 1 be used.

In measuring gas and steam, care should be taken that water collecting upstream and downstream from the throttling device be removed by a separator. There should be neither gaskets nor rough welded seams upstream from the throttling device. Therefore gaskets should be cut specially large - say about 1.1 D - to make certain that there is no projection at the mounting ring. (See fig. 5, Data Sheet 1.)

The throttling device is to be accurately centered in the pipe. For this requirement, it is generally satisfactory if the outside diameter of the mounting rings is equal to the flange bolt circle diameter minus the bolt hole diameter.

26. When gas containing dust, fog, or tar is to be measured continuously, it is necessary to make certain that the edge stays sharp and does not become dirty. In such cases it is recommended that the orifice be provided with a spray device and an observation arrangement⁸, approximately as in figure I. Such an arrangement allows the condition of the edge to be observed and to be sprayed from time to time as occasion demands.

27. The true pipe diameter is to be measured accurately for large ratios of areas, since it frequently varies from the nominal diameter. The pipe should be cylindrical upstream from the throttling device. The greatest change of the pipe diameter in a length of 2D ahead of the orifice - in consequence of noncircularity, conical expansion, e.g., as when flanges are rolled on to the pipe, etc. - should be within ± 0.5 percent for ratios of areas $m > 0.3$ and within ± 2 percent for $m < 0.3$.

28. The pipe line should be smooth or service-rough. The effect of pipe roughness on the discharge coefficient may be taken from paragraphs 47, 50, 51, and 54. The relative roughness, i.e., the ratio of the average height of the roughness to the pipe diameter, is determinative. For

⁸ Tests with two nipples 0.25 D in diameter welded into the pipe at a distance 9.5 D upstream from the orifice showed that they have no effect on the discharge coefficient α up to $m \leq 0.7$.

⁹ According to unpublished tests of the I. G. Farbenindustrie A.-G. with water gas and power gas containing approximately 12 mg of dust per cubic meter, the value of α increased by several percent in a few days.

this reason pipes of the same absolute roughness may be considered more nearly smooth as the diameter increases. Quantitative roughness may only be estimated. As a guide, a new cast-iron Internally tarred pipe of 200 mm I.D. may be considered smooth. A service-rough pipe may be understood to be a cast-iron pipe which has rusted internally through long use, but which has no thick encrustation on the surface. When measuring gas and steam in vertical pipes, the pressure taps must be made very carefully because of the possibility of collecting water.

29. The pipe line upstream and downstream from the throttling device (upstream section, downstream section) must be rectilinear and of constant diameter. The upstream section must be so long that all disturbances have died away at the throttling device, the downstream section so long that a disturbance at its end cannot work back to the throttling device. Data Sheet 2 gives for different, frequently encountered installations, the minimum necessary disturbance-free straight lengths of pipe, measured in pipe diameters D (upstream section E , downstream section A). If only half the given lengths of pipe are available, it may be assumed as a guide that, for $m \leq 0.5$, the additional error will be approximately 0.5 percent. For nozzlees with $m > 0.5$, no data can be given.

30. The necessary straight lengths of pipes upstream and downstream from the throttling device are, in general, shorter the smaller the ratio of areas. Consequently, in many cases where a sufficiently long straight length of pipe is not available, the choice of a smaller ratio of areas will be a help, insofar as the resultant increase in differential pressure will allow.

31. It is not permissible to install a throttling device close downstream from an expanding tapered piece. With orifices, a straight section of pipe of constant diameter must be between such a piece and the orifice, of a length increasing with increasing ratio of areas of the orifice and with increasing expansion in the tapered piece. As a guide will serve the fact that with orifices and an expansion to twice the pipe diameter, a straight upstream section of at least $10 D$ for $m < 0.2$ and at least $20 D$ for $m = 0.25$ to 0.50 must be used. With insufficiently long upstream sections, very considerable metering errors may result. (See fig. 34, Data Sheet 8.)

An installation downstream from a **tapered** reducer is **less unsatisfactory**. In this **case** also the error becomes greater as the ratio of areas **increases** and as the amount of **reduction** increases. Therefore, the greater the ratio of areas **m**, the longer the straight length of pipe of constant **diameter necessary** between the **throttling** device and the **reducer**. As an example, this distance should be at least 15 D when **m** = 0.3.

Sudden reductions in pipe size (without a tapered piece) **give greater errors**, tending to **reduce** the **discharge coefficient** (reference 9).

32. Space elbows (double elbows in two or more planes perpendicular to **each other**) cause **large errors**, since through them the flow **acquires a twisting** which is dissipated very slowly.

33. **Straightening** vanes (for making the lines of flow parallel to the walls of the pipes) are expedient only if there is a twist in the flow. To be **effective, they** must be from 1 to 2 D long. Downstream from single elbows, a straightening vane is usually harmful. The distance from the straightening vane to the **throttling** device should be so **great** that the **disturbance** produced by the straightening vane is dissipated in it. This distance will vary with the ratio of areas and the design of the **straightening** vane and will amount to 5 D for small values of **m** and as much as 20 D for high values of **m**.

34. Very **large** errors are caused by partially open valves. Table I on Data Sheet 2 shows the necessary **straight** disturbance-free length of pipe as a function of the **opening** of the **valve**.¹⁰ Valves installed downstream from the **throttling device** disturb the flow less: accordingly, when possible, they are installed downstream from the **throttling device**.

35. The disturbances have less **effect** when using annular chambers than when **using single** taps. **Although** there are no tests on the subject, it may be assumed that an annular chamber is **more** effective as the chamber volume **increases** and as the slit **connecting it** to the pipe **becomes** narrower.

36. Only **if** all the stipulations in this section are fulfilled, should an installation be termed standardized.

¹⁰According to unpublished work of the Oppau Works Control of the I. G. Farbenindustrie A.-G.

IV - WORKING FORMULAS

(See also Data Sheet 3)

37. In part B, section I, the general **discharge** equation is derived and put into the form

$$G = a \epsilon F_0 \sqrt{2g \gamma_1 (P_1 - P_2)} \quad (\text{kg/sec}) \quad (\text{I})$$

$$Q = G/\gamma_1 \quad (\text{m}^3/\text{sec}) \quad (\text{Ia})$$

measured **under** the conditions **upstream** from the **throttling** device. In these formulae γ_1 is the density in kg/m^3 , $P_1 - P_2$ is the differential pressure in kg/m^2 , and a and ϵ are empirical values which are to be taken from part V. (See also Data Sheets 5 and 6.) F_0 is the area of the free opening of the throttling device at the temperature of measurement **in square** meters. (See paragraph 40.)

38. The **discharge** coefficient a **takes** the following **factors** into consideration. (See also sections 78-80.)

1. The velocity of **approach** upstream from the **orifice**.
2. The friction of the **flowing** material upstream from and in the **throttling** device.
3. **With** orifices, the contraction downstream from the throttling device.
4. The pressure taps **in** the corners upstream and downstream from the throttling **device**; these pressures are not exactly equal to the **theoretical pressures** upstream and **downstream** from the **throttling** device.

39. The **expansion** factor ϵ takes into account the Irregular behavior of **gases** and vapors as compared **with** that of practically incompressible fluids (with $\epsilon = 1$): with gases and **vapors** the density **changes** in accordance with **thermodynamic** relations in **flowing through** the **throttling** device: ϵ **increases** in effect **as** the differential **pressure increases** with respect to the absolute pressure ahead of the throttling device (in the plane of pressure tap: see **paragraph** 65).

40. On the **basis** of tests, it may be **assumed** that the **diameter** of the opening of the throttling device **changes**

with the **linear** temperature coefficient of expansion; **figure 17** on Data Sheet-3 gives the increase in area with temperature for the most important materials, if **d** is measured at **20° C**. If the throttling device and the pipe are of different materials, the ratio of areas **m** changes with temperature since **d** and **D** are calculated with different correction faatore.

41. On Data Sheet 3 are aolleted the formulas for practical use. In these the diameter in millimeter⁶ la used instead of the area, and the discharge is referred to hours instead of seconds, in accordance with technical custom. The other values are to be eubetitted in terms of meters, kilograms, and seconds. After carrying out the continually recurring auxiliary calculation

$$3600 \times 10^{-6} \frac{\pi}{4} \sqrt{2g} = 0.01252$$

special formulas are given for liquids, steam, dry and wet gases. These formulae are worked with the density γ_h of the manometer sealing fluid and consider the fact that the density of the flowing substance over the sealing fluid and in the connecting lines map be quite different than under the conditions upstream from the throttling device.

42. See section A 71 concerning the measurement of the differential pressure $P_1 - P_2$ and the determination of the characterietice of the substances.

43. When calculating the diameter of throttling devices far given measurement conditions and given differential pressure, it is simplest to use figures 19 and 20 of Data Sheet 4.

.Substituting $d^2 = m D^2$ in the discharge equation⁸ and making some reductions, the result is the formulae

$$m \alpha = \frac{Q}{0.01252 \epsilon D^2 \sqrt{h \frac{(\alpha)}{\gamma_1} - \gamma_1}} \quad \text{with } Q \text{ in m}^3/\text{hr} \\ D \text{ in mm}$$

$$m a = \frac{G}{0.01252 \sim D^2 \sqrt{h(\gamma_h - \gamma_1) \gamma_1}} \quad \text{with } G \text{ in kg/hr} \\ D \text{ in mm} \quad (b)$$

For gases and vapors an estimated value of the expansion factor ϵ is to be substituted first and with this an approximate value of m is to be calculated.

With this value of m , the discharge coefficient C_d is read off on the figure already mentioned, and m is calculated. If a line above the tolerance limit and no addition for the pipe roughness is necessary, the value is final. If not, an approximation calculation must be made. (See example 1.) For incompressible fluids, $d = D \sqrt{m}$. For gases and vapors, the exact value for ϵ is first to be taken from figures 25 to 28, Data Sheet 6, and then the calculation is repeated, as many times as necessary. In this connection, see also the example in section AVII.

Figure II gives the equivalent ratios of areas m for nozzles and orifices which have the same differential pressure for the same discharge.

V - THE VALUES AND TOLERANCES OF THE DISCHARGE COEFFICIENTS AND CORRECTIONS

a) For Incompressible Fluids

44. For standard models and standard installations, the discharge coefficients and corrections with their respective tolerances are given on Data Sheet 5. These tolerances refer only to the discharge coefficients and the corrections.

45. The discharge coefficients C_d for the 1930 German Standard Nozzle are reproduced on a semilogarithmic chart shown in figure III as functions of the Reynolds number Re_D (tolerance limit) (see table I), the discharge coefficients are constant. These discharge coefficients are shown in figure 21, Data Sheet 5, as a function of the ratio of areas m .

46. Above the tolerance limit, the values of the discharge coefficients are valid with the basic tolerance (see references 21 to 24) given in figure 23d, Data Sheet 5; this tolerance increases from 0.5 to 1.3 percent with increasing values of m . Below the tolerance limit the basic tolerance amounts to ± 1.5 percent.

TABLE I. Limiting Values of Re_D

| Ratio of areas m | Limiting values of the Reynolds number | |
|-----------------------|--|---------|
| | Orifice | Nozzle |
| 0.05 | 20,000 | 70,000 |
| .1 | 30,000 | 73,000 |
| .2 | 50,000 | 91,000 |
| .3 | 80,000 | 110,000 |
| .4 | 125,000 | 170,000 |
| .5 | 170,000 | 190,000 |
| .6 | 225,000 | 200,000 |
| .65 | 260,000 | 200,000 |
| .7 | 300,000 | -- |

47. With small ratios of areas m for standard nozzles, no noticeable effect of pipe roughness is observed as long as the pipes are only normally rough. For $m \geq 0.3$ the roughness becomes evident through an increase in the discharge coefficient. Figure 23a gives, as a function of m , the correction factors for service-rough pipes of different diameters; figure 23e gives the corresponding additional tolerances.

48. The discharge coefficients α for the 1930 German Standard Orifice are shown in a semilogarithmic representation in figure IV as a function of the Reynolds number Re_D referred to the pipe diameter. The discharge coefficients are practically constant above certain limiting values of the Reynolds number Re_D .¹¹ These discharge coefficients are shown in figure 22 of Data Sheet 5 as a function of the ratio of areas m .

49. Above the tolerance limit the discharge coefficients for the orifice are valid for smooth pipe and for an absolutely sharp upstream edge of the orifice with the basic tolerance shown in figure 24d, Data Sheet 5. For $m < 0.35$ ¹² it amounts to ± 0.5 percent, and it increases steadily to ± 0.1 percent at $m = 0.7$. Below the toler-

¹¹The line connecting these limiting values for m is called the tolerance limit. As compared with the third edition, the line is displaced in the direction of higher Reynolds number for $m > 0.3$.

¹²Different from the third edition.

ance limit, the discharge coefficients are valid with a basic tolerance of ± 1 percent.

50. The pipe roughness causes a greater increase in the discharge coefficient α for the standard orifice than for the standard nozzle, and here also the effect increases with increasing ratios of area m , while with low ratios of area the effect of the roughness of service-rough pipe disappears. (See paragraph 28.)

51. The correction factors of the discharge coefficient for orifices in service-rough pipe and the additions to the basic tolerance for the effect of roughness are given in figures 24a and 24e on Data Sheet 5.

52. If the edge sharpness does not conform absolutely to the requirements given in paragraph 17, i.e., if a reflection of light from the edge is easily noticeable with $d = 109$ mm, further additions must be made for the lack of sharpness of the edge. These additions are dependent on the orifice diameter d ; the correction factors, for the lack of edge sharpness, are to be taken from figure 24b, Data Sheet 5.

53. Since the sharpness of the edge cannot be defined accurately, the increased discharge coefficients are valid with a tolerance consisting of the basic tolerance and an addition thereto; this addition is to be taken from figure 24f, Data Sheet 5.

54. In general, lack of sharpness of the edge and roughness of the pipe are to be considered simultaneously. In this case both correction factors, as determined from figures 24a and 24b, are to be applied to the discharge coefficient and the total tolerance of the discharge coefficient α , according to the law for combining errors, is equal to the square root of the sum of the squares of the tolerances for the lack of edge sharpness and for the pipe roughness; see examples, section VII.¹³

¹³For measuring gas with the standard orifice, with $D = 100$ mm, $m \approx 0.35$, $Re_D = 30,000$, in commercially rough pipe and with the orifice edge not absolutely sharp:

(Continued on p. 17)

55. If, in addition to the effects of pipe roughness and of lack of edge sharpness, there is also an effect from the viscosity of the flowing medium, the corrections for the different effects are to be attached separately. The tolerance for the corrected discharge coefficient is determined by means of the law for the combining of errors. (See paragraph 6.)

'b) For Gases and Vapors.

56. For the 19.30 Standard Nozzle, the expansion factor ϵ is to be taken from figures 25 and 26 of Data Sheet 6 for low differential pressures and from figures 27 and 28 for high differential pressures, for $\kappa = 1.31$ and $\kappa = 1.4$,¹⁴ respectively. Figure V gives the expansion factor for any value of κ as a function of $(P_2/P_1)^{1/\kappa}$ up to the critical pressure.

57. The calculated expansion factor for the standard nozzle for $m \leq 0.4$ shows absolute agreement with experi-

(Continued from p. 16)

1. The value of the discharge coefficient α
 - a. Basic value (fig. 22) 0.646
 - b. Correction factor for roughness (fig. 24a) 1.007
 - c. Correction factor for lack of edge sharpness (fig. 24b) 1.007
 - d. Correction for the effect of viscosity (fig. 24c) 1.008

The corrected value of a is $a \cdot b \cdot c \cdot d = 0.660$
2. The value of the expansion factor for $\frac{P_1 - P_2}{P_1} = 0.15$ (assumed) (fig. 26, Data Sheet 6). 0.95
3. Tolerance for a ϵ
 - e. Basic tolerance for a from 49 below the tolerance limit $\pm 1.0\%$
 - f. Additional tolerance because of roughness (fig. 24e) $\pm 0.9\%$
 - g. Additional tolerance for lack of edge sharpness (fig. 24f). $\pm 0.6\%$
 - h. Tolerance for ϵ (table II) $\pm 0.5\%$

$\sqrt{e^2 + f^2 + g^2} + he = \dots \pm 1.6\%$

¹⁴Different from the third edition, to agree with usual practice; the value of ϵ is practically not affected.

mental values. (See reference 22.) The tests were conducted with superheated steam up to the critical pressure ratio, in a range of the Reynolds number above the tolerance limit, and consequently without a simultaneous effect of viscosity. The same was found¹⁸ for air (with the 1912 standard nozzle) at a pressure ratio of $P_2/P_1 = 0.9$.

Therefore it may be expected that the expansion factor ϵ may be calculated with the theoretical equations for other Cases and superheated steam with the same or different κ . However, a tolerance for ϵ , given in table II, is recommended for the gases and vapors that have not been tested. For superheated steam and air the expansion factor ϵ is applied without tolerance, as long as $m \leq 0.4$ and

$\frac{P_1 - P_2}{P_1} \leq 0.2$ and as long as the Reynolds number is above the tolerance limit. Table II gives a summary of the tolerances.

TABLE II. Tolerances of the Expansion Factor ϵ for Standard Nozzles and Standard Orifices

| Pressure ratio $\frac{P_1 - P_2}{P_1}$ | Orifices | | Nozzles | | | |
|---|---------------------------|----------------------------|---------------------------|-------------------------|-------------------------|-----------------------------------|
| | For gases per- cent | For vapors per- cent | Superheated steam and air | | | Other gases and vapors percent |
| | | | $m \leq 0.4$ percent | $m \leq 0.5$ percent | $m \leq 0.6$ percent | |
| 0 to 0.01 | 0.0 | 0.0 | 0 | 0 | 0 | ±1 |
| 0.01 " 0.02 | ±0.5 | ±0.5 | | | | |
| 0.02 " 0.1 | ±0.5 | ±1.5 | | | | |
| 0.1 " 0.2 | ±0.5 | ±1.5 | ±1 | ±1 | ±1 | ±2 |
| over 0.2 | ±2 | ±2 | ±1 | ±2 | ±2 | |

58. The expansion factors for the 1930 standard orifice as determined by tests are also shown in figures 25 to 28. They are valid with the tolerance shown in table II as long as the Reynolds number is above the tolerance limit.

59. The tolerances for orifices are introduced principally because of the still unexplained jumps found in the tests with steam. (See reference 23.)

60. The tolerances for ϵ are to be combined with the tolerances for the discharge coefficient by means of the law for combining errors.

¹⁸ Unpublished tests of H. Mueller and H. Peters (Göttingen), with air.

VI - DETERMINING THE METEHING MAGNITUDES 'AND

THE MATERIAL VALUES

a) Differential Pressure

61. Differential pressure meters, with the exception of the two-legged U-tube, must be calibrated. They must be so designed that the unavoidable error of the device and the error of reading are within the limits corresponding to the desired test accuracy at the lowest differential pressure to be measured. Because of the quadratic relation between differential pressure and discharge, small errors in the differential pressure enter into a discharge measurement at approximately half rate.

With U-tubes the vertical distance between the levels (displacement) of the sealing fluid is to be determined.

| | | | |
|----|-------------|--|----------------------|
| If | h | is the displacement of the sealing fluid | (m) |
| | γ_h' | density of the scaling fluid | (kg/m ³) |
| | γ' | density of the material over the sealing fluid and in the lines to the pressure taps | (kg/m ³) |

than the differential pressure $P_1 - P_2 = h(\gamma_h' - \gamma')$ (kg/m²). The value of h in millimeters may be substituted for h in meters if γ_h' and γ' are used in kilograms per liter instead of kilograms per cubic meter.

