Nicholas P. Cheremisinoff
Paul N. Cheremisinoff

INSTRUMENTATION FOR

Process Flow Engineering

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BY

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Foreword

This volume provides a summary of flow instrumentation hardware available to the processing industries. Application and selection guidelines are furnished along with typical service factor data, operating ranges and manufacturers' specifications. The body of information and data presented are applicable to all the process industries, although it should be noted that the same type of application is not necessarily handled in the same manner in all these industries.

The volume is intended as a quick reference for process, environmental, mechanical and flow system engineers. The chapters are prepared in a short, concise format to provide working knowledge of specific flow measurement instrumentation. Sample calculations are provided to illustrate the range of applicability and basis for selecting one instrument over another for an intended application. Theoretical discussions of flow dynamics are excluded from this volume; however, ample references are provided for more in-depth reading.

NICHOLAS P. CHEREMISINOFF
PAUL N. CHEREMISINOFF

Biographies

Nicholas P. Cheremisinoff heads the Product Development Group of the Elastomers Technology Division of Exxon Chemical Company, Linden, New Jersey. He is involved in product research and development of polymeric materials for the consumer market and design/development of multiphase reactors and instrumentation for complex process control. He is the author/co-author of numerous books and papers on the subject of applied fluid mechanics. Dr. Cheremisinoff received his B.S., M.S. and Ph.D. degrees in Chemical Engineering from Clarkson College of Technology. He is a member of several professional and honor societies, including Tau Beta Pi and Sigma Xi.

Paul N. Cheremisinoff is Professor of Civil and Environmental Engineering and Director of the Physical Treatment Division of the Industry/University Cooperative Center for Research in Hazardous and Toxic Substances, New Jersey Institute of Technology. Professor Cheremisinoff has more than 35 years' experience in research, engineering design, and instrumentation for a wide range of organizations. He is author and coauthor of numerous papers and books and is a licensed professional engineer, Fellow of the New York Academy of Sciences, and member of Sigma Xi and Tau Beta Pi.

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METERING PERFORMANCE AND ESTIMATING DISCHARGE COEFFICIENTS OF FLOW METERS

DISCHARGE COEFFICIENT

SELECTING THE MOST suitable flow meter for a given application is not always easy. The reason is a lack of guidance that would lead to the proper choice by using the mass of unbiased scientific papers, ASME and other flow metering codes, and the often more or less biased technical and sales publications of flow meter manufacturers.

This chapter outlines basic information for assessing the metering performance of widely used differential pressure devices. Hopefully, this will assist the user in the typical analysis of any kind of flow meter.

The coefficient of discharge, C, for all types of differential producers is defined as

$$C = \frac{\text{actual rate of flow } (q_a)}{\text{theoretical rate of flow } (q_{th})} \tag{1}$$

where q_{th} is calculated based on device characteristics that are measurable "on the bench" (without the device being yet exposed to flow) and based on the signal emitted by the device due to flow passing through it.

The actual rate of flow $= q_{th} \times C$. The behavior and tolerance of this C value is critical when analyzing the metering performance of differential producers.

While the definition of C in Equation (1) is well known, the generality of its meaning is seldom realized. According to ASME[1], C is the bridge between the "invisible" rate of flow and the "visible," therefore measurable, device parameters not only for differential producers, but in one way or another for any kind of flow meter.

This general rule applies to differential producers through the basic actual volume rate formula

$$q_a = A \times V_A \times C \tag{2}$$

where

A = some cross-sectional area provided (and preserved) by the device, therefore, measurable.

V_A = average velocity related to cross section A and calculated from ΔH_a.

ΔH_a = the differential pressure signal produced by the device due to q_a passing through it.

The exact relation between V_A and ΔH_a is often not known and the actual cross section of the flow at A and its true velocity may not be the same as A and V_a, but deviate from them to a greater or lesser degree, depending on the "visible" geometry or hydraulic shape of the device. Therefore, the empirically established discharge coefficient, C, is used to account for these deviations.