62. If the lines to the pressure taps are filled with gas at low pressure $\gamma' = 0$. For water and mercury as sealing fluids and for mercury as a sealing fluid, if water is over it in the lines to the pressure taps, γ_h' , γ' , and $(\gamma_h' - \gamma)$ in kg/liter. (h in mm!) may be taken from table 2 of Data Sheet 3.

63. The connecting lines must be absolutely tight and be filled with the same medium over their whole lengths up to the pressure taps; In measuring gases, accordingly, care must be taken that no condensing-liquids, such as water from the moisture of the gas, collect over the sealing fluid: if they do collect, they must be measured and taken into con-

sideration. In measuring liquids, no bubbles of air or gas should be above the sealing fluid. In measuring vapors which might condense in the inlet lines, both lines must be filled up to the same level with the condensate. For this reason, the lines must be blown out with the vapor, with the shut-off valves of the differential pressure meter closed during the blowing-out. The vapor will then condense in the lines through cooling. For more detailed information, see part C, section V.

b) Density

64. The density of the flowing medium must be determined with the same care as the differential pressure. As with the differential pressure, the error enters into the result at only half rate.

65. The density γ_1 of the flowing medium is usually known for base conditions through measurement or calculation based on a chemical analysis. Consequently, with incompressible fluids, a measurement of temperature is sufficient; with compressible fluids, the pressure and temperature upstream from the throttling device must be measured in order to calculate γ_1 with the help of the equation of state. (See Data Sheet 3.) The pressure for this purpose is to be measured in the plane of the differential pressure tap upstream from the throttling device. This requirement need not be observed with small values of m and with small ratios of differential pressure to measurement pressure. The measurement temperature is to be taken as closely as possible to the throttling device; however, care should be taken that the flow is not disturbed through the installation of the thermometer.¹⁶ Table 3 of Data Sheet 7 gives the density γ_n for commercially important gases at two different "standard conditions": 20° C, 1 kg/m³ and 0° C, 760 mm of Hg.

¹⁶ According to unpublished tests of R. Witte, Ludwigshafen, with steam measurement, a thermometer well of outside diameter $s \leq 0.04 D$ at a distance of 13 D upstream from the throttling device has no effect on α , even at the highest values of m . On the contrary, a well with $s = 0.13 D$ at the same distance of 13 D reduced the values of α as follows: with nozzles, $m \leq 0.35$, 0 percent, with nozzles, $m = 0.6$, 1 percent; with orifices, $m = 0.5$, 1 percent, with orifices, $m = 0.7$, 1.5 percent. Consequently, the best location is downstream from the throttling device, since the installation disturbance would be less. The Joule-Thomson effect may usually be neglected in practice.

66. At high pressure, especially near the saturation line, the deviations from the ideal gas law are not to be disregarded. In the formula they are to be considered with the aid of the "compressibility factor" K . K is the ratio of the density calculated from γ_n by the laws for ideal gases to the true density. The values of K , for example, are procured from the Landolt-Börnstein physicochemical tables, but more easily from figures of the sort of those on Data Sheet 9, referred to 1 absolute atmosphere.¹⁷

With vapors, particularly steam, γ_1 is to be determined from steam charts or tables.

With gas mixtures of not accurately known composition, the density may be determined with the Bunsen-Schilling device¹⁸ or with a density recorder: it is to be recalculated to the measurement conditions.

67. If the density of the flowing fluid varies over a period of measurement, it must be ascertained at short intervals of time and the discharge must be determined currently with the ascertained values.

68. The density γ_1 of most gases, i.e., mixed with superheated water vapor, is to be calculated with equation (9) of Data Sheet 3. The theoretical derivation is given in part B, paragraphs 87 to 90.

Figure 18 of Data Sheet 3 shows the errors that result if the moisture of a saturated gas is neglected in calculating γ_1 with a pressure of 1 absolute atmosphere in the line. They are small at low temperatures as well as for gas mixtures with γ_n near 0.7 to 0.9; (0.727 is the ideal normal density of water vapor at 1 absolute atmosphere and 20° C, calculated by the ideal gas law from the density and the saturation pressure at 20° C). They become smaller with increasing absolute pressure P_1 .

¹⁷Witte presents a collection of the compressibility deviations for the most important gases and vapors in Eucken-Jakob's "Chemie-Ingenieur," vol. II, 2, p. 16 ff.

¹⁸The correct procedure and valuation of the tests with this device require great care; see Kretzschmer, *Forschung*, vol. 3, 1932, p. 150. Vunsch and Herning, "Bestimmung der Gasdichte nach dem Ausströmverfahren" (Determination of Gas Density by the Efflux Method). *Gas-u. Wasserfach*, vol. 79, 1936, p. 177.

69. Formulas (10) to (12) of Data Sheet 3 show the calculation of the measured discharged volume Q to dry volume and standard conditions. For dry gas, the formula is

$$Q_n = Q \frac{P_1 T_n}{P_n T_1} \frac{1}{K} \quad (\text{II})$$

in which K is the factor for the deviation from the Ideal gas law: see Data Sheet 9. For gas mixtures with water vapor, the water-vapor content is to be removed first in order to obtain the dry portion of the gas:

$$Q_{tr} = Q \frac{P_1 - \phi P_D}{P_1} \quad (\text{III})$$

With Q_{tr} calculated to standard conditions by equation (II), and Q substituted from equation (2), Data Sheet 3, and γ_1 substituted from equation (9), Data Sheet 3, equation (11) is obtained. For details of the effect of the moisture of gases, see paragraphs 87 to 90.

The approximate equation (12) is easier than equation (11): it is derived from equation (11) in such a way that only the dry gas is used in calculating γ_1 ; an error half as great as in figure 18, Data Sheet 3, is made thereby.

c) Viscosity and the Reynolds number

73. The viscosity of fluids is considerably dependent on temperature, but on the contrary, only slightly dependent on pressure. The dynamic viscosity of gases is independent of pressure only insofar as the gases conform to the ideal gas law.

The dynamic viscosities at the saturation line for ammonia, carbon dioxide and sulphur dioxide, and for liquefied gases and water are given in table 4, Data Sheet 7. Further, figure VI in the text gives the dynamic viscosities of the more-frequently encountered gases, figure VII that of water-up to 160°, and figure VIII those of steam from the values which at present may be considered as most probably correct; see also section 73.

71. The viscosity of gas mixtures does not follow the simple rule of mixtures, particularly not when the water-vapor content is high. Approximate equations have been de-

veloped.¹⁹ A formula, sufficient for most technical gases
 -(reference 26) is:

$$\nu_{\text{mixture}} = \frac{100 \nu_m}{O_2 + CO + CH_4 + N_2 + 2(CO_2 + C_m H_n) + \frac{1}{7} H_2} \quad (IV)$$

in which

ν_{mixture} is the kinematic viscosity of the mixture
 (cm²/sec)

ν the average kinematic viscosity of N₂, CH₄,
 CO, and O₂ with the value 0.1528 (at 20°)

O₂, CO, etc. are the constituents of the gas in percent
 by volume.

72. The Reynolds number is a criterion by which the
 flowing condition of an incompressible fluid is character-
 ized clearly. Further explanations are given in part B,
 section II.

The Reynolds number is used in this standard in the
 form:

$$Re_D = \frac{w \cdot D}{\nu}$$

in which

w is the velocity in the pipe line (m/sec)

D diameter of the pipe (m)

ν kinematic viscosity (m²/sec)

Accordingly, it is referred to the pipe diameter. Often
 it is referred to the diameter of the throttling device;
 then it is given the symbol Re_D .

73. Data Sheet 7 gives formulas and a nomographic
 chart for the easy determination of the Reynolds number
 Re_D . Table 4 gives the dynamic viscosities η and their
 changes with temperature for the technically important

¹⁹ See Mann, Gas- und Wasserfach, 1930, p. 570, and refer-
 ence 26.

gases. The nomographic chart of figure 29 gives η for steam, and ν for water and some other fluids. For gases, the calculating value $\nu' = \frac{\xi \eta}{n}$ is used as ordinate: this value, because of its independence of pressure, may be shown as a function of the temperature and is easy to use when the gas discharge is referred to normal conditions: it has no physical significance.

VII - PERMANENT PRESSURE LOSS

74. The permanent pressure loss is always less than the differential pressure. Figure IX shows the calculated pressure loss for the standard nozzle and for the standard orifice as a function of the ratio of areas m and expressed in percent of the differential pressure. The values of the figure are indicative only; so far as tests are available, they confirm these values for $m \leq 0.4$: For large values of m they no longer wholly agree, because the impact was not considered. Consequently, values of α' should at no time be derived from these data for the case when the downstream pressure is taken after the pressure recovery - at least $6D$ below the throttling device. For computing the pressure loss of orifices, the Faust rule applies - that is, expressed in percentage, as $(1 - m) \cdot 100$.

75. Figure 10 shows that the pressure losses of the standard nozzle and the standard orifice are practically equal if the ratio of orifices is so chosen that in both cases the same differential pressure corresponds to the same discharge.

B. THEORETICAL PRINCIPLES

I - BASIC EQUATIONS FOR INCOMPRESSIBLE FLUIDS

76. The symbols to be used in the following are:

F cross-sectional area in m^2

w average velocity in m/sec

P absolute pressure in kg/m^2

γ density in kg/m^3

g gravitational acceleration in m/sec^2

Q volume rate of flow in m^3/sec

G weight of flow in kg/sec

Ideal pipe flow will be assumed as a beginning. The Bernoulli equation supplies a relation between the pressure and the velocity at two cross-sectional areas. These must be so chosen that the lines of flow are parallel and the pressure accordingly constant over the whole area. Consideration of the picture of the flow in connection with the course of the pressure will show that this assumption is fulfilled in the pipe cross-sectional area I upstream and in the jet cross-sectional area II downstream from the throttling device. (See fig. XI.)

For these two areas the Bernoulli principle supplies the equation.

$$P_1' - P_2' = \frac{\gamma}{2g} (w_2'^2 - w_1'^2) \quad (V)$$

for incompressible fluids; it may also be used for approximate calculations of gases and vapors when the changes in pressure are small. A further relation supplies the continuity equation:

$$F_1 w_1 = F_2 w_2' \quad (VI)$$

The unknown cross-sectional area F_2 differs from the opening of the throttling device according to the shape of the throat (i.e., the sharpness or roundness of the edge k in figure XI). The contraction factor μ is introduced, accordingly; then $F_2 = \mu F_0$. Using the ratio of areas $m = F_0/F_1$, we have from equation VI:

$$w_1 = w_2' \mu m \quad (VII)$$

From (V) and (VII), we have a velocity w_2' in the area F_2 :

$$w_2' = \frac{1}{J-1 - \mu^2 m^2} \sqrt{\frac{2g}{\gamma} (P_1' - P_2')} \quad (VIII)$$

77. In practical measurement the more easily obtainable pressures P_1 and P_a are used instead of the pres-

tures P_1' and P_2' . Further deviations are brought about by the effects of friction and viscosity occurring under actual flowing conditions and which are expressed by a velocity distribution greatly different from the ideal in the upstream flow." By introducing a factor $\xi = \frac{w_a}{w_2'}$,

we collect these and the deviations due to the change in the positions of the pressure taps, and we obtain the volume rate of flow:

$$Q = \mu F_0 w_2 = \frac{\xi \mu}{\sqrt{1 - \mu^2 m^2}} F_0 \sqrt{\frac{2g}{\gamma} (P_1 - P_2)} \quad (\text{IX})$$

78. It is easy to see that the change of the upstream flow profile produced by surface friction and viscosity also changes the contraction factor. For example, the contraction is reduced by pointing the profile. Consequently ξ and μ are practically nonseparable. Therefore, the two factors are collected together with the factor that considers the finite upstream velocity to form the discharge coefficient:

$$\alpha = \frac{\xi \mu}{\sqrt{1 - \mu^2 m^2}} \quad (\text{X})$$

(By including $\frac{1}{\sqrt{1 - \mu^2 m^2}}$ α for $\mu \approx 1$ and large val-

ues of m may become greater than 1.3, as is the case with nozzles.)

The equations for the discharge in final form are:

$$Q = \alpha F_0 \sqrt{\frac{2g}{\gamma} (P_1 - P_2)} \quad (\text{m}^3/\text{sec}) \quad (\text{XI})$$

$$G = \alpha F_0 \sqrt{2g \gamma (P_1 - P_2)} \quad (\text{kg}/\text{sec}) \quad (\text{XII})$$

in which $G = Q \gamma$ is the weight of flow.

²⁰ More accurate investigations will be found in Kretzschmer's "Strömungsform und Durchflusszahl der Messdrosseln" (Form of Flow and Discharge Coefficients of Metering Throttles). Forschungsheft 381, 1937 VDI - Verlag.

II - REYNOLDS NUMBER

79. The Reynolds similarity law allows the metering results obtained with a definite material and throttling device to be carried over to other measurements of materials of different viscosity and density as well as to geometrically similar throttling devices as long as other characteristics of the material (compressibility, surface tension, heat conductivity) may be disregarded. If both geometrical and mechanical similarity prevail between two flow processes, the same discharge coefficient applies. Mechanical similarity is shown when the ratio of the forces of inertia to the forces of viscosity is the same in the cases under comparison.

Then

$$\frac{\gamma_x L_x w_x}{g \eta_x} = \frac{\gamma L w}{g \eta} = \frac{L w}{\nu} = Be \quad (\text{XIII})$$

In this

- L (m) are the characteristic lengths of the arrangement under comparison
- w (m/sec) the velocities
- η (kg sec/m²) the dynamic viscosities
- $\nu = \frac{\eta g}{\gamma}$ (m²/sec) the kinematic viscosities

80. The Reynolds number Be , according to its definition, is the ratio between two forces and is consequently a pure number. Any flow situation of any incompressible fluid is definitely characterized by this number. BY a small Reynolds number is meant one in which the forces of viscosity are large in contrast to the forces of inertia: a large Reynolds number is the opposite.

Research is consequently greatly simplified by the use of the similarity law because it is enough to change any value contained in the Reynolds number, such as the velocity, in order to obtain at the same time the effect of the other values contained in Be . The Reynolds number, therefore, serves as the abscissa for plotting the discharge coefficient as long as the density and the viscosity are the only effective characteristics of the fluid.

81. At sufficiently high values of the Reynolds number the discharge coefficients of the standard nozzles and standard orifices become constant, i.e., independent of the velocity, viscosity, and density of the flowing substance. Hence it is important to know the Reynolds numbers at which the independence stops and at which, consequently, the discharge coefficients become variable, whether as a result of a reduction in the velocity or in the diameter or of an increase in the viscosity. These Reynolds numbers are characterized in the Standards. as "tolerance limits."

82. The validity of the Reynolds similarity law has been demonstrated by tests for discharge measurements with nozzles and orifices when geometrical similarity is observed strictly. Included with geometrical similarity are the similarity of the installation, of the pressure taps, and of the condition of the surface of the pipe and the throttling device.

III - BASIC EQUATIONS FOR GASES AND VAPORS

83. With gases and vapors the change in density should, in general, not be neglected when the differential pressure is high and when high accuracy is demanded. If a change of state according to the adiabatic law

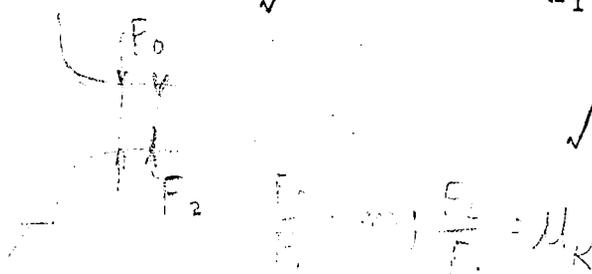
$$\frac{\gamma_2}{\gamma_1} = \left(\frac{P_2}{P_1} \right)^{1/\kappa}$$

is assumed in the contraction, the general energy equation (V) may be integrated, and with the help of the continuity equation for compressible fluids

$$w_1 F_1 \gamma_1 = w_2 F_2 \gamma_2 \quad (\text{XIV})$$

the velocity in the vena contracta is obtained:

$$w_2 = \frac{\zeta}{\sqrt{1 - \mu \kappa^2 m^2 \left(\frac{P_2}{P_1} \right)^{2/\kappa}}} \sqrt{\frac{2g P_1}{\gamma_1} \frac{\kappa}{\kappa - 1} \left[1 - \left(\frac{P_2}{P_1} \right)^{\frac{\kappa-1}{\kappa}} \right]} \quad (\text{XV})$$



The contraction factor is designated μ_k to distinguish it from the factor for incompressible fluids, since it is dependent on the pressure ratio P_2/P_1 for throttling devices with considerable contraction, particularly for orifices. This dependence results from the fact that an expansion is also possible in a lateral direction because the jet is not guided laterally. If the equation for the weight of flow

$$G = \mu_k F_0 w_2 \gamma_2 = \mu_k F_0 w_1 \gamma_1 \left(\frac{P_2}{P_1}\right)^{1/\kappa} \quad (\text{XVI})$$

is expanded with

$$\sqrt{1 - \mu_k^2 m^2}, \alpha \text{ and } \sqrt{P_1 - P_2}$$

there is obtained an equation similar in form to (XII):

$$G = \alpha \epsilon F_0 \sqrt{2g \gamma_1 (P_1 - P_2)} \quad (\text{kg/sec}) \quad (\text{XVII})$$

where, in

$$\epsilon = \frac{\alpha_k}{\alpha} \sqrt{\frac{1 - \mu_k^2 m^2}{1 - \mu_k^2 m^2 \left(\frac{P_2}{P_1}\right)^{2/\kappa}}} \sqrt{\frac{1}{1 - \frac{P_2}{P_1}} \frac{\kappa}{\kappa - 1} \left[\left(\frac{P_2}{P_1}\right)^{2/\kappa} - \left(\frac{P_2}{P_1}\right)^{\frac{\kappa+1}{\kappa}} \right]} \quad (\text{XVIII})$$

the total effect of the compressibility is included. In this formula

$$\alpha_k = \frac{\xi \mu_k}{\sqrt{1 - \mu_k^2 m^2}} \quad (\text{See paragraphs 76 to 78.})$$

In the same way the volume rate of flow referred to conditions upstream from the metering location is found to be

$$Q = \alpha \epsilon F_0 \sqrt{\frac{2g}{\gamma_1} (P_1 - P_2)} \quad (\text{m}^3/\text{sec}) \quad (\text{XIX})$$

84. These developed equations apply only as long as the velocity of sound is not reached.

85. For nozzles with a contraction factor $\mu \approx \mu_k \approx 1$

the radial expansion may in general be neglected; so that $\frac{\alpha}{a} = 1$. The good agreement of the discharge calculated from equation (XVII), neglecting the radial expansion with that obtained experimentally, shows that the expansion correction ϵ calculated above is valid for nozzles.

86. When the contraction is considerable, particularly with orifices, the effect of lateral expansion on the contraction factor may be brought out by an additional expression (see references 1, 2, 14, and 22), but only with simplifying assumptions. In this case, therefore, it is better from the practical standpoint to substitute the empirically determined expansion factor ϵ in equations (XVII) and (XIX).

IV - EFFECT OF MOISTURE IN GASES

87. The density γ_f of a moist gas is equal to the sum of the density of its constituents in the form of dry gas γ_{tr} and of water vapor γ_D . It should be considered that these constituents are under only partial pressures corresponding to their proportional volumes. These proportional volumes are usually not given, but generally only the relative humidity φ of the gas mixture is known; this is defined as the ratio of the weight of vapor actually contained in a unit volume to the weight of vapor that would be contained in the saturated gas at the same temperature.

88. The method of calculating the density of the gas mixture is as follows: The density γ_D of the saturated water vapor at the measurement temperature is taken from a steam table and multiplied by the relative humidity φ . If the density γ_n of the dry gas at standard conditions is given, this is to be converted with the help of the temperature and the ratio of the standard pressure to the partial pressure at measurement conditions. The partial pressure is equal to the total pressure minus the partial pressure of the water vapor. The latter may be taken from a steam table as a function of the saturation temperature and is φP_D .

Then

$$\gamma_f = \gamma_{tr} + \varphi \gamma_D \quad (\text{kg/m}^3) \quad (\text{XX})$$

89. In converting the gas mixture to another set of conditions, the water vapor may generally be treated as an ideal gas as long as it remains superheated; see table 3, Data Sheet 7. Only near the saturated condition does the water vapor cease to act as an ideal gas; when high accuracy is required, it is recommended, accordingly, that the conversion be undertaken on the basis of the known steam tables.

90. In converting a measured volume of gas to standard conditions, it should be borne in mind that the water vapor would be almost completely precipitated in the liquid form. Usually the standard volume of a moist gas is so defined that only the dry part of the gas mixture is converted to standard conditions. In order to perform this conversion, the dry part of the measured gas mixture must be calculated. However, the true density of the mixture is always to be substituted in the equations for discharge measurement, since this true density is a factor of the measurement. With saturated gas at approximately atmospheric pressure, the density correction for the moisture content may be taken directly from figure 18, Data Sheet 3.

C. SUPPLEMENTARY INFORMATION ON THE STANDARDS

I - EFFECT OF DIFFERENCES FROM THE STANDARD INSTALLATION

. (See also section A III)

91. With nonstandard pressure taps the discharge coefficients of section AV no longer apply. Figures XII and XIII show the changes of the static pressure at the pipe wall, with a diameter of the pressure tap equal to 0.01D, as functions of the distance from the faces of the nozzle or orifice. The ordinate is the difference between the static pressure measured at the pipe wall for the given distance and that at the faces for standard pressure taps, expressed in percent of the differential pressure.

92. Because of the quadratic relation between differential pressure and discharge, the deviations of the discharge coefficient are only half as great as the deviations given in figures XII and XIII. For the special case of equal spacing of the pressure taps upstream and downstream from the throttling device, the correction factors are given in figures 31 and 32, Data Sheet 8; to take account of this effect, multiply the discharge coefficient by the correction factor.

93. Nonstandard taps as in sections 91 and 92 reduce the metering accuracy. The following additional tolerances are to be added to the other tolerances given on Data Sheet 5.

± 0.5 percent for $m \begin{cases} \leq 0.4 & \text{for nozzles} \\ \leq 0.5 & \text{for orifices} \end{cases}$

± 1.0 percent for higher values of m

94. With nonstandard installations, i.e., if the requirements prescribed in section A III are not fulfilled, the error in measurement increases. The principal reasons are:

1. Noncircular pipe.
2. Nonaccurately centered installation of the throttling device or the gasket.
3. Diameter of the upstream mounting ring smaller than that of the pipe.
4. Conical expansion of the pipe, especially with rolled-on flanges.
5. Difference between the nominal diameter of the pipe used in the calculation and the actual diameter.

95. For considering some of these differences, the following essential data concerning their effect on the discharge coefficient are given, insofar as reliable test results are available.

a) Diameter of the Mounting Ring
Smaller than the Pipe Diameter

Figure XIV shows the installation; figures XV and XVI give the correction factors for the discharge coefficients of nozzles and orifices, as functions of a ratio of areas m^2 . This is referred to the area of the mounting ring instead of, as usual, to the area of the pipe. This sort

²¹For nozzles according to Witte, Forschung 5, 1934, p. 205, and unpublished tests of G. Ruppel, Berlin, (to appear in Forschung 6, no. 5, 1935). For orifices according to Ruppel and Jordan, Forschung 2, 1931, p. 207, and Kretzschmer and Wälzholz, Forschung 5, 1934, p. 28, and Witte, Forschung Gebiete Ingenieurewesen 5, 1934, p. 205.

of plotting is chosen since in practice often the diameter of the mounting ring D' is known and the, true pipe diameter D can only be estimated. Metering errors are due, on the one hand, to the hydrodynamic disturbance produced by the upstream edge of the mounting ring and, on the other hand, to the incorrect calculation of the upstream velocity which is referred to the diameter D' instead of to D . Since the two effects enter the result with different signs, the correction factors for a are sometimes larger and sometimes smaller than 1.

b) Gaskets Cut too Small

Figure XVII shows the installation, and figures XVIII and XIX show the correction factors for a , which, however, can give only an approximate idea of the order of magnitude of the resultant errors. Of course, it is always imperative to be certain and to take care that the gasket does not project; see the precautions in paragraph 25.

c) Disturbances Due to Expanding Pieces and Riveted Flanges with Standard Orifices

Figure 34, Data Sheet 8, gives the order of magnitude of the error to be expected with some installations^{2a} compared with the standard installation.

II - THROTTLING DEVICES AT THE INLET AND OUTLET OF PIPES

96. For the case when the material to be measured flows out of or into a space filled with the same material, say air, measurements with the standard orifice and the standard nozzle at the inlet and outlet of a pipe line are applicable. (See reference 10.) The differential pressure here is the difference between the pressure in the free space (usually atmospheric pressure) and the pressure at a standard pressure tap inside the pipe. The pressure taken in the pipe is the "negative pressure" when measuring at the inlet of a pipe, and the "positive pressure" when measuring at the outlet of a pipe.

If another material is in the space upstream or downstream from the throttling device and if at least one of the materials is a liquid, the surface tension has an effect which has not yet been completely investigated. (See

^{2a} According to tests of Ruppel and Jordan, *Forschung* 2, 1931, p. 207, and R. Witte, *Forschung* 5, 1934, p. 205.

reference 10:) For this reason the standard nozzle, particularly, is unsuitable for **discharging** liquids into air or gas-filled spaces.

97. When measuring at the inlet of a pipe, the **diameter** of the front surface of the throttling device should be at least 1.5 times the diameter of the opening; also, it should be smooth and free of holding arrangements, screws, and the like. When measuring at the inlet of a pipe, there should be no objects, walls, etc., that might disturb the entering flow within a distance of at least 20 d from the throttling device, and at least 10 d laterally from the axis of the throttling device.

When measuring at the outlet of a pipe, these distances may be reduced one-half.

98. At the inlet, the discharge coefficient above the tolerance limit, which lies for both at $Re_D = 55,000$, has a value of

$$\alpha_e = 0.99 \quad \text{for the standard nozzle,}$$

$$\alpha_e = 0.60 \quad \text{for the standard orifice.}$$

99. The basic tolerance of the discharge coefficients **above** the tolerance limit is

$$\pm 1 \text{ percent} \quad \text{for the standard nozzle,}$$

$$\pm 1.5 \text{ percent} \quad \text{for the standard orifice.}$$

Lack of sharpness of the edge is to be considered, if necessary, through the additional tolerance of Data Sheet 5.

100. At the outlet, the tolerance limits of table I (see paragraph 45) apply. The discharge coefficients above these limits for the corresponding **ratios** of area m' are:

For standard orifices, equal to those of figure 22, Data Sheet 5,

For Standard nozzles, as given in figure 21, Data Sheet 5, but reduced by 0.5 percent.

101. The basic tolerance for the discharge coefficients above the tolerance limits are:

± 1 percent for the standard nozzle,
 ± 1.5 percent for the standard orifice.

Pipe roughness and lack of edge sharpness are to be considered, if necessary, in accordance with Data Sheet 5.

102. Below the tolerance limit there are no reliable values for nozzles and orifices located at the inlet and outlet of pipes. Measurements in this range are consequently to be conducted only after special tests.

III - THROTTLE DEVICES AT RATIOS OF PRESSURES ABOVE THE CRITICAL

103. Measurements at pressure ratios above the critical may be performed when measuring flows of gases and vapors if considerable expansion is permissible in the throttling device. This may be the case, for instance, when expanding into a vacuum in acceptance tests of vacuum pumps or in acceptance tests of steam reaction turbines. With this method, it is sufficient to measure the pressure and temperature of the flowing medium upstream from the throttling device; measurement of the differential pressure is unnecessary.

To be sure, the process requires that the initial pressure must be changed in changing the discharge. For this reason and because of the high pressure loss in general, the method is not satisfactory for continuous measurement.

104. The method assumes that the velocity of flow in the vena contracta is equal to the velocity of sound under the conditions at that point. This is the case for a definite "critical" pressure ratio $(P_2/P_1)_{kr}$, which depends on μ , κ , and n . With the pressure difference increasing, i.e., with the back pressures becoming smaller, the discharge through the throttling device does not increase, because the pressure in the vena contracts is no longer affected by the back pressure P_a . From equations (XI) and (XIII), leaving out $(P_1 - P_a)$, we have as an equation for the flow

$$G = \alpha_{\kappa} F_0 \psi \sqrt{P_1 \gamma_1} \quad (\text{kg/sec}) \quad (\text{XXI})$$

in which

$$\psi = \sqrt{\frac{1 - m^2 \mu_{\kappa}^2}{1 - m^2 \mu_{\kappa}^2 (P_2/P_1)^{2/\kappa}}} \sqrt{2g \frac{\kappa-1}{\kappa} \left[\left(\frac{P_2}{P_1}\right)^{2/\kappa} - \left(\frac{P_2}{P_1}\right)^{\frac{\kappa+1}{\kappa}} \right]} \quad (\text{XXII})$$

From part B, section II, equation (XVIII), it is found that

$$\frac{\psi}{\sqrt{2g}} = \epsilon \sqrt{1 - \frac{P_2}{P_1}}. \quad \text{The value of } \psi \text{ attains its maximum value at the critical pressure ratio and then remains constant.}$$

105. With noticeable contraction of the jet in the throttling device, the critical pressure ratio is first reached when μ_{κ} becomes equal to unity, because of the increase of μ_{κ} with decreasing values of P_2/P_1 . With the standard orifice, particularly, no critical pressure ratio at all seems to be attained, as tests of Schiller (see reference 18) have shown; see also figure XX. The standard orifice is consequently not suitable for this method.

106. Table III shows, as far as is known, the critical pressure ratios as functions of m and κ for well-rounded nozzles without contraction: thus for $\mu = \mu_{\kappa} \approx 1$, particularly for the standard nozzle.

TABLE III. The Critical Pressure Ratio for the Standard Nozzle as a Function of m and κ

| κ | Area ratio, $d^2/d^2 =$ | | | |
|----------|-------------------------|-------|-------|-------|
| | m | | | |
| | .0 | 0.2 | 0.4 | 0.6 |
| 1.31 | 0.544 | 0.549 | 0.565 | 0.598 |
| 1.41 | .527 | .532 | .548 | .581 |

The equation

$$G = \frac{\alpha \psi}{\sqrt{2g}} 1.2522 d^2 \sqrt{P_1 \gamma_1} \quad (\text{kg/hr}) \quad (\text{XXIII})$$

using the nomenclature of the Standard, applies for $P_2/P_1 = (P_2/P_1)_{kr}$. In this $\alpha\psi/\sqrt{2g}$, as a dimensionless factor, is to be taken from figure XX, and d is to be substituted in millimeter's; p_1 in kg/cm^2 as absolute pressure, and γ_1 in kg/m^3 . In case the product $p_1 v_1$ is available in tables, as in steam tables, it is easier to write

$$G = \frac{\alpha\psi}{\sqrt{2g}} 1.2522 d^2 \frac{p_1}{\sqrt{p_1 v_1}} \quad (\text{XXIV})$$

Since the measured values may be stated very exactly, transition to other values of κ is permissible without further details. According to available measurements with standard nozzles for $\kappa = 1.31$, $\alpha\psi/\sqrt{2g}$ amounts to about 0.468, and for $\kappa = 1.4$, it becomes about 0.481 for $m \leq 0.4$. For higher values of m , the value seems to rise somewhat.

IV- THROTTLING DEVICES WITH CONSTANT DISCHARGE COEFFICIENTS AT LOW VALUES OF THE REYNOLDS NUMBER

107. The discharge coefficients of the standard nozzle and the standard orifice chance with the Reynolds number below the tolerance limit and, on the other hand, measurement at very small Reynolds numbers has practical significance, particularly with oil measurements, when measuring gas with high hydrogen content or very hot, and with very small diameters. Accordingly, various tests have been made to develop special forms of throttling devices that have constant discharge coefficients even at low values of the Reynolds number.

108. These tests were started with the observation that below the tolerance limit of the Standard the discharge coefficient of the standard nozzle decreases as the Reynolds number becomes smaller (fig. III), which apparently is to be explained by the increased effect of fluid friction; on the contrary the discharge coefficient of the standard orifice increases (fig. IV) because the constriction of the jet below the orifice becomes less. If now it were possible to develop a throttling device in which the con-

trary effects of fluid friction, and jet constriction are mutually compensated, then constant discharge coefficients might be expected from such a throttling device at smaller Reynolds numbers than from the standard nozzle and the standard orifice. The results of these tests have shown this method to be workable. A standard form for any ratio of areas, however, has not been found.

109. Of the throttling devices investigated in this connection, the following forms have been found to be particularly noteworthy.

| | |
|---|---------------------|
| Standard nozzle without a cylindrical part, | figure XXI |
| Doublo-bavoled orifice, | figure XXII |
| Double-rounded orifice, | figure XXIII |
| Double orifice | Bibliography 20a |

110. These throttling devices appear to be suitable, according to the tests that have been performed,²³ for the following ranges:

| Throttling device as in | Re_D | and for m approximately |
|-------------------------|------------------|---------------------------|
| Figure XXI | 4,000 to 100,000 | 0.3 to 0.4 |
| Figure XXII | 3,000 to 100,000 | 0.16 to 0.25 |
| Figure XXIII | 3,000 to 100,000 | < 0.2 |

Because of the present state of knowledge of the subject, a calibration of each of the throttling devices in accordance with paragraph 109 is necessary at the same Reynolds numbers as that at which it will be used. Besides, according to previous experience, disturbing factors in the installation or in the upstream section probably have greater effect at small Reynolds numbers than at higher Reynolds numbers. Consequently, it is recommended that care be taken to provide throttling devices for small Reynolds numbers with long lengths of disturbance-free pipe and careful installations, as well as that, for accurate measurements, the throttling device be calibrated under the same installation conditions as those with which it is to be used.

²³ According to unpublished test results of R. Witte; also for throttling devices like fig. XXI (see reference 23); for throttling devices like fig. XXII (see references 5 and 23); for throttling devices like fig. XXIII (see references 4 and 12).

V - GENERAL INFORMATION ON TEE CONNECTING LINES
AND THE DIFFERENTIAL PRESSURE GAGE

a) General

111. For acceptance and laboratory tests, a one- or two-legged U-tube is used for the differential pressure gage. If the differential pressure fluctuates, the only gage to be recommended is the single-legged U-tube, the second leg of which is replaced by a container of preferably a hundred times greater size. Direct reading of a scale behind the glass tube is accurate only when the differential pressure is quiet. A slide may be used to observe the meniscus without parallax and then the reading may be made later on a scale at the side.

112. The glass tubes should be at least 8 mm in diameter when mercury is used, and at least 15 mm in diameter when water or carbon tetrachloride is used, in order to avoid errors due to capillarity. For readings smaller than 50 mm, U-tubes are to be avoided. Small mercury readings in steam or water measurements may be amplified by using some other sealing fluid. A useful one is acetylene tetrabromide. Its density is approximately 3 kg/liter. However, when used for longer than a week, many disagreeable characteristics develop and make clean work difficult. Carbon tetrachloride is more pleasant and, at the same time, gives an amplified reading. Table IV gives the relation of its density to temperature.

TABLE IV. Density of Carbon Tetrachloride

| Temperature, °C | 15 | 20 | 25 |
|-------------------|-------|-------|-------|
| Density, kg/liter | 1.605 | 1.595 | 1.585 |

113. Inverted two-legged U-tubes filled with air or benzol over the water, may also be used.

114. Special micromanometers have been developed for measuring very small differential pressures of Oases; among them are the Askania-Minimeter and the Debro-Miniskop. These two devices are, suitable for testing service differential pressure gages (ring balances, etc.). Single-legged

slope-gage micromanometers are **very** difficult to use accurately and require very special care.

115. In choosing the most suitable differential pressure meter, several matters must be considered from the technical aspect of metering. The amount of the line pressure and the differential pressure are the chief factors determining the type of the differential pressure meter - thus; whether a float manometer, ring balance, bell float, etc., is to be chosen. The ratio of areas **giving** the **largest** differential pressure for the throttling device under the *given* service conditions is to be chosen. If a reduction of the effect of expansion is desired, low differential pressure and so, **large** ratio of areas, *are* to be used; on the other hand, it is recommended that small ratios of areas and so, large differential pressures, be chosen since, under these conditions, the tolerances for the discharge coefficient and the errors due to installation disturbances are smaller, and also the actuating force for the differential pressure meter is increased.

b) Differential Pressure Meter for Continuous Service

116. Differential pressure meters, both **indicating** and recording, usually have an arrangement for extracting the square root and therefore have equal divisions for the **discharge**, at least over the greater part of the scale. Since the differential pressure decreases as the square of the discharge, standardization near the zero point is uncertain. Important measurement should therefore be made with a sufficiently high differential pressure. For low flows - discharges of less than 10 to 20 percent of the maximum discharge, according to the design - sufficient accuracy cannot be guaranteed.

117. If the scale of the differential pressure meter is divided into units of the discharge, it can apply only for a definite state of the flowing material, termed the "base conditions." The actual state of the material, termed the "measurement conditions," is usually more or less different. The difference between base and measurement conditions may be considered with gases and liquids, by multiplying the volume rate of flow read at the apparatus by the value $\sqrt{\gamma_r/\gamma_b}$ and with steam and water by multiplying the **weight** of flow by the value $\sqrt{\gamma_b/\gamma_r}$. Here

γ_b is the density under measurement conditions,

γ_r density under base conditions.

If necessary, changes in the density of the **sealing** fluid and of the substance ~~over the sealing fluid~~ in the connecting lines is **also** to be considered.

118. For very accurate continuous measurement, **it is** also to be considered **that the** expansion factor ϵ **changes with** the differential pressure, **while** the calculations of the differential pressure meter are **usually** carried out considering ϵ as constant. An average value ϵ_m is used by most manufacturers in calculating the scale, **corresponding** to $2/3$ of the normal discharge or $4/9$ of the normal differential pressure.

119. The **following** discussion of the different **types** of apparatus and their evaluation refers only to the theory and practice. The quality and serviceability of the apparatus to a **large degree** are dependent on the care exercised in its manufacture and **especially**, on the perfection of its structural form, **With** regard to the latter, it must be reckoned that in all fine **apparatus** bearings the friction error increases more than linearly with the dead weight of the parts moving in the bearing. **Consequently**, with respect to this friction error, the finer, more elegantly built apparatus is **generally** preferable to the more **ruggedly** built apparatus, from the standpoint of **metering technology**. Extracting the square root, **especially**, places very **high** requirements on the **design** and manufacture. Its accuracy is very considerably limited insofar as the bearings **spacing** is maintained constant with respect to other operating parts in the structural **design**.