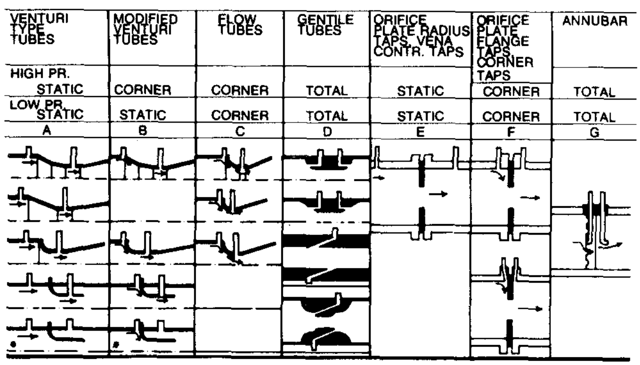
The value and behavior of C depends on the hydraulic shape of the device. Therefore, determining the correlation between the two becomes an important task.

Figure 1-1 summarizes the commonly used differential producers. These are divided into seven groups according to their "visible" shape and/or the way they sense high and low pressure.

Figure 1-2a shows typical empirical C values for these groups as a function of the beta ratio. Reviewing these figures, one can make the following observations. (For the identification of tube sections, see Figure 1-3.)

For Group A, venturi-type tubes, the common characteristic of the hydraulic shapes of the tubes is to provide flow parallel to the tube wall at the cross section of the inlet and throat taps. The hydraulic shape achieves this by

1. Locating the inlet tap sufficiently upstream (in the entrance section) from the beginning of the narrowing transition section, thus assuring



*By their basic hydraulic mechanism, these devices belong to groups shown, but, not sensing low pressure in the throat, their uncalibrated tolerance level is inherently poorer, also their R_D sensitivity is greater and poorly predictable.

Figure 1-1. Types of differential producers.

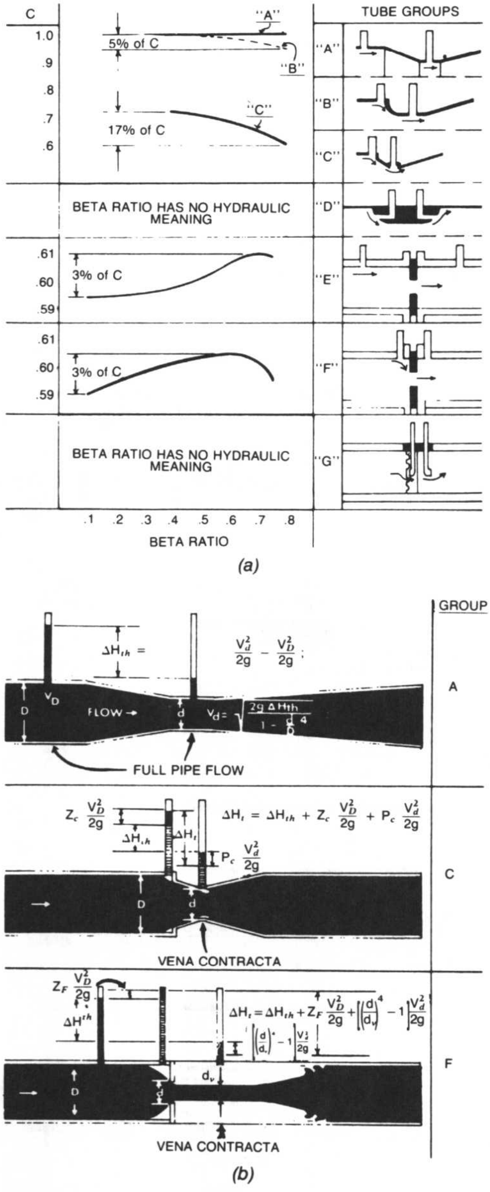


Figure 1–2. (a) Shows discharge coefficient's dependence on beta ratio. (b) Schematic of flow mechanisms.