120. **With** float manometers, extracting the square root is done either by a **corresponding** formation of the chamber in one leg of a U-tube, or by mechanical methods, i.e., by a cam or electrically. Extracting the square root by means of the chamber shape is theoretically possible only down to a minimum differential reading, which is **loner**, the greater the ratio of the differential pressure to the float travel. **Consequently**, and because of certain structural **advantages** occurring incidentally, float manometers extracting the square root by the chamber shape are useful particularly for steam and water meters, for which a sufficiently **high** differential pressure is available.

Float manometers **with** cylindrical chambers, in which the square root is extracted by a cam, have the same range of use. **With** these, extracting the square root is theoretically **possible** at small ratios of the differential

pressure to the stroke nearly down to the zero point but, practically, because of unavoidable friction errors and because of the very considerable translation ratio necessary at the zero level, it is possible only down to a point that is dependent on the design and workmanship. Note should be taken that, because of the quadratic law, when the discharge is a tenth of the maximum, the differential pressure has only a hundredth part of its maximum value.

A disadvantage of all float manometers is the necessity of transferring the float movement outside the pressure space. This is done through stuffing boxes or by magnetic couplings,

121. With ring balances, extracting the square root is theoretically possible down to the zero point. They have a very high internal actuating force. This characteristic makes them particularly suitable for low differential pressures. For differential pressures above 3 meters of water, the size is inconveniently large. In spite of this, they are often used in this range in the chemical industry because of their good characteristics. Other advantageous characteristics are that, in contrast to most other differential pressure meters, their reading is independent of the specific gravity and amount of the sealing fluid when the metering drum is made symmetrically with respect to the axis, and that the range of differential pressure (end readings of the scale) may be changed over a wide range by changing the sensitivity.

122. Extracting the square root with the ring balance is done either by means of the force or the travel. In extracting the square root by means of the force, the external actuating force is made to increase as the square of the differential pressure and therefore proportionally to the discharge." In extracting the square root by means of the travel, on the contrary, only the indicator movement is proportional to the discharge, while the movement of the balance ring goes approximately linearly with the differential pressure.

The first method has the advantage that near the zero point there is a relatively large angle of rotation of the ring balance and so a high capability for doing work. In practice, however, there is a considerable delay in coming to rest, so that, with low differential pressures, the second method is usually preferred.

123. Bell-float devices have the advantage that they deliver a great deal of power, and-because of this and their simple design they are very sturdy and reliable. They are therefore well suited for rough service conditions and for unusual services. If the part of the bell immersed in the sealing fluid **has** practically no volume, changes in density and, within certain limits, the quantity of **sealing** fluid have no effect on the measurement. With these metering devices, the external **actuating** force is delivered through **countersprings** or **weights** to a movable lever arm. If, on the other hand, the buoyancy of the bell-float volume varies **with** the stroke and is responsible for the external actuating force, **changes** in the density and the quantity of sealing fluid enter into the measurement.

124. The diaphragm differential pressure meter must operate **with** a transfer arrangement that operates **with** a high ratio, on account of the diaphragm stroke amounting to only a few millimeters. They are especially important where it is necessary to **operate** without a fluid, i.e., in ship service, or where mercury must not be used for reasons of health or corrosion.

125. In addition, there is a large number of methods in which the differential pressure varies same electrical quantity directly without using any intermediate **mechanical** members: for example, mercury closes contacts or **varies the** flow of current to points on the secondary of a transformer, or a magnetic or inductive resistance is changed by means of a suitable float, etc.

126. In the commercial differential pressure meters, the discharge is usually indicated on a scale or recorded on a chart. If the recording pen moves on the chart **proportionally** to the discharge, the chart may be **planimetered** and so the total quantity discharged in a given period of time may be determined. Usually the charts cannot be **planimetered** down to the zero line. In this case the true zero line does **not** agree with the so-called **planimeter** zero line, which is obtained theoretically **through extrapolation** of the **proportional divisions**.

127. Additional apparatus may be installed in or added to many differential pressure meters for electric telemetering or to release control impulses or, which is most important, to record the total flow. The resistors, of which several **mechanical** and electrical constructions are available, **integrate** the discharge against time. If the density

of the material being metered changes, it may be desirable to **use automatic** correcting arrangements so that the discharge (weight or volume) of the flowing material will be registered accurately with its changes in density.

c) Connections-to **the** Differential Pressure Meter

128. In measuring noncompressible fluids, the differential pressure meter should, if possible, be located **below** the throttling device and, in case of steam measurement, below the condensing chambers; this is in order that air separating in the differential pressure lines will not collect in the differential pressure meter but may escape into the pipe line. If the static pressure in the pipe line is small, the differential pressure is to be set so far below the throttling device that a small excess pressure is always acting on the low pressure side of the differential pressure meter. If the differential pressure meter is located above the throttling device, provision is to be made for removing the gas through a gas trap at the highest point of the differential pressure line.

129. In measuring gases, the differential pressure meter is to be set as high as possible above the throttling device, **so that** liquids condensing in the differential pressure lines may flow back to the pipe line. If the pressure in the pipe line is below atmospheric, special **care is** to be taken to see that the differential pressure lines and their connections are tight. If the differential pressure meter must be located below the throttling device, it is recommended that the differential pressure lines be led to a point below the throttling device and the differential pressure meter, and that a separatory chamber for condensates of water, tar, etc., be provided at that point.

130. The differential pressure lines should be at least 9 mm in inside diameter and should be as short as possible. In measuring steam or hot fluids or hot gases, they must be so arranged that no hot fluids can enter the differential pressure meter. The differential pressure lines should **always** be laid with a gradient (at least 1:10, preferably less than 45°). Collection of water in gas measurement and collection of air in liquid measurement are to be avoided by proper laying of the pipes or by the installation of suitable separators. Sharp bends are not permissible. If the differential pressure lines contain water, care must be taken to lay them where they will not

freeze. Figure XXIV shows an arrangement of double pressure taps independent of each other, that may be recommended for steam measurement.

VI - APPLICATION TO ACCEPTANCE STANDARDS IN STEAM AND CONDENSATE MEASUREMENT

a) Tolerances for Reading the Differential Pressure $P_1 - P_2$

131. The measurement of $P_1 - P_a$, requires the greatest care. For the requirements of the differential pressure meter, see paragraphs 61 and 111 ff.

132. The connecting lines to the differential pressure meter must be tested for tightness, including any connecting lines there may be between the two sides of the U-tube. The lines must be filled with a definitely known substance, which is insured by repeated vigorous blowing-out of the lines. (See also paragraph 63, as well as paragraphs 128 to 130.)

133. For acceptance tests, two independent differential pressure meters are to be provided for reasons of certainty, each with its own lines and with condensate chambers for the upper water levels in each of the two lines, and no test should be begun before the two readings agree. Figure XXIV shows a proven arrangement.- When condensate chambers with overflows are used, great care should be taken that the overflow lines to the steam pipe have no constrictions, in the form of either welded seams or projecting gaskets, that they be true, i.e., without sudden changes, and that they be as short as possible and insulated. In condensate chambers with horizontal steam inlets, the side taps and the connecting lines must lie in the same horizontal plane.

Before starting the test, accordingly, care must be taken that the chambers are filled with condensate or water, when necessary, the chambers are to be cooled.

134. The zero point is to be checked before and after the test.

$P_1 - P_2$ should be read directly from the differen-

tial pressure meters. The smallest metering area should be so chosen that the displacement of the sealing fluid is more than approximately 75 mm,

135. The average of the two differential pressure meters is to be used in the calculations. If the agreement of the two differential pressure meters is within a percent, an additional tolerance of $\pm a/4$, but not less than ± 0.5 percent, is to be applied to $\sqrt{P_1 - P_a}$ and thus to the discharge in order to take care of the possibility of error of the differential pressure meter.

136. If the manometer reading changes slowly by more than 5 percent, the average value is not to be taken of the readings but of the square root of the readings.

Short-period fluctuations of the order of ± 3 percent may be taken into account by means of frequent readings.

b) Tolerance for γ_1

137. The density under the conditions upstream from the throttling device must be measured with the same care as the differential pressure. Consequently, there must be known: for water measurements, the temperature; for steam measurements, the temperature, the pressure, and the relation of the density to the pressure and temperature. (See also paragraph 64 ff.)

138. The temperature should not be measured too far from the meter (see paragraph 65), observing, however, the installation prescriptions; thermometers of known accuracy are to be used. It is recommended that a second thermometer of another type be used in a different location.

139. Pressures up to 3 atm abs are preferably read with mercury; above that Bourdon tube manometers of known accuracy are to be used. Care is to be taken with the pressure tap that the bore is flush with the wall and has no burr. With high values of m end large differential pressures, the pressure P_1 is to be read in the plane of the differential pressure tap. (See paragraph 65.)

140. The absolute pressure and the absolute temperature ($273 + t^{\circ} \text{C}$) can be determined with careful manipulation to ± 1 percent, and 0.5 percent, respectively. The

dependence of the density on pressure and temperature in the case of steam is known (see references 7, 11, and 19) within ± 0.7 percent up to about 50 atm abs. The total tolerance for γ_1 is determined by the law for combining errors to be $\sqrt{1 + 0.25 + 0.49} = 1.4$ percent approximately. In determining the tolerance for the discharge, it is to be figured at half-rate, or ± 0.7 percent, since $\sqrt{\gamma_1}$ enters the matter. For separate tolerances other than those given above, the total tolerance for the density must be corrected accordingly.

c) Total Tolerance

141. The total tolerance is to be determined from the individual tolerances for α , ϵ , $\sqrt{\gamma_1}$, and $\sqrt{P_1 - P_2}$ by the law for combining error. Two examples are given:

a. In measuring water with a basic tolerance of ± 0.5 percent for α , an additional tolerance of ± 0.5 percent for pipe roughness, of ± 0.1 percent for $\sqrt{\gamma_1}$, and of ± 0.5 percent for $\sqrt{P_1 - P_2}$, the total tolerance is $\sqrt{0.25 + 0.25 + 0.01 + 0.25} = 0.9$ percent.

b. In measuring steam with a basic tolerance of ± 0.5 percent for α , an additional tolerance of ± 0.5 percent for pipe roughness, a tolerance of ± 0.5 percent for ϵ , of ± 0.7 percent for $\sqrt{\gamma_1}$, and of ± 0.5 percent for $\sqrt{P_1 - P_2}$, the total tolerance is $\sqrt{0.25 + 0.25 + 0.25 + 0.49 + 0.25} = \pm 1.2$ percent.

D. EXAMPLES

1. Calculation of the diameter of throttling device for given service conditions and a chosen differential pressure (examples 1 to 3).
2. Determining the discharge for a given throttling device (examples 4 and 5).

| | | | | | |
|---------|-------------|---|------------|-------------------|---------------------|
| Given: | Discharge | | | | |
| Sought: | Diameter of | { | Example 1, | Flowing material, | Water |
| | the throt- | " | " | " | " |
| | ting device | " | 2, | " | , Dry air |
| | | " | 3, | " | , Superheated steam |
| Given: | Metering | } | " | 4, | " |
| | Arrangement | | " | " | " |
| Sought: | Discharge | " | 5, | " | " |
| | | | | | , Moist air |

EXAMPLE 1

Flowing material: Water

Given:

| | |
|-------------------------------|---|
| Maximum discharge | $G = 40,000 \text{ kg/hr}$ |
| Measured line diameter | $D = 100 \text{ mm}$ |
| Maximum differential pressure | $h = 100 \text{ mm (Hg - H}_2\text{O)}$ |
| $\Delta P = P_1 - P_2$ | $= 1255 \text{ kg/m}^2$ |

(with mercury and water at a temperature of about 20° C)

Sought: Diameter of the throttling device.

| Item No. | | Nozzle | | Orifice | | |
|----------|---|-------------------|--------|---------|--------|--------|
| | | a | b | a | b | |
| 1 | Water temperature | °C | 15 | 100 | 15 | 100 |
| 2 | Density of water, γ_1 | kg/m ³ | 1000 | 958 | 1000 | 958 |
| 3 | m_a from paragraph 43(b) of the Standard; see also Data Sheet 4 | -- | 0.2845 | 0.2905 | 0.2845 | 0.2905 |
| 4 | λ from figs. 19 & 20, Data Sheet 4 | -- | 0.012 | 0.014 | 0.670 | 0.672 |
| 5 | $m = \frac{m_a}{m}$, (item B/item 4) | -- | 0.281 | 0.2865 | 0.425 | 0.432 |
| 6 | Reynolds number Re_D (from Data Sheet 7 at maximum flow) | -- | 23,000 | 88,000 | 23,000 | 88,000 |
| 7 | Reynolds number at the tolerance limit | -- | 10,000 | 10,000 | 30,000 | 30,000 |
| 8 | Discharge at which the tolerance limit is reached $G_T = G \frac{Re_T}{Re_D}$ | kg/hr | 35,800 | 9,000 | 42,200 | 10,700 |
| 9 | Correction factor for pipe roughness (for nozzles, fig. 23a, for orifices, fig. 24a, Data Sheet 5) | -- | -- | -- | 1.009 | 1.0095 |
| 10 | Correction factor for lack of edge sharpness (fig. 24b, Data Sheet 5) | -- | -- | -- | 1.008 | 1.0075 |
| 11 | Corrected value of u , a | -- | -- | -- | 0.681 | 0.683 |
| 12 | Recalculated value of m using the corrected value of u from $\frac{m_a}{a_1}$ | -- | -- | -- | 0.418 | 0.425 |
| 13 | Diameter of the throttling device $d = \sqrt{mD^2}$ (at measurement temperature) | mm | 53 | 53.5 | 64.6 | 65.2 |
| 14 | d at 200 (V2A steel) fig. 17, Data Sheet 3 | mm | 53 | 53.4 | 64.6 | 65.0 |
| 15 | Basic tolerance for a (above the tolerance limit) | percent | 0.5 | 0.5 | 0.5 | 0.5 |
| 16 | Additional tolerance for pipe roughness | percent | -- | -- | 1.2 | 1.2 |
| 17 | Additional tolerance for lack of edge sharpness | percent | -- | -- | 0.5 | 0.5 |
| 18 | Total tolerance for a $(\text{Item 15})^2 + (\text{Item 16})^2 + (\text{Item 17})^2$ | percent | 0.5 | 0.5 | 1.4 | 1.4 |
| 19 | Tolerance for $\sqrt{P_1 - P_2}$ (assumed reading on U-tube manometer) | percent | 0.5 | 0.5 | 0.5 | 0.5 |
| 20 | Tolerance for $\sqrt{\gamma}$ (assumed) | percent | 0 | 0 | 0 | 0 |
| 21 | Tolerance of the measurement $\sqrt{(\text{Item 18})^2 + (\text{Item 19})^2 + (\text{Item 20})^2}$ | percent | 0.7 | 0.7 | 1.5 | 1.5 |

EXAMPLE 2

Flowing material: -Air

| | | |
|---------|---|--|
| Given : | Maximum volume discharged at standard conditions | $Q = 1300 \text{ m}^3/\text{hr}^*$ |
| | Maximum differential pressure $\Delta P = P_1 - P_2$ (at 20°) | $h = 100 \text{ mm H}_2\text{O}$ $= 100 \text{ kg/m}^2$ |
| | Air temperature in the line | $t_1 = 60^\circ \text{ C}$ $T_1 = 333^\circ \text{ K}$ |
| | Measured line diameter | $D = 300 \text{ mm}$ |
| | Barometer reading at 10° corresponding to | $B_0 = 755 \text{ mm Hg}$ $P_0 = 10,265 \text{ kg/me}$ |
| | Gage pressure upstream from the throttling device | $P_1 = 450 \text{ mm H}_2\text{O}$ |
| | Pressure upstream from the throttling device | $P_1 = 10,715 \text{ kg/m}^2$ |

It is assumed that the differential pressure meter used has a scale divided into units of the discharge, so that it is calculated with a constant average value for ϵ . (See paragraph 113.)

Sought: Diameter of the throttling device

| Item No. | | | Nozzle | Orifice |
|----------|---|------------------------|--------|---------|
| 1 | Volume rate of flow | Std.m ³ /hr | 1300 | 1300 |
| 2 | Factor for recalculating to measurement conditions, $\sqrt{\frac{P_1 T_N}{P_N T_1}}$ ($P_N = 1033 \text{ kg/me}$; $T_N = 273^\circ \text{ K}$) (see equation (10), Data Sheet 3) | -- | 0.9220 | 0.9220 |
| 3 | Pressure ratio $\frac{P_1 - P_2}{P_1}$ | -- | 0.0093 | 0.0093 |
| 4 | ϵ for $m = 0.2$ (estimated from Data Sheet 6 for $(\frac{4}{9}) \frac{P_1 - P_2}{P_1}$) (see paragraph 111) | -- | 0.997 | 0.999 |
| 5 | $m\alpha$ from paragraph 43 (See also Data Sheet 4) | -- | 0.1430 | 0.1430 |
| 6 | α from figs. 19 and 20 (Data Sheet 4) | -- | 0.993 | 0.620 |
| 7 | $m = \frac{n\alpha}{c}$ (first approximation) | -- | 0.1440 | 0.2305 |

*See footnote on page

| Item No. | | | Nozzle | Orifice |
|------------|---|------------------|--------|---------|
| 8 | Reynolds number Re_D at maximum discharge (from Data Sheet 7) | -- | 92,000 | 92,000 |
| 9 | Reynolds number at the tolerance limit Re (paragraph 45) | -- | 75,000 | 55,000 |
| 10 | Discharge at which the tolerance limit is reached, $Q_T = Q \frac{Re_T}{Re_D}$ | Std. m^3/hr | 1,070 | 785 |
| 11 | Corrected value of ϵ for calculated value of m (from Data Sheet 6) | -- | 0.997 | 0.999 |
| 12 | Now m_a is to be recalculated: as compared with the first assumed value, there is no noticeable difference; a and m may be retained as in Items 6 and 7 | -- | 0.1430 | 0.1430 |
| 13 | Correction factor for pipe roughness, from Data Sheet 5 | -- | -- | 1.000 |
| 14 | Correction factor for lack of edge sharpness, from Data Sheet 5 | -- | -- | 1.002 |
| 15 | Correction factor for the effect of viscosity, from Data Sheet 5 | -- | -- | -- |
| 16 | Corrected value of a , a_1 | -- | 0.993 | 0.620 |
| 17 | Corrected value of m , m_1 | -- | 0.144 | 0.230 |
| 18 | $d = \sqrt{m_1 D^2}$ at temperature of measurement | mm | 113.E | 144.1 |
| 19 | d at 20° (aluminum) fig. 17, Data Sheet 3 | mm | 113.E | 143.8 |
| Tolerances | | | | |
| 20 | Basic tolerances for a , from Data Sheet 5 | percent | 0.5 | 0.5 |
| 21 | Tolerance for pipe roughness, from Data Sheet 5 | percent | -- | -- |
| 22 | Tolerance for lack of edge sharpness, from Data Sheet 5 | percent | -- | -- |
| 23 | Tolerance for a | percent | 0.5 | 0.5 |
| 24 | Tolerance for ϵ , from table II | percent | 0 | 0 |
| 25 | Tolerance for $\sqrt{P_1 - P_2}$ (from data of the manufacturer of the differential pressure meter, applicable in percent of the particular theoretical value in the guaranteed metering range) (estimated) | percent | 1.5 | 1.5 |
| 26 | Tolerance for $\sqrt{\gamma}$, (assumed) | percent | 0.2 | 0.2 |
| 27 | Total tolerance of the measurement, $\sqrt{(\text{Item 23})^2 + (\text{Item 24})^2 + (\text{Item 25})^2 + (\text{Item 26})^2}$ | percent | 1.6 | 1.6 |

*Standard conditions are 760 mm of mercury and 0° C; see Item 2. "11 r /hr" as in equation (10) refers to standard conditions of 1 atmosphere (1 $\frac{g}{m^2}$) and 20° C.

EXAMPLE 3

Flowing material: Superheated steam

Given: Maximum discharge $G = 40,000 \text{ kg/hr}$
 Maximum differential pressure $= 6 \text{ m H}_2\text{O} = 6988 \text{ kg/m}^2$
 $P_i = P_a$ (at 20°)
 Pipe line diameter $D = 200 \text{ mm}$
 Steam pressure upstream from the throttling device $P_1 = 12 \text{ atm abs}$
 $120,000 \text{ kg/m}^2$
 Steam temperature $t_1 = 360^\circ \text{ C}$
 Specific volume (from Knoblauch's Steam Table, 1932) $v_1 = 0.2432 \text{ m}^3/\text{kg}$
 Density $\gamma_1 = 4.112 \text{ kg/m}^3$

The differential pressure meter is a float manometer, with a scale divided into units of flow, so that it is calculated with a constant value of ϵ ; see paragraph 117.

Sought: Diameter of the throttling device

| Item No. | | | Nozzle | Orifice |
|----------|--|-------|-----------|-----------|
| 1 | Maximum discharge | kg/hr | 40,000 | 40,000 |
| 2 | Ratio of pressures $\frac{P_1 - P_2}{P_1}$ | -- | 0.06 | 0.06 |
| 3 | ϵ estimated for $m = 0.2$ and $\frac{\frac{4}{9}(P_1 - P_2)}{P_1} = 0.022$, from Data Sheet 6 | -- | 0.986 | 0.992 |
| 4 | m_1 from Data Sheet 4 (see also paragraph 43) | -- | 0.6168 | 0.5131 |
| 6 | m from fig. 19 and 20, Data Sheet 4 | -- | 1.074 | 0.778 |
| 6 | $m = \frac{m\alpha}{\alpha}$ | -- | 0.491 | 0.669 |
| 7 | $m_1 \alpha$ for ϵ_1 | -- | 0.5170 | 0.5140 |
| 8 | α_1 from figs. 19 and 20 (approximation sufficient) | -- | 1.074 | 0.779 |
| 9 | m_1 from $\frac{m\alpha}{\alpha_1}$ | -- | 0.482 | 0.660 |
| 10 | Reynolds number Re_p at maximum discharge* (Data Sheet 7), extrapolated | -- | 3,000,000 | 3,000,000 |
| 11 | Reynolds number Re_p at the tolerance limit | -- | 200,000 | 270,000 |
| 12 | Correction factor for pipe roughness from Data Sheet 5 | -- | 1.002 | 1.006 |
| 13 | Correction factor for lack of edge sharpness | -- | -- | -- |

*In industrial steam measurement, the Reynolds number generally lies so high that a trial to see whether it lies above the tolerance limit is not necessary.

| Item No. | | | Nozzle | Orifice |
|------------|--|---------|--------|---------|
| 14 | Corrected value of α, α_2 | -- | 1.076 | 0.783 |
| 15 | Final value of m, m_2 | -- | 0.481 | 0.657 |
| 16 | $d = \sqrt{m_2 D^2}$ at temperature of measurement | mm | 138.9 | 162.1 |
| 17 | d at 20° C (V2A steel) | mm | 137.8 | 161.0 |
| Tolerances | | | | |
| 18 | Basic tolerance for d_L , from Data Sheet 5 | percent | 0.5 | 0.8 |
| 19 | Tolerance for pipe roughness | percent | 0.5 | 1.5 |
| 20 | Tolerance for lack of edge sharpness | percent | -- | -- |
| 21 | Total tolerance for a | percent | 0.7 | 1.7 |
| 22 | Tolerance for ϵ , from table II | percent | 0 | 1.5 |
| 23 | Tolerance for $\sqrt{P_1 - P_2}$ (from data of the manufacturer, applicable in the metering range) | percent | 1.0 | 1.0 |
| 24 | Tolerance for $\sqrt{\gamma}$ | percent | 0.7 | 0.7 |
| 25 | Total tolerance of the measurement | | | |
| | $\sqrt{(\text{Item 21})^2 + (\text{Item 22})^2 + (\text{Item 23})^2 + (\text{Item 24})^2}$ | percent | 1.4 | 2.58 |

EXAMPLE 4

Flowing material: Superheated steam

Given: Standard nozzle $d = 40$ mm at 20° C
 Material, cast iron
 Measured inside pipe diameter $D = 100$ mm
 Ratio of areas $\frac{d^2}{D^2} = m = 0.16$
 Installation Standard
 Gage pressure of steam 40.5 atm
 Steam temperature 400°

Sought: Weight of flow G kg/hr at a differential pressure of 525 mm
 ($H_g - H_2O$)

| Item No. | Metering and Calculation Results | | Nozzle |
|------------|--|--|--------|
| 1 | Gage pressure of the steam upstream from the nozzle (corrected according to the manometer calibration curve and considering the column of water in the manometer line) | atm | 40.5 |
| 2 | External air pressure P_0 at 0° C | mm Hg | 767 |
| 3 | Absolute pressure of the steam upstream from the nozzle P_1 | atm abs | 1.043 |
| 4 | Temperature of the steam upstream from the nozzle t_1 | $^\circ$ C | 41.54 |
| 5 | Density of the steam upstream from the nozzle γ_1 (from Knoblauch's Steam Table, 1932) | kg/m ³ | 400 |
| 6 | Differential pressure (measured at 22° C) h | mm | 13.94 |
| 7 | Differential pressure $P_1 - P_2 = h(\gamma_{H_2O}' - \gamma')$ (see equation (3), Data Sheet 3) | (Hg-H ₂ O) kg/m ² | 525 |
| 8 | Ratio of pressures $\frac{P_1 - P_2}{P_1} = \frac{6580}{415\ 430}$ | --- | 6580 |
| 9 | Expansion factor ϵ , from fig. 27, Data Sheet 6 | --- | 0.0158 |
| 10 | Discharge coefficient α , from fig. 21, Data Sheet 5 (see footnote to example 3) | --- | 0.993 |
| 11 | Correction factor for pipe roughness | --- | --- |
| 12 | Correction factor for the diameter of the nozzle at 400° (from fig. 17, Data Sheet 3) | --- | 1.010 |
| 13 | Weight of flow G, from equation 1, Data Sheet 3 | kg/hr | 6080 |
| Tolerances | | | |
| 14 | Basic tolerance for α from Data Sheet 5, fig. 23 | percent | 0.5 |
| 15 | Tolerance for ϵ from table II | percent | 0 |
| 16 | Tolerance for reading the differential pressure $\sqrt{P_1 - P_2}$ (with V-tube manometer) | percent | 0.2 |
| 17 | Tolerance for ϵ (including tolerance for pressure and temperature measurement) | percent | 0.6 |
| 18 | Total tolerance of the measurement | percent | 0.81 |

EXAMPLE 5

Flowing material, Moist air

Given: Standard orifice $d = 282.8 \text{ mm}$ at 20° C
 Material, V2A steel
 Actual inside diameter of pipe $D = 400 \text{ mm}$

Ratio of areas $\frac{d^2}{D^2} = m = 0.5$

Installation, standard

Sought: Volume rate of flow Q referred to standard conditions, m^3/hr

| Item No. | Metering and Calculation Results | | Orifice |
|----------|---|----------------------------|-----------------|
| 1 | Gage pressure of the moist air upstream from the orifice | mm H ₂ O | 370 |
| 2 | External pressure P_o at 0° C | mm Hg kg/m ² | 759.8 10,330 |
| 3 | Absolute pressure of the moist air upstream from the orifice P_1 | kg/m ² | 10,700 |
| 4 | Temperature of the moist air upstream from the orifice t_1 | $^\circ \text{ C}$ | 60 |
| 5 | Relative humidity ϕ | 0 | 0.70 |
| 6 | Density of the saturated water vapor (at t_1), γ_D kg/m ³ | kg/m ³ | 0.1301 |
| 7 | Density of the water vapor (at its partial pressure) $\phi \gamma_D$ kg/m ³ | kg/m ³ | 0.0911 |
| 8 | Saturation pressure P_D of the water vapor at t_1 (from Knoblauch's Steam Tables, 1932) | kg/m ² | 2031 |
| 9 | Partial pressure of the vapor in the moist air (approximately) ϕP_D kg/m ² | | 1422 |
| 10 | Partial pressure of the dry air constituent $P_1 - \phi P_D$ kg/m ² | | 9278 |
| 11 | Density of the dry air constituent (from equation 9, Data Sheet 3) γ kg/m ³ | | 0.889 |
| 12 | Density of the moist air $\gamma_1 = \gamma_{tr} + \phi \gamma_D$ kg/m ³ | | 1.019 |
| 13 | Differential pressure (measured at 20° C) h mm H ₂ O | | 254 |
| 14 | Differential pressure $P_1 - P_2 = h \gamma_1$ kg/m ² | | 254 |
| 15 | Ratio pressure $\frac{P_1 - P_2}{P_1} = \frac{254}{10700}$ | | 0.0235 |
| 16 | Expansion factor ϵ , from Data Sheet 6 | | 0.993 |
| 17 | Discharge coefficient C_d , from figure 22, Data Sheet 5 | | 0.696 |

| Item No | | ϕ | | |
|---|--|-------------------------|---------|---------|
| | | 0 | 0.70 | 1.00 |
| 18 | Correction factor for pipe roughness | --- | --- | --- |
| 19 | Correction factor for lack of edge sharpness, from Data Sheet 5 | --- | --- | --- |
| 20 | Correction factor for the effect of viscosity, see (25)*; ($Re_D > Re_T$) | --- | --- | --- |
| 21 | Diameter of the orifice at measurement temperature d_1 | mm | 283.1 | 283.1 |
| 22 | Volume rate of flow of moist air under measurement conditions, from Q, equation (7), Data Sheet 3 | m ³ /hr | 10,550 | 10,820 |
| 23 | Volume rate of flow of the dry air constituent at the partial pressure of the air, $P_1 - \phi P_D$ and t_1 °C | m ³ /hr | 10,550 | 10,820 |
| 24 | Volume rate of flow of the dry air constituent under standard conditions (0° C, 760 mm Hg), $Q_N = Q \frac{\gamma_{tr}}{\gamma_N}$ | Std. m ³ /hr | 8,950 | 7,950 |
| 25 | Reynolds number Re_D in which η is assumed for dry air (since fig. 29, Data Sheet 7, is not sufficient to allow the calculation from equation (13)) | -- | 500,000 | 400,000 |
| 26 | Reynolds number at the tolerance limit Re_T | -- | 170,000 | 170,000 |
| Tolerances | | | | |
| 27 | Basic tolerance for a, from Data Sheet 5 | percent | | 0.5 |
| 28 | Tolerance for pipe roughness, from Data Sheet 5 | percent | | 0 |
| 29 | Tolerance for lack of edge sharpness, from Data Sheet 5 | percent | | 0 |
| 30 | Tolerance for ϵ , from table II | --- | | 0 |
| 31 | Tolerance for reading the differential pressure on the U-tube manometer for $\sqrt{P_1 - P_2}$ (assumed) | percent | | 0.5 |
| 32 | Tolerance for $\sqrt{\gamma}$ (assumed) | percent | | 0.6 |
| 33 | Total tolerance of the measurement | | | |
| $(\text{Item } 27)^2 + (\text{Item } 30)^2 + (\text{Item } 31)^2 + (\text{Item } 32)^2$ | | percent | | 0.93 |

*Reference 25.

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27. **Bering, F., and Zipperer, L.:** Beitrag zur Berechnung der **Zahigkeit** technischer Gasgemische (Contribution on the Calculation of the Viscosity of Technical Gas Mixtures). Das **Gas- und Wasserfach** 79, 49 (1936).

F. INDEX

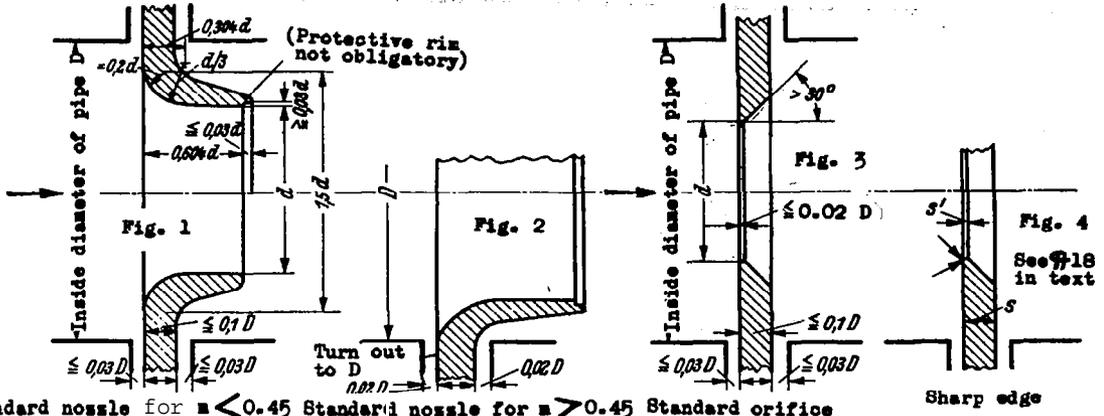
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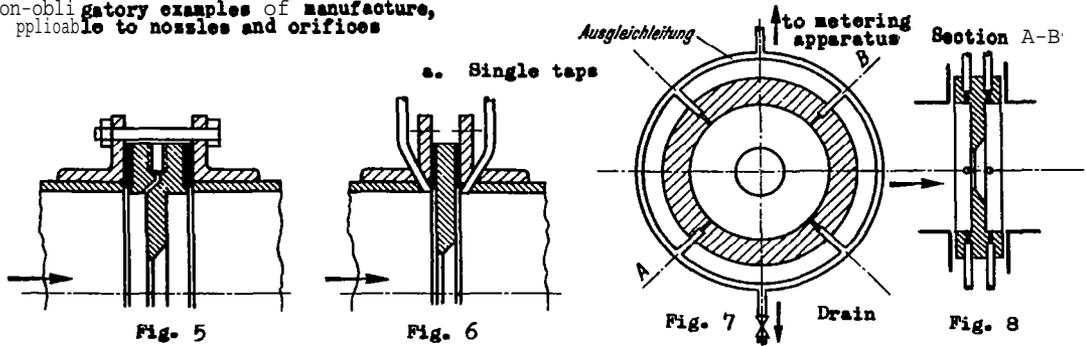
VDI Flow Meter Standard

A. Standard prescription for the form and pressure taps of standard nozzles and orifices (See ¶ 14 to 23).

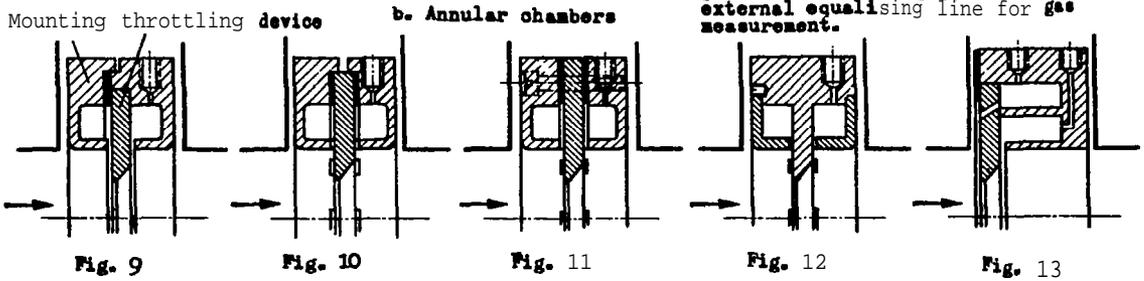


Standard nozzle for $m < 0.45$ Standard nozzle for $m > 0.45$ Standard orifice

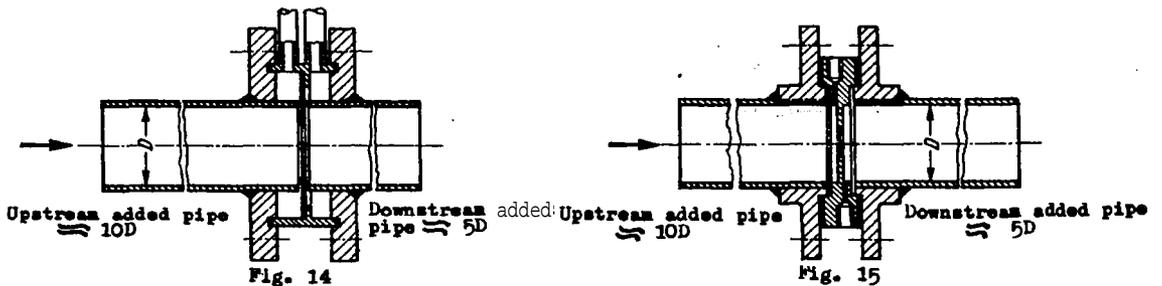
B. Non-obligatory examples of manufacture, applicable to nozzles and orifices



Figs. 7 & 8. Single pressure taps with external equalising line for gas measurement.



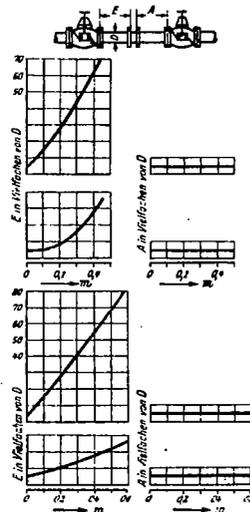
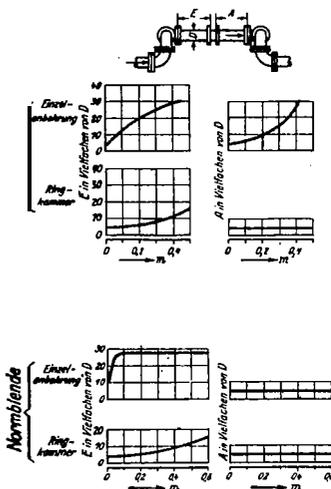
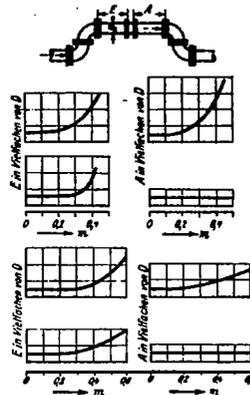
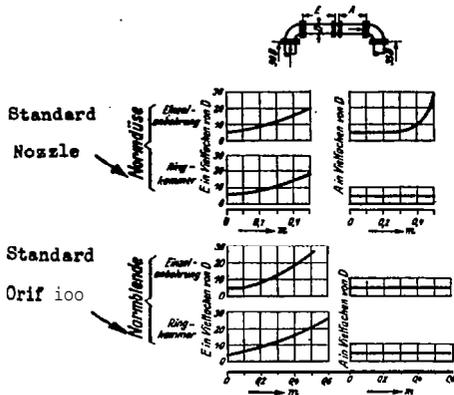
c. For pipe diameters below 70 mm (see ¶ 25)



Figs. 14 and 15. Installation types with added pipes for pipe diameters below 70 mm.