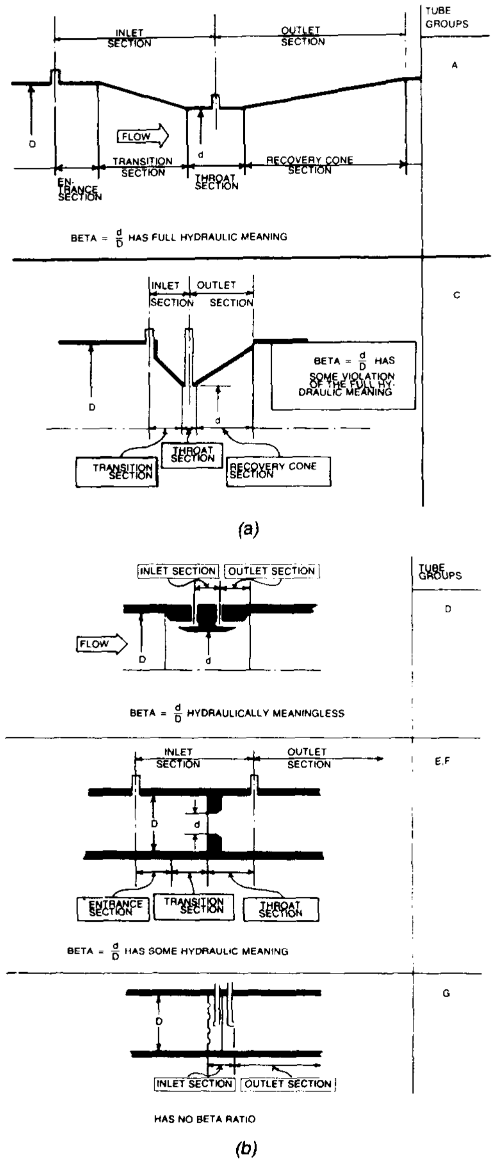


Figure 1–3. Parameters of hydraulic shapes of differential producers.

no, or insignificant, corner tap type effect for the inlet pressure sensation.

2. A long transition section, reducing the inlet diameter to that of the throat.
3. A long throat section where the throat tap is located sufficiently far away from the entrance and exit cross sections of the throat.

This hydraulic shape provides true static pressure sensation at inlet and throat taps which, according to Figure 1–2a, yields high *C* values (close to 1) which are independent of the beta ratio.

For Group B, modified venturi tubes, the hydraulic shape of the tubes in this group (Figure 1–1) differs from the ones in Group A in the inlet pressure sensation only; namely, the inlet tap is moved in the corner formed by the pipe wall and the beginning of the transition section. These hydraulic shapes, therefore, have no entrance section. Consequently, the inlet tap "sees" the effect of the change in direction of the flow caused by the hydraulic shape (Pitot effect), while the throat tap sees true static pressure, having the same design as Group A tubes.

The typical result of this hydraulic shape according to Figure 1–2a is: a still high *C* value, which, however, becomes somewhat dependent on the beta ratio. As beta becomes higher, *C* becomes lower.

For Group C, flow tubes, the common characteristics of hydraulic shape are (Figure 1–1) that both inlet and throat taps are located at cross sections where the direction of flow is changed by the hydraulic shape of the device. Hence, their pressure sensation is influenced by the Pitot effect. They can be distinguished by

1. The corner inlet tap.
2. The short and sharp transition section.
3. The length of the cylindrical throat, which is either zero or extremely short, forming an "outside" corner in front of the throat tap.

This hydraulic shape results in a low *C* value that is greatly influenced by beta ratio.

Group D, gentile tubes (beta flow tubes) have a drastically different hydraulic shape which does not provide the physical phenomenon on which the venturi metering principle is based. Their inlet tap is not located at the inlet cross section (inlet diameter) of the tube, and their "throat" tap is not located in the throat. Instead, their hydraulic shape has symmetrical inlet and outlet sections where the upstream and downstream total pressure taps are located symmetrically at some diameter smaller than the pipe size.

In this case, the *C*-value is low and changes with a hydraulically meaningless beta ratio in an undefinable, incorrelatable manner, which is the reason for not showing a typical curve for these tubes in Figure 1–2a.

Group E, orifice plates with "static" taps will be discussed, having the inlet tap one pipe diameter upstream of the inlet face of the plate, while their outlet tap is either at the location of the vena contracta or just one-half pipe diameter downstream of the inlet face of the plate (which location closely approximates that of the vena contracta for commonly used beta ratios).

The hydraulic shape of this device provides

1. True inlet static pressure sensation.
2. "Some" static pressure sensation downstream of the geometrical throat.
3. A geometrical throat section with very short axial length. The result of this hydraulic shape is a C whose value is low and moderately dependent on the beta ratio, as shown in Figure 1–2a.

Group F, orifice plates with corner taps are essentially the same as in Group E, except inlet and outlet taps are located in or near the corner formed by the pipe wall and the plate. Note that the corner tap has basically the same hydraulic meaning for all line sizes, whereas the flange taps change their hydraulic meaning with line size. This can be appreciated by realizing that the location of flange taps is 1 inch upstream and downstream of the inlet face of the plate for any line size. Consequently, they act almost as corner taps for large line sizes and almost as static taps for small line sizes when distances are measured in terms of pipe diameters, as they should be.