VDI Flow Meter Standard

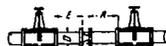
A. Necessary disturbance-free length of straight pipe (in pipe diameters D) as a function of m , with elbows and valves near the orifice



Einzelbohrung = Single pressure tap
 Ringkammer = Annular chamber
 E in Vielfachen von D = E in pipe diameters D
 A in Vielfachen von D = A in pipe diameters D

Table I

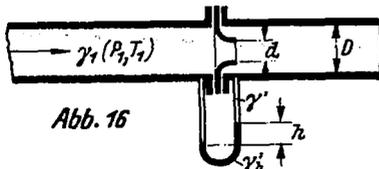
B. Necessary disturbance-free straight length of pipe (in pipe diameters D) between throttling device and gate valves for different ratios of area of the valve



Ratio of areas Nozzle / Orifice of the valve, f/F

| | | Einzelbohrung | | | | | | | | | | |
|---|--------------|---------------|-----|-----|-----|-----|-----|-----|------|------|------|----|
| | | 0,1 | 0,2 | 0,4 | 0,5 | 0,7 | 0,8 | 0,9 | 0,95 | 0,98 | 0,99 | |
| E | 0 to 0,1 | 88 | 78 | 70 | 60 | 50 | 100 | 90 | 85 | 78 | 74 | 70 |
| E | 0,1 to 0,2 | 50 | 45 | 35 | 30 | 25 | 20 | 15 | 12 | 10 | 8 | 7 |
| E | 0,2 to 0,4 | 30 | 25 | 20 | 15 | 12 | 10 | 8 | 7 | 6 | 5 | 4 |
| E | 0,4 to 0,5 | 20 | 15 | 12 | 10 | 8 | 7 | 6 | 5 | 4 | 3 | 2 |
| E | 0,5 to 0,7 | 15 | 10 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | 1 | 1 |
| E | 0,7 to 0,8 | 10 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | 1 | 1 | 1 |
| E | 0,8 to 0,9 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | 1 | 1 | 1 | 1 |
| E | 0,9 to 0,95 | 7 | 6 | 5 | 4 | 3 | 2 | 1 | 1 | 1 | 1 | 1 |
| E | 0,95 to 0,98 | 6 | 5 | 4 | 3 | 2 | 1 | 1 | 1 | 1 | 1 | 1 |
| E | 0,98 to 0,99 | 5 | 4 | 3 | 2 | 1 | 1 | 1 | 1 | 1 | 1 | 1 |
| E | 0,99 to 1,0 | 4 | 3 | 2 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 |
| A | 0 to 0,1 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |
| A | 0,1 to 0,2 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |
| A | 0,2 to 0,4 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |
| A | 0,4 to 0,5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |
| A | 0,5 to 0,7 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |
| A | 0,7 to 0,8 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |
| A | 0,8 to 0,9 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |
| A | 0,9 to 0,95 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |
| A | 0,95 to 0,98 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |
| A | 0,98 to 0,99 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |
| A | 0,99 to 1,0 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |

f/F = area of the opening of the valve / cross-sectional area of the pipe, (approximately = lift of gate / total possible lift).



A. General formulas, for liquids, gases and vapors (see # 37)

$$G = 0,01252 \alpha \epsilon d^2 \sqrt{\gamma_1} \sqrt{P_1 - P_2} \quad [kg/h] \quad (1)$$

$$Q = 0,01252 \alpha \epsilon d^2 \sqrt{\frac{P_1}{\gamma_1}} \sqrt{P_1 - P_2} \quad [m^3/h] \quad (2)$$

B. Special formulas

If the differential pressure is not measured in millimeters of water, $P_1 - P_2 = h_n (\gamma_n' - \gamma_1)$ (jx) is to be substituted in equations (1) and (2). P' or γ_n' and γ_1 see Table 2.

- a) For liquids, $\epsilon = 1$
For water with $\gamma_1 = \gamma_1' = 1 \text{ kg/l}$ and with mercury at 20° C as the manometer fluid (for other temperatures, see Table 2):

$$G = 1,403 \alpha \epsilon d^2 \sqrt{h_n} \quad [kg/h] \quad (4)$$

- b) For dry steam
With mercury and water at 20° C in the manometer:

$$G = 0,04436 \alpha \epsilon d^2 \sqrt{\gamma_1} \sqrt{h_n} \quad [kg/h] \quad (5)$$

For other temperatures of the manometer fluid, see Table 2.

- c) For gases
 γ_1' is usually negligible compared with γ_n' , so that

$$G = 0,01252 \alpha \epsilon d^2 \sqrt{\gamma_1} \sqrt{\gamma_n'} \sqrt{h_n} \quad [kg/h] \quad (6)$$

$$Q = 0,01252 \alpha \epsilon d^2 \sqrt{\frac{\gamma_1}{\gamma_n'}} \sqrt{\gamma_n'} \sqrt{h_n} \quad [m^3/h] \quad (7)$$

C. Formulas for γ_1 , kg/m^3

γ_1 is taken from tables (see Data Sheet 7) for the proper working condition or is calculated from the equation of state:

- a) For dry gases:

$$\gamma_1 = \gamma_n \frac{P_1 T_n}{P_n T_1} \frac{1}{K} \quad [kg/m^3] \quad (8)$$

- b) For gases mixed with dry water vapor:

$$\begin{aligned} \gamma_1 &= \gamma_n + \varphi \gamma_D \quad [kg/m^3] \\ \gamma_n &= \gamma_n \frac{(P_1 - \varphi P_D)}{P_1} \end{aligned} \quad (9)$$

For the error in γ_1 when humidity is not considered, see Fig. 18. Frequently the density must be determined from the flowing fluid, by means of a pycnometer or with the density meter, and converted to the service conditions.

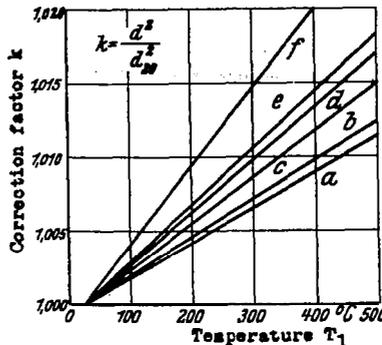
D. Conversion of the measured gas volume Q to the dry state and standard conditions.

For dry gases, $Q_n = 0,01252 \alpha \epsilon d^2 \sqrt{\frac{P_1 - P_2}{\gamma_n}} \sqrt{\frac{P_1 T_n}{P_n T_1 K}} \quad [m^3/h] \quad (10)$

For wet gases, $Q_n = 0,01252 \alpha \epsilon d^2 \sqrt{\frac{P_1 - P_2}{\gamma_1}} \sqrt{\frac{(P_1 - \varphi P_D) T_n}{P_n T_1 K}} \quad [m^3/h] \quad (11)$
(See # 35 & 38)

Approximate equation with half the error of Fig. 18.

$$Q_n = 0,01252 \alpha \epsilon d^2 \sqrt{\frac{P_1 - P_2}{\gamma_n}} \sqrt{\frac{P_1 - \varphi P_D}{P_1}} \sqrt{\frac{P_1 T_n}{P_n T_1 K}} \quad [m^3/h] \quad (12)$$



- a) Krupp V2M & V2N k.i
- b) Cast 0 to 1, gray out iron & wrought iron
- c) Nickel
- d) Red brass
- e) Krupp V2A steel
- f) Aluminum

Fig. 17 - Temperature correction for the area of the opening.

NACA Technical Memorandum No.952

VDI Flow Meter
Standard

FORMULAS

Data Sheet 3
(continued)

- G Weight of flow (kg/hr)
- Q Volume rate of flow (m³/hr) at measurement conditions P₁, T₁.
- Q_n Volume rate of flow (m³/hr) at standard conditions P_n, T_n.
- α Discharge coefficient from Data Sheet 5.
- E Expansion factor from Data Sheet 6.
- $\gamma_1 = \frac{1}{\gamma_2}$ Density of the material (kg/m³) at measurement conditions P₁, T₁.
- v₁ Specific volume of the material (m³/kg) at measurement condition P₁, T₁.
- P₁-P₂ Differential pressure (kg/m²) = (mm of water)
- d Diameter of the free opening of the throttling device (mm) at measurement condition T₁.
- d₂₀ Diameter measured at 20° C (ma).
d² = d₂₀² · k (See Fig. 17).
- h Differential pressure manometer reading (mm).
- h' Density of the manometer fluid (kg/l).
- γ Density of the material in the lines to the manometer (kg/l).
- γ_n Density at standard conditions.
- P_n Absolute pressure at standard conditionm.
- T_n Absolute temperature at standard conditionm.
- T₁ Temperature at measurement conditions.
- K Factor for the deviation from the ideal gas law from Data Sheet g.
- γ_{tr} Density of the dry gas at the partial pressure under measurement conditionm P₁, T₁ (kg/m³).
- PD Saturation pressure of water vapor at T₁. (Partial pressure of the water vapor = φ P_D (kg/m²)).
- φ Relative humidity, the ratio of the weight of water vapor to the weight of water vapor at saturation under measurement conditionm P₁, T₁ (kg/kg).
- γ_D Density of the saturated water vapor at PD (kg/m³).
- P₁ Absolute pressure at the standard pressure tap location upstream from the throttling device (kg/m²).
- P₂ Absolute pressure at the standard pressure tap location downstream from the throttling device (kg/m²).

Table 2

| °C | Density γ' (kg/l) | | | Value of the numerical constant of equation | |
|----|-----------------------|----------------------|---------|---|---------|
| | Hg(γ' _{Hg}) | H ₂ O(γ') | γ' - γ' | (4) | (5) |
| 0 | 13,596 | 1,000 | 12,596 | 1,4059 | 0,04445 |
| 10 | 13,571 | 1,000 | 12,571 | 1,4042 | 0,04441 |
| 20 | 13,546 | 0,998 | 12,548 | 1,4029 | 0,04436 |
| 30 | 13,522 | 0,996 | 12,526 | 1,4017 | 0,04433 |
| 40 | 13,497 | 0,992 | 12,505 | 1,4005 | 0,04429 |
| 50 | 13,473 | 0,988 | 12,485 | 1,3994 | 0,04425 |

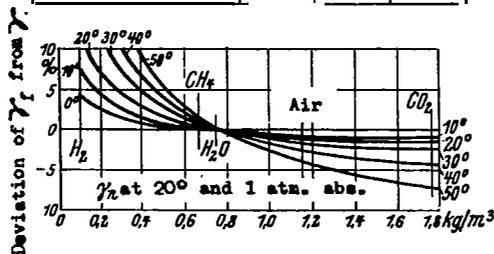
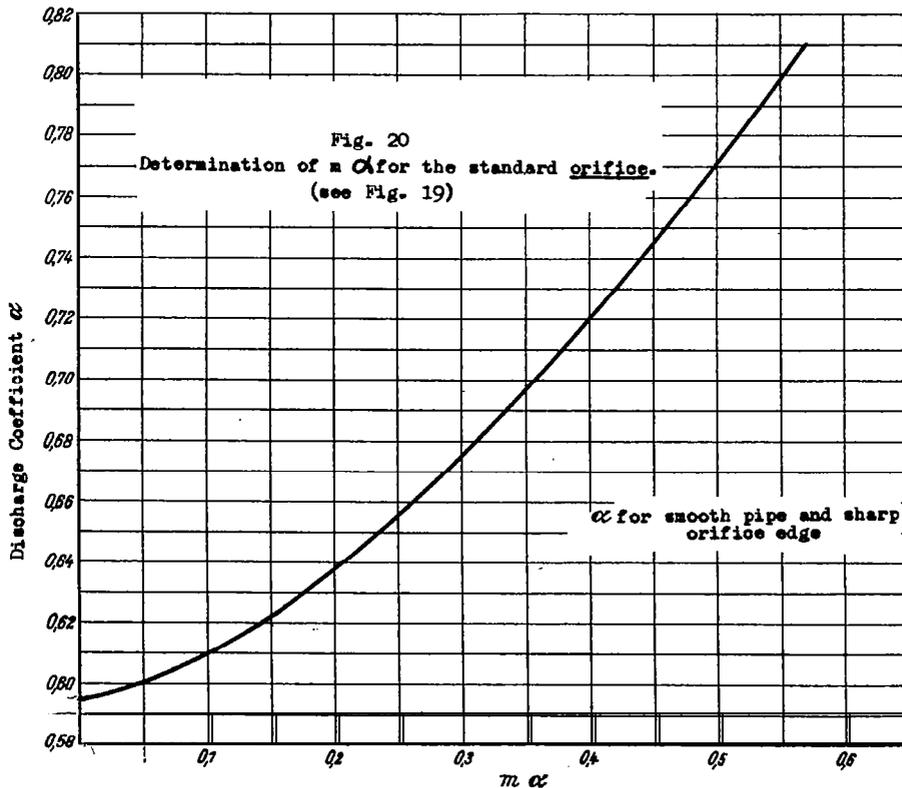
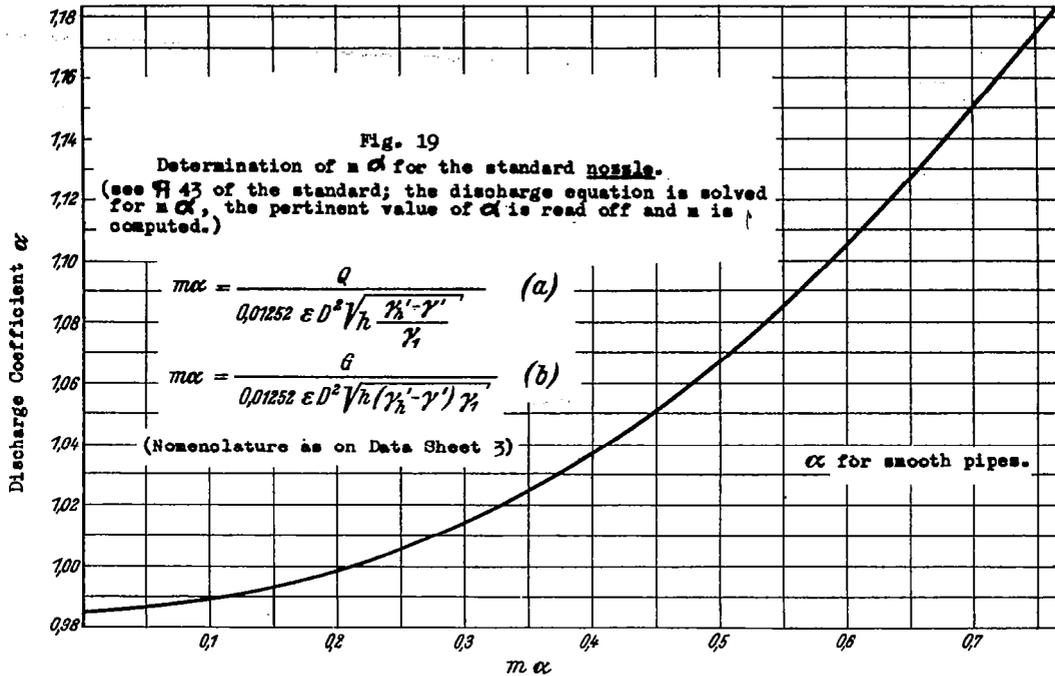


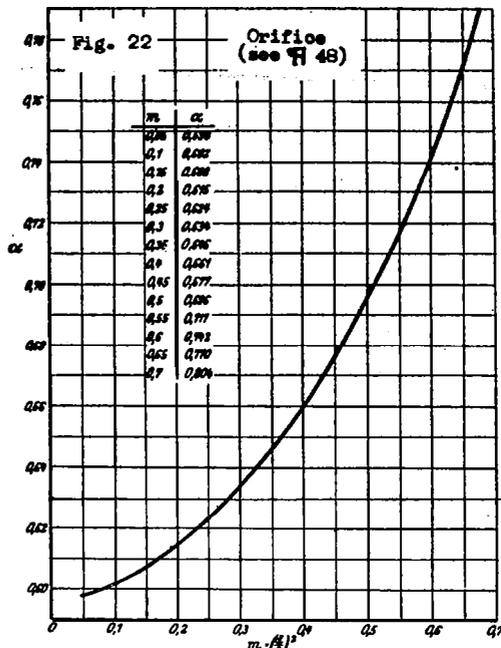
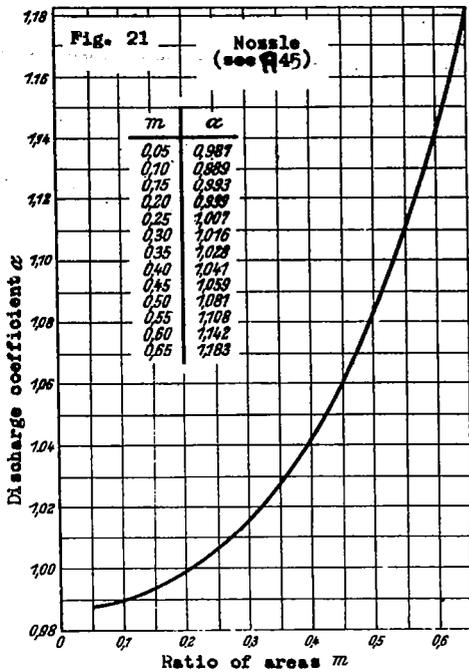
Fig. 18 - Deviation of the density of the saturated gas at 1 atmosphere absolute and at 0 to 50° C from that of dry gas.



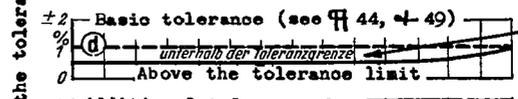
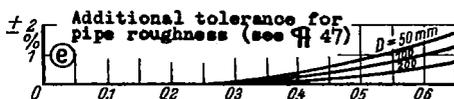
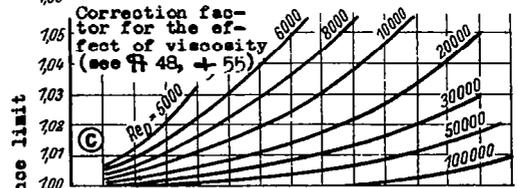
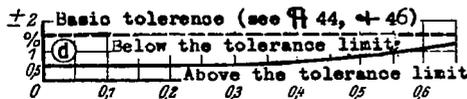
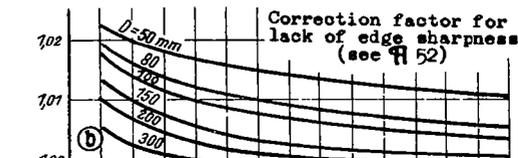
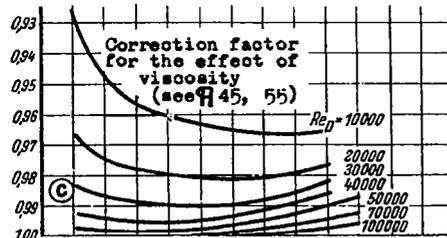
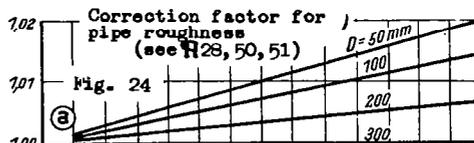
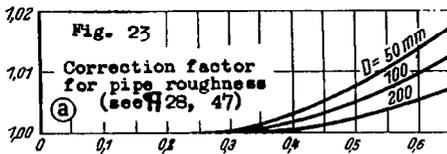
NACA Technical Memorandum No. 958

DISCHARGE COEFFICIENTS, ADDITIONAL FACTORS, AND TOLERANCES FOR STANDARD NOZZLES AND ORIFICES

VDI Flow Meter Standard



$$C_d = \frac{C_d}{\sqrt{1-m^2}}$$



Below the tolerance limit

NOTE: The correction factors are to be considered separately, the tolerances are to be set together according to the law for combining error..

NACA Technical Memorandum No. 952

EXPANSION FACTOR FOR SUPERHEATED STEAM
AND DIATOMIC GASES

Data Sheet 6

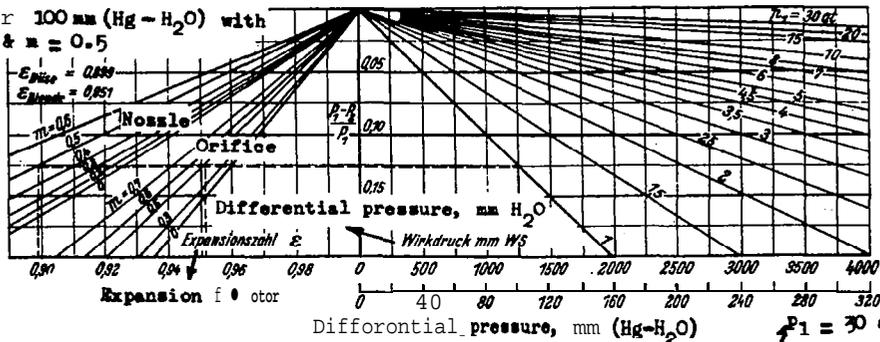
VDI Flow Meter
Standards

A. Expansion factor ϵ , as a function of the differential pressure and the absolute pressure.

Example: for 100 mm (Hg-H₂O) with
 $P_1 = 1 \text{ atm.}$ & $n = 0.5$

Fig. 25
 $k = 1.31$

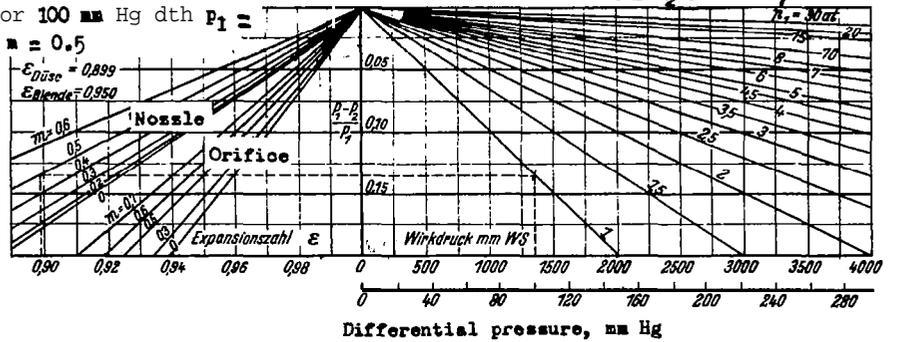
(Superheated Steam)



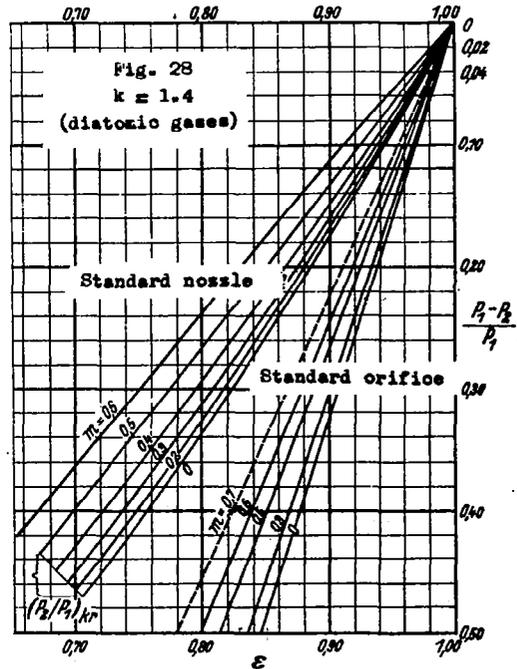
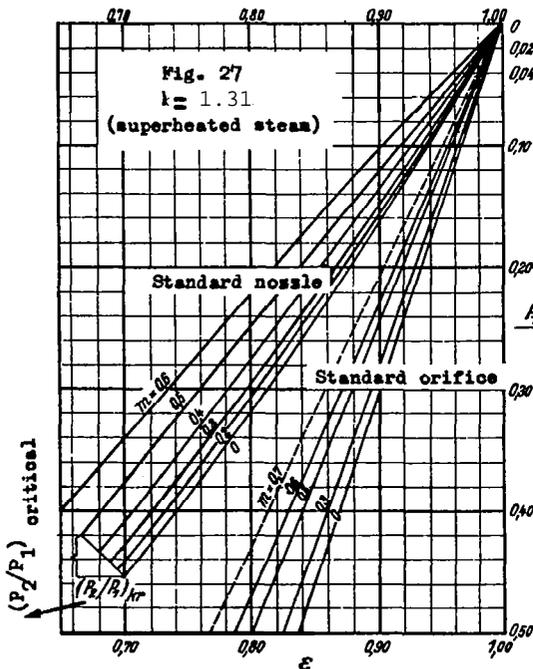
Example: for 100 mm Hg dth $P_1 =$
1 atm. and $n = 0.5$

Fig. 26
 $k = 1.4$

(diatomic gases, air)



B. ϵ as a function of the ratio of the differential pressure to the absolute pressure and for greater expansions. (For other values of k , see Fig. 5 in the text).



NOTE: The tolerances for ϵ are to be taken from § 57 and § 58 of the text.

Table 3
Density γ_n and dynamic viscosity $\eta \cdot 10^6$

Table 4
Dynamic viscosity $\eta \cdot 10^6$ (kg sec/m²)

| | γ_n [kg/m ³] | | QS = Hg | | $\eta \cdot 10^6$ [kg/m ²] | | | | |
|--------------------------|---------------------------------|-----------------|------------|------|--|------|------|------|--|
| | 20°C / 1 kg/cm ³ | 0°C / 760 mm Hg | -30° | 0° | +50° | 100° | 200° | 300° | |
| Air | 1,165 | 1,293 | 1,59 | 1,77 | 2,04 | 2,27 | 2,65 | 3,03 | |
| O ₂ | 1,268 | 1,425 | 1,77 | 1,94 | 2,22 | 2,51 | 3,06 | | |
| N ₂ , CO | 1,126 | 1,230 | 1,55 | 1,70 | 1,94 | 2,17 | 2,57 | | |
| CO ₂ | 1,783 | 1,977 | 1,30 | 1,44 | 1,66 | 1,89 | 2,32 | | |
| NH ₃ | 0,695 | 0,771 | 0,85 | 0,94 | 1,15 | 1,34 | | | |
| Cl ₂ | 2,9 | 3,220 | 1,16 | 1,28 | 1,50 | 1,72 | | | |
| H ₂ | 0,0809 | 0,0899 | 0,82 | 0,89 | 0,99 | 1,09 | 1,26 | 1,42 | |
| CH ₄ | 0,647 | 0,717 | 0,95 | 1,05 | 1,22 | 1,40 | | | |
| H ₂ S | 1,387 | 1,539 | 1,07 | 1,20 | 1,42 | 1,64 | | | |
| Luft- starkstoff | 1,133 | 1,237 | | | | | | | |
| H ₂ O - Vapor | (0,727)* | (0,810)* | Calculated | | | | | | |

| | -20 | -10 | 0 | +10 | +20°C |
|-------------------------------------|------|------|------|------|-------|
| 1) of gases at the saturation line. | | | | | |
| NH ₃ | 1,11 | 1,15 | 1,19 | 1,25 | 1,34 |
| CO ₂ | 1,85 | 1,70 | 1,77 | 1,87 | 2,07 |
| SO ₂ | 1,08 | 1,14 | 1,25 | 1,30 | 1,54 |
| 2) of liquified gases. | | | | | |
| NH ₃ | 25,8 | 25,0 | 24,4 | 23,5 | 22,6 |
| CO ₂ | 12 | 11,3 | 10,2 | 9 | 7,2 |
| SO ₂ | 48,4 | 44,4 | 39,2 | 35,6 | 27,6 |
| H ₂ O | | | 183 | 133 | 103 |

Atmospheric Nitrogen

For other technical gases, see DIN E1871 & E1872

Calculation of the Reynolds Number (see § 72 ff. of the Standard)

If the discharge G is given in kg/hr, Q_n in m³/hr, or Q_n in normal m³/hr, it is expedient to set

$$Re_D = 36,1 \cdot 10^{-3} \frac{G}{D \eta} \quad \dots (13)$$

$$= 354 \cdot 10^{-3} \frac{Q}{D \nu} \quad \dots (14)$$

$$\text{or} \quad = 354 \cdot 10^{-3} \frac{Q_n}{D \nu'} \quad \dots (15)$$

in which η = the dynamic viscosity (kg sec/m²)

$\nu = \frac{\eta}{\gamma}$, the kinematic viscosity (m²/sec)

$\nu' = \frac{Q_n}{\gamma \rho}$ a calculating quantity

D = the pipe diameter (mm)

Fig. 29 enables an easy determination of the Reynolds number to be made graphically (see § 73 of the Standard). In Fig. 29, η is given for steam, ν for water, and ν' for gases. In using the figure, η or ν' for the material in question is sought and the start is made from the left ordinate into the nomograph, as the following examples show.

Examples:

1. Air at 20°, volumo rate of flow $Q_n = 45$ m³/hr. Line diameter D = 20 mm. First find ν' for air at 20°, go horizontally to the left ordinate and connect the intormotion with the value 45 m³/hr On the Q line of the nomograph, continuing the line to the auxiliary line H. Then connect this intersection with the value of 20 on the D line. Prolonging this line beyond H gives $Re_D = 52500$, approximately.
2. Steam, 500 kg/hr at 10° tm. abs. and 275°. Pipe line diameter D = 200 mm. Find η , go from the ordinate, crossing the G line of the nomograph at the Point 5000, to the auxiliary line H. Then connect this intersection with 200 on the D line; the prolongation of this line gives $Re_D = 40000$.

Conversion of the Poise to Technical Units.
If η is given in c.g.s. units (poise) (as in Landolt-Börnstein), divide it by 98.1 to convert to technical units, kg sec/m².

Conversion of Engler Degrees to Kinematic Viscosity.

If the viscosity is given in Engler degrees, conversion is made with the formula:

$$10^6 \cdot \nu = 7,32 E - \frac{6,31}{E} \quad \dots (16)$$

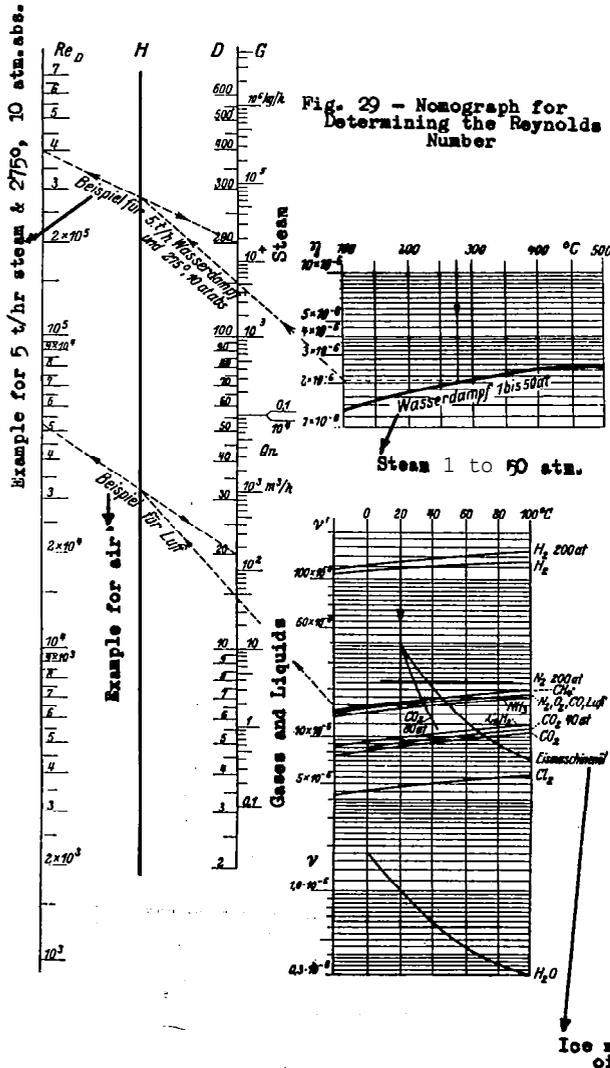


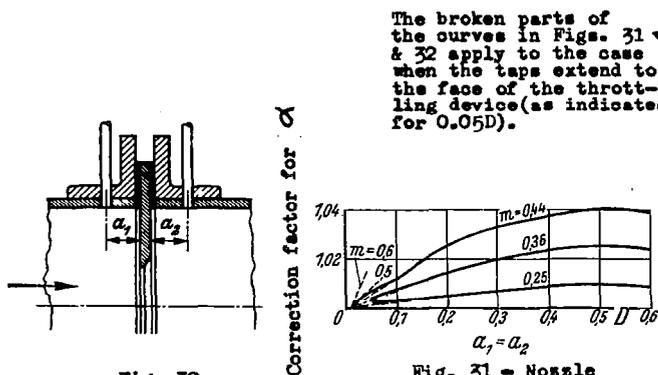
Fig. 29 - Nomograph for Determining the Reynolds Number

Steam 1 to 50 atm.

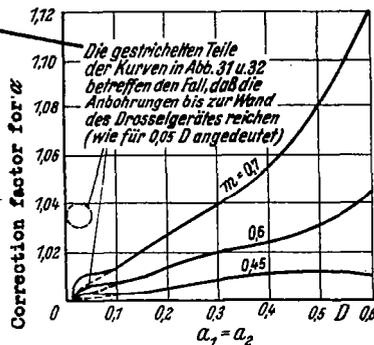
Gases and Liquids

Ice machine oil

A. Pressure Taps Similar to the Standard.



The broken parts of the curves in Figs. 31 & 32 apply to the case when the taps extend to the face of the throttling device (as indicated for 0.05D).

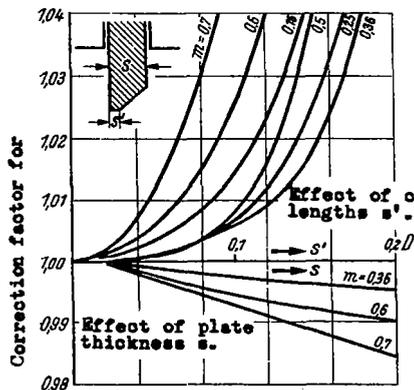


Figs. 31 & 32 - Correction factor for α with pressure taps similar to the Standard as shown in Fig. 30 at distances $a_1 = a_2$ from the faces of the plates in normally rough pipes (see also the course of pressure in Fig. 8 of the test).

Additional tolerance 0.5% up to $m = 0.4$ for nozzles or 0.5 for orifices } with $0.03D < (a_1 = a_2) < 0.2D$
 1% for larger values of m for nozzles & orifices

Fig. 34 - Error Caused by Different Installation Disturbances with Orifices.

S. Orifices with Non-Standard Plate Thicknesses and Cylindrical Lengths.



Correction factor for the discharge coefficient for non-standard plate thicknesses and lengths of the cylindrical parts.

Effect of the cylindrical length s (upper half) and the plate thickness s (lower half).

Additional tolerance for $\alpha \pm 1\%$

| Installation Disturbance | Ratio of areas | Maximum order of the metering error to be expected, % (+ indicates that the flow is greater than actual, - smaller than actual). |
|--|----------------------------|---|
| 1. Installed downstream from a tapered expanding piece. | $m = 0.16$ 0.49 | -10 -50 |
| 2. Installed downstream from a tapered reducer. | 0.16 0.49 | ± 0.5 +2 |
| 3. Installed upstream from a tapered reducer. | ≈ 0.16 0.49 | ± 1 -1 |
| 4a. Einbau in eine Rohrleitung mit geneigten Flanschen (Diam. nach DIN 2604, W40.) | 0.18 0.43 0.53 | -0.5 -2.5 -3 |
| 4b. 200 mm diam. | 0.18 0.43 0.53 | -0.5 -2.5 -3 |
| 4c. | < 0.5 0.53 | < -1 -1 |

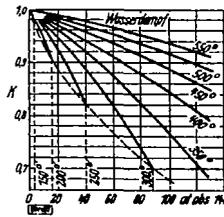
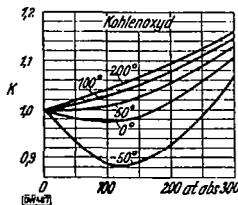
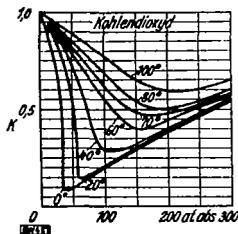
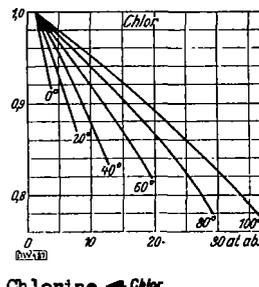
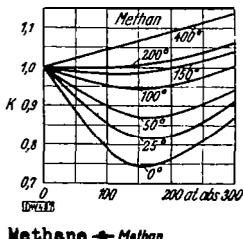
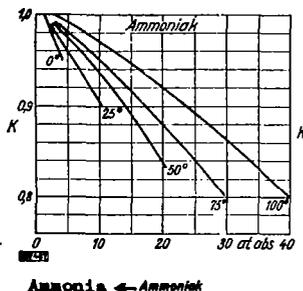
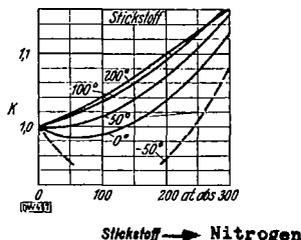
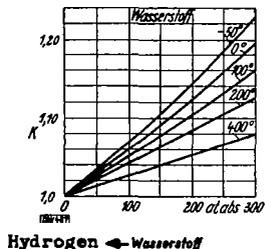
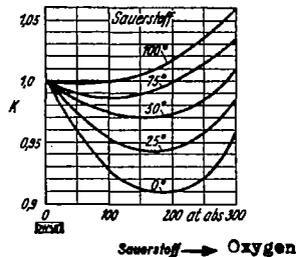
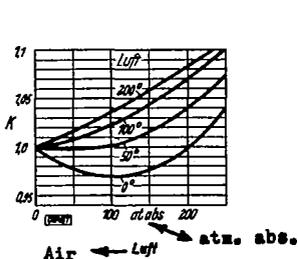
4a. Installed in a pipe line with riveted flanges. (Rivets in accordance with DIN 2604, W40.)

with nozzles

At high pressures the actual density γ deviates from the theoretical value γ_{th} of the ideal gas according to the gas law (see # 66), and

$$K\gamma = \gamma_{th}$$

In this, $K =$ referred to 1 atm. abs. - is to be taken from the figures below for different gases.



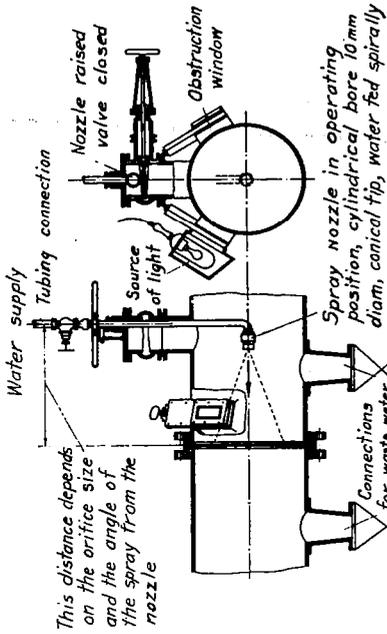


Figure I.- Spray device for orifices measuring gases containing dust and tar.