The result of the deviation of the hydraulic shape of these plates from the Group E ones is shown in Figure 1–2a. Namely, the C's at smaller beta ratios have almost the same values as those of Group E plates, while at higher betas they become lower due to the increasing inlet velocity head effect on the inlet tap, as in the case of Group B tubes.

Group G, velocity meters (annubars) are not, in reality, flow meters because they do not provide and preserve the area to which the velocity indicated by the differential output of the device can be related [Equation (2)]. They do not have any hydraulic shape. Consequently, they have nothing to do with the venturi tube metering principle. Their characteristics, therefore, cannot be described in terms of the beta ratio. They are basically modified Pitot tubes, comparing total inlet pressure to some outlet pressure.

DESIGN FORMULAS

For all differential producers, the flow rate is calculated by the same "venturi" formula. We should, therefore, review the physical meaning of the venturi metering principle.

The actual volume rate of flow through venturi tubes for incompressible flow can be calculated from

$$q_a = C \times a \times \sqrt{\frac{2gx\Delta H_a}{1 - \beta^4}} \tag{3}$$

Note that

$$a = \left(\frac{d}{D}\right)^2 D^2 \times \frac{\pi}{4} = \beta^2 D^2 \frac{\pi}{4} \tag{4}$$

Therefore, Equation (3) becomes

$$q_a = CD^2 \frac{\pi}{4} \sqrt{2g\Delta H_a} \frac{\beta^2}{\sqrt{1 - \beta^4}} \tag{5}$$

This shows the strong dependence on β.

The theoretical or ideal equation (Figure 1–2b) is

$$q_{th} = D^2 \frac{\pi}{4} \sqrt{2g\Delta H_{th}} \frac{\beta^2}{\sqrt{1 - \beta^4}} \tag{6}$$

where

$$\Delta H_{th} = \frac{V_d^2}{2g} - \frac{V_D^2}{2g} = H_{Dth} - H_{dth} \tag{7}$$

Now, for the case of $q_a = q_{th}$

$$C = \sqrt{\frac{\Delta H_{th}}{\Delta H_a}} \tag{8}$$

from which

$$\Delta H_a = \frac{\Delta H_{th}}{C^2} \tag{9}$$

Note that Equations (6) and (7) would be true only under the following conditions:

1. If the beta ratio had true "hydraulic meaning," namely, if it were the ratio of the diameters of such throat and inlet cross sections, where
   A. The actual cross section of the flow was equal to these geometrical areas, and

B. If, at these areas, true static pressures were sensed.

2. If the inlet and throat velocity profiles were perfectly blunt.
3. If no energy loss occurred due to flow in the tube between inlet and throat pressure taps.

Since the flow of real fluids through different hydraulic shapes deviates from some or all of the foregoing requirements to different degrees, the deviation of C from the value of 1 [Equation (8)] or the deviation of ΔH_a from ΔH_{th} [Equation (9)] becomes a good measure of the extent to which devices of different hydraulic shapes violate the ideal, or theoretical, venturi tube metering principle [Equations (6) and (7)].

For Group A, venturi type tubes, the value of the C found empirically (Figure 1–2a) for these devices is practically independent of beta and can be between 0.97 and 1. This means, by Equation (9), that ΔH_a can be equal to or 6.4 percent greater than ΔH_{th}.

The fact that the actual kinetic energy content of the flowing fluid is not $V^2/2g$ at the inlet and throat taps, because the actual velocity profiles are not blunt, entails that Condition 1 is not satisfied. Energy loss (or head loss) occurs between the inlet and throat taps increasing the ΔH_a over ΔH_{th} that would be justified by flow acceleration only in the throat; consequently, Condition 3 is not satisfied.

The flow mechanism in venturi-type tubes, therefore, does deviate from the theoretical Equations (6) and (7). It is important to note, however, that the deviation of the actual flow mechanism from the theoretical one is slight and that this deviation is due mainly to the nature of real fluid flow in pipes, i.e., head loss, velocity profile change, which is only slightly influenced by the hydraulic shape of the device.

For Group A tubes, therefore,

$$\Delta H_a = \Delta H_{th} + H_L - (\alpha_D - 1)\frac{V_D^2}{2g} + (\alpha_d - 1)\frac{V_d^2}{2g} \qquad (10)$$

where

H_L = head loss, occurring in the tube between inlet and throat tap, consequently, influenced by the hydraulic shape of the device, but only slightly

$$\alpha_D = \frac{\text{actual kinetic energy content at inlet tap cross section}}{\frac{V_D^2}{2g}}$$

This value is uninfluenced by the hydraulic shape of the device,

$$\alpha_d = \frac{\text{actual kinetic energy content at throat tap cross section}}{\frac{V_d^2}{2g}}$$

For Group B, modified venturi tubes, the empirical C-value, according to Figure 1–2a, of these devices is about the same for lower beta ratios (below 0.5) as for that of Group A tubes, but it is beta sensitive.

The cause of this behavior lies in the fact that the hydraulic shape of these devices is the same as that of Group A tubes, except the entrance section is eliminated by locating the inlet tap in the corner as shown in Figure 1–1.

This means that Condition 1 is violated at the inlet pressure tap sensation, inasmuch as the geometrical diameter at the inlet tap does not represent a normal flow diameter. Furthermore, the inlet tap does not sense true static pressure since it is in a region where the direction of flow is changed by the hydraulic shape of the device. Equation (10) should be modified, therefore, thus:

$$\Delta H_a = \Delta H_{th} + H_L + [-(\alpha_D - 1) + Z_B] \frac{V_D^2}{2g} + (\alpha_d - 1) \frac{V_d^2}{2g} \tag{11}$$

where Z_B is a portion of the inlet velocity head converted into apparent static pressure by the corner tap. Since Z_B is nearly independent of beta, its effect in terms of differential changes with beta (Figure 1–4) renders the C of these devices beta-sensitive. It should be noted, however, that this beta sensitivity is relatively slight, about 5 percent on C or 10 percent on differential at the highest beta [Equations (8) and (9)] representing a small violation of the ideal venturi metering principle.

For Group C, flow tubes, the C-value is much lower than 1 and is strongly dependent on the beta ratio (Figure 1–2a).

The causes of this behavior are related to the hydraulic shape of these devices and are as follows.

Condition 1 is drastically violated both at the inlet and throat tap regions because: (a) the cross sections of the flow at the inlet and throat taps are not the same as the geometrical areas, and (b) the inlet and throat taps are located in regions where the direction of flow is changed by the hydraulic shape of the device. Consequently, the pressures sensed are not true static pressures.

Condition 2 is violated especially at the throat due to the shape and/or steep transition section of the device.

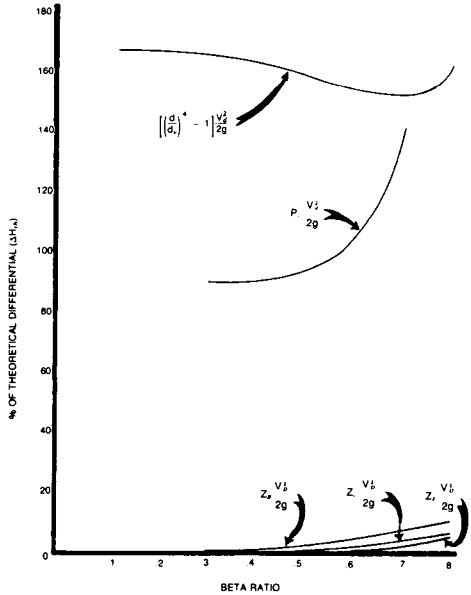


Figure 1–4. Relationship between kinetic energy and beta ratio.

The effect of deviation from Condition 3 here is small in comparison to the effect of the violation of Conditions 1 and 2. These factors lead to restatement of the design equation to

$$\Delta H_a = \Delta H_{th} + H_L + \left[-(\alpha_D - 1) + Z_C \frac{V_D^2}{2g} \right] + [(\alpha_d - 1) + P_c] \frac{V_d^2}{2g} \tag{12}$$

where

Z_c = portion of inlet velocity head (Pitot effect) converted into apparent static pressure by the corner tap

P_c = portion of the throat velocity head lowering the "normal static pressure," partly due to the vena contracta formed in the region of the throat tap providing a flowing cross section smaller than the actual throat area (Figure 1–2b) and partly due to the fact that the hydraulic shape of the tube "shoots" the flow at an angle towards the throat tap creating a suction effect (negative Pitot effect)

Z_c and P_c are nearly independent of beta, as are their effect in terms of differential changes with beta. This makes the *C*'s of these devices beta-sensitive