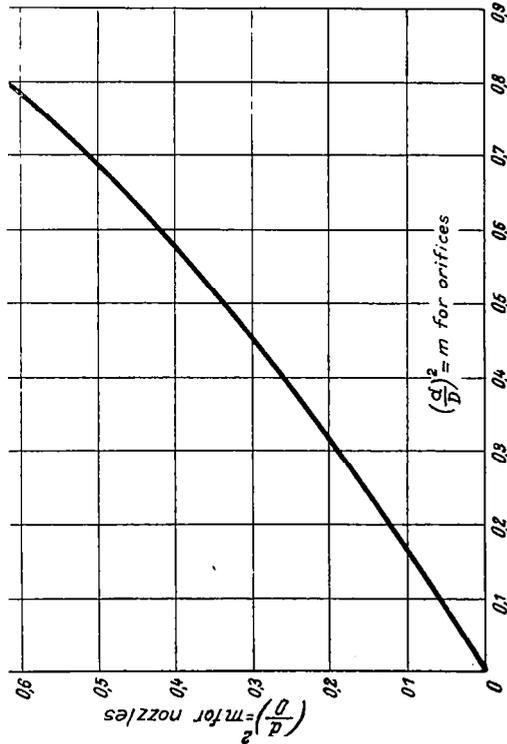


Figure II.- Equivalent ratios of area m for standard orifices and nozzles with the same differential pressure for the same discharge.

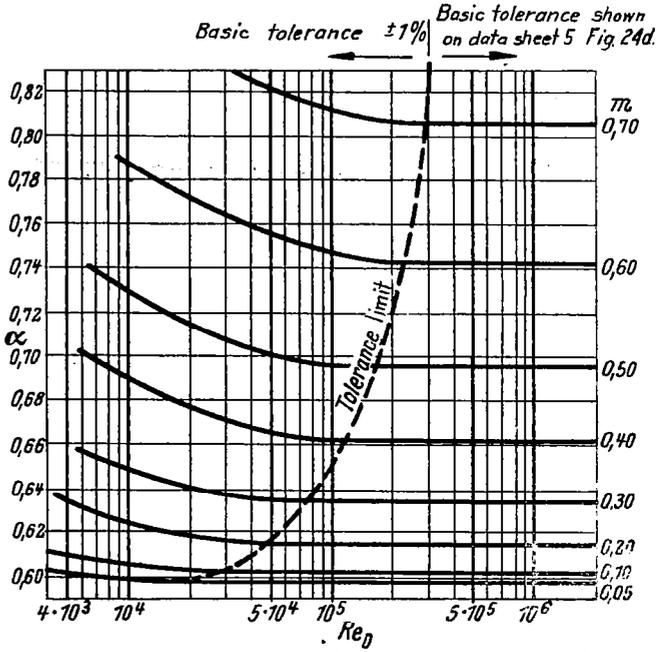


Figure III.- Discharge coefficients α of the 1930 standard orifice as a function of the Reynolds number Re_D drawn logarithmically.

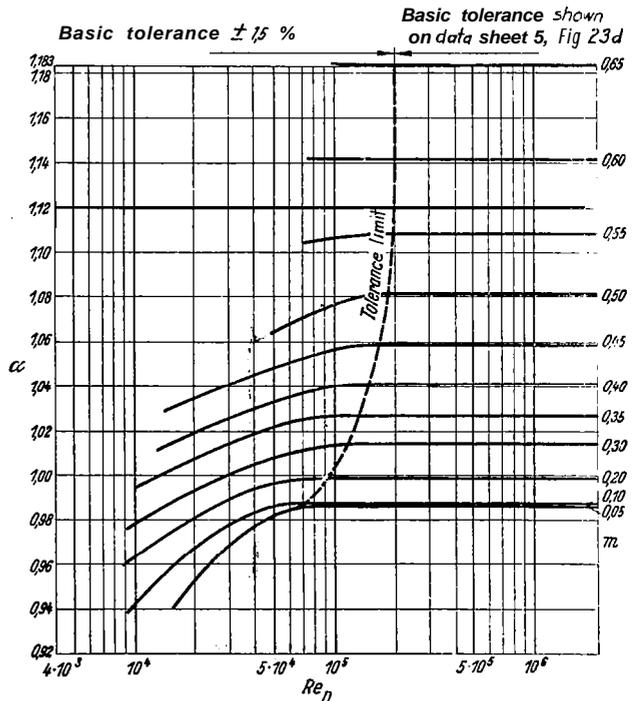


Figure IV.- Discharge coefficients α of the 1930 standard nozzle as a function of the Reynolds number Re_D (referred to the pipe diameter D) drawn logarithmically.

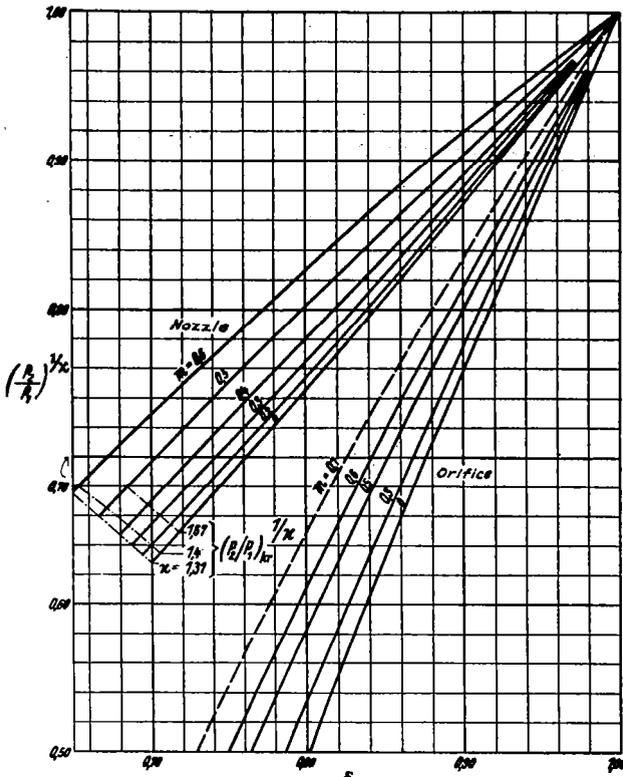


Figure V.- Expansion factor ϵ for any value of k as a function of $(p_2/p_1)^{1/k}$ (figs. 27 and 28 of data sheet 6 show ϵ as a function of p_2/p_1 for $k=1.31$ end 1.4).

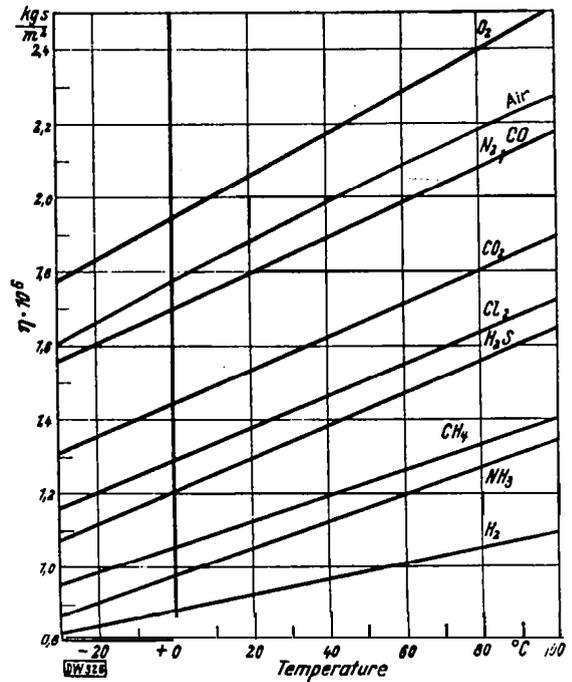


Figure VI.- Dynamic viscosity η of gases (Landolt-Börnstein, 1927).

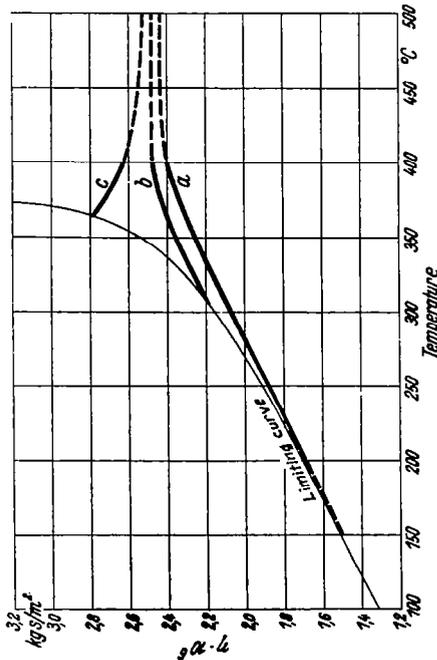


Figure VIII.- Dynamic viscosity η of water vapor (ref.18). **a** holds for 1 to 25 at, **b** for 100 at, **c** for 200 at. Intermediate values may be interpolated.

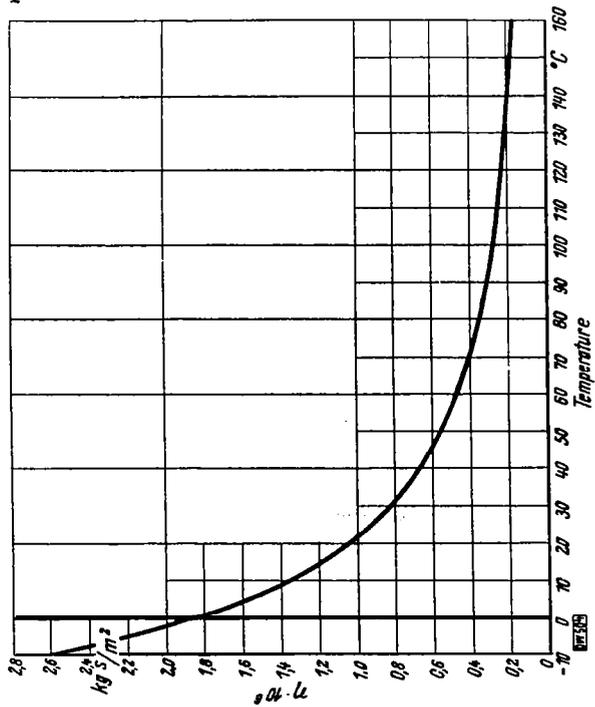


Figure VI.- Dynamic viscosity η of water.

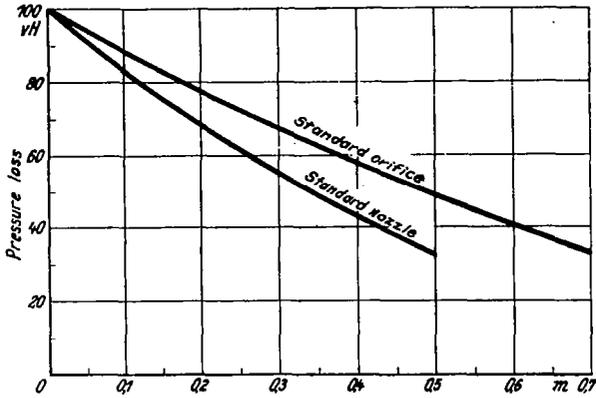


Figure IX.- Residual pressure loss for standard nozzles and standard orifices in percent of differential pressure $P_1 - P_2$.

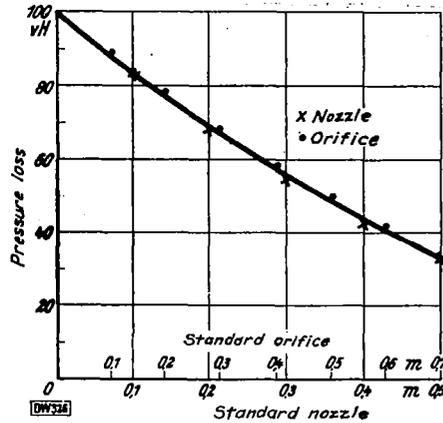
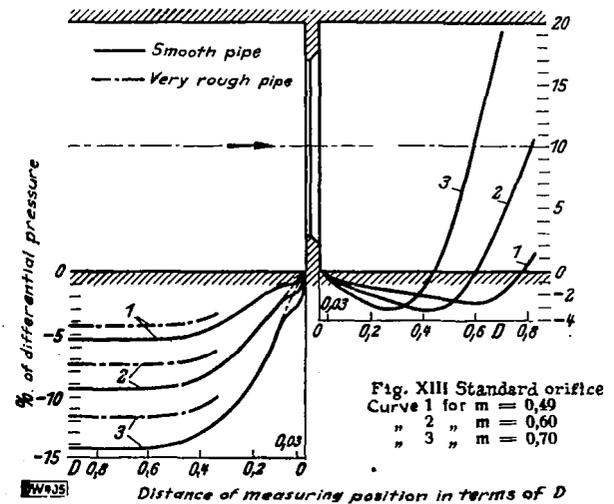
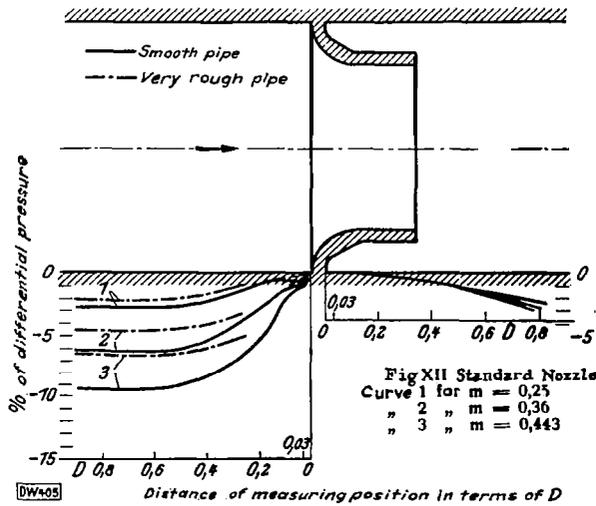


Figure X.- Residual pressure loss for standard nozzles and standard orifices for equal differential pressure and for equal discharge.



Figures XII and XIII.- Deviations of the pressure at the pipe wall from the pressure at the standardized measuring position, expressed in percent of differential pressure (see also A-B. 8, fig. 30 to 32).

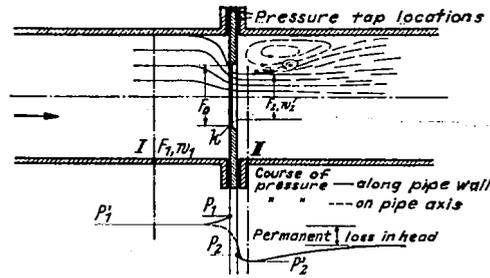
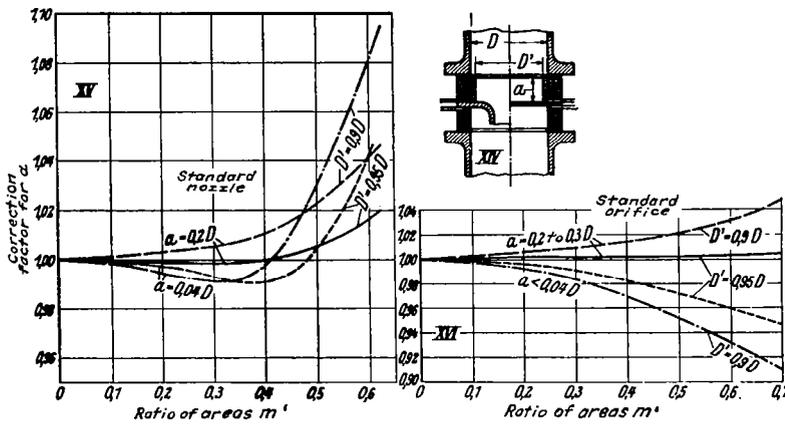
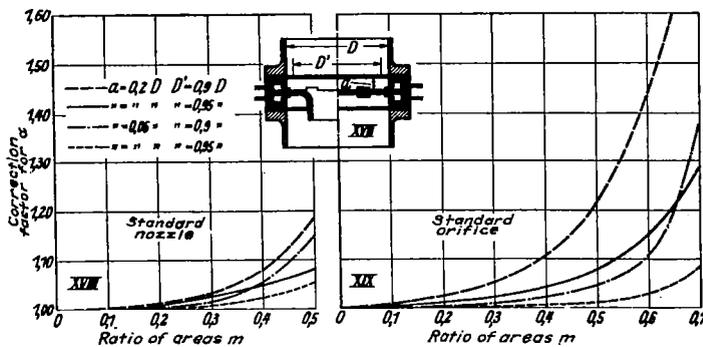


Figure XI.- Flow picture and course of pressure in throttling devices.



Figures XIV, XV, and XVI.- Effect of a projecting mounting ring on the standard nozzle and the standard orifice.



Figures XVII, XVIII, and XIX.- Effect of projecting gaskets on the standard nozzle and the standard orifice.

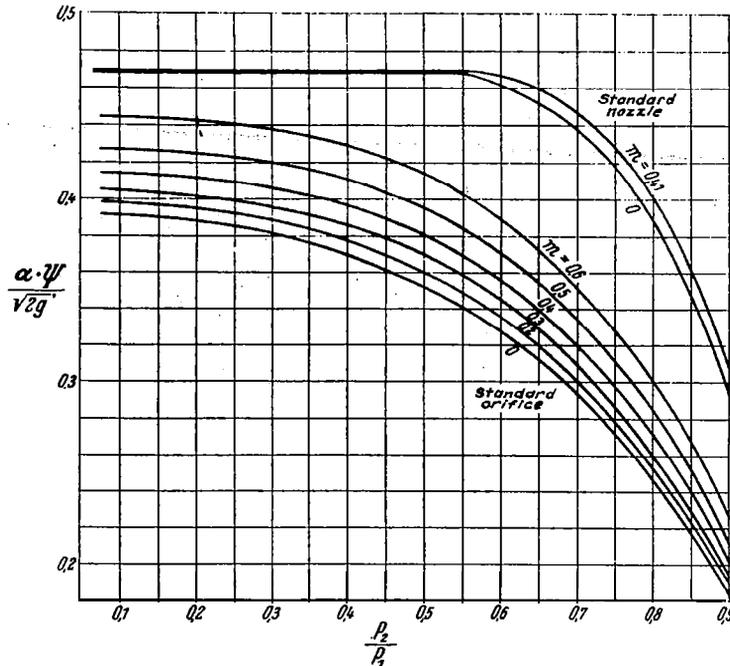
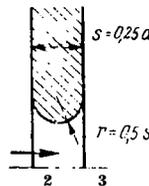
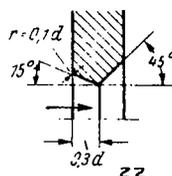


Figure XX.- Values of $\frac{\alpha \cdot \psi}{\sqrt{2g}}$ as functions of p_2/p_1 and m for $k = 1.31$ (for recalculation to $k = 1.41$, see ¶ 102).

| Throttling devices | for Re_D | and for m approx. |
|--------------------|------------------|---------------------|
| Fig. XXI | 4 000 to 100 000 | 0.3 to 0.4 |
| Fig. XXII | 3 000 to 100 000 | 0.16 to 0.25 |
| Fig. XXIII | 3 000 to 100 000 | < 0.4 |



Figures XXI, XXII and XXIII - Types of throttling devices reassembled for small Reynolds-numbers.

Figure XXI - Standard nozzle without cylindrical throat

Figure XXII - Double-bevelled orifice with rounded approach edge and sharp middle edge.

Figure XXIII - Double-rounded nozzle.

Figure XXIV - Arrangement for double-pressure type independent of each other.

