2.29 Venturi Tubes, Flow Tubes, and Flow Nozzles


Design Types
A. Venturi tubes
B. Flow tubes
C. Flow nozzles

Design Pressure
Usually limited only by transmitter, readout device or by pipe pressure ratings

Design Temperature
Limited only by readout device, if operation is at very low or high temperature

Sizes
A. 1 in. (25 mm) up to 120 in. (3000 mm)
B. 4 in. (100 mm) up to 48 in. (1200 mm)
C. 1 in. (25 mm) up to 60 in. (1500 mm)

Fluids
Liquids, gases, and steam

Flow Range
Limited only by minimum and maximum beta (β) ratio and available pipe size range

Inaccuracy
Values given are for flow elements only; d/p cell and readout errors are additional
A. ±0.75% of rate uncalibrated to ±0.25% of rate calibrated in a flow laboratory
B. May range from ±0.5 to ±3% of rate depending on the particular design and variations in fluid operating conditions
C. ±1% of rate uncalibrated to ±0.25% calibrated

Materials of Construction
Virtually unlimited. Cast venturi tubes are usually cast iron, but fabricated venturi tubes can be made from carbon steel, stainless steel, most available alloys, and fiberglass plastic composites.
Flow nozzles are commonly made from alloy steel and stainless steel.

Pressure Recovery
Ninety percent of the pressure loss is recovered by a low-loss venturi when the beta (β) ratio is 0.3, whereas an orifice plate recovers only 12%. (The corresponding energy savings in a 24-in. [600-mm] waterline is about 20 HP.)

Reynolds Numbers
Venturi and flow tube discharge coefficients are constant at Re > 100,000. Flow nozzles are used at high pipeline velocities (100 ft/s or 30.5 m/s), usually corresponding to Re > 5 million. Critical flow venturi nozzles operate under choked conditions at sonic velocity.

Costs
Flow nozzles are less expensive than venturi or flow tubes, but cost more than orifices.
ASME gas flow nozzles in aluminum for 3- to 8-in. (75- to 200-mm) lines cost from $300 to $1000. Epoxy-fiberglass nozzles for 12- to 32-in. (300- to 812-mm) lines cost from $1000 to $3000. The relative costs of Herschel venturies and flow tubes in different sizes and materials are given below:

<table>
<thead>
<tr>
<th></th>
<th>6-in. Stainless Steel</th>
<th>8-in. Cast Iron</th>
<th>12-in. Steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Herschel venturi</td>
<td>$9000</td>
<td>$6000</td>
<td>$6500</td>
</tr>
<tr>
<td>Flow tube</td>
<td>$4000</td>
<td>$2500</td>
<td>$3200</td>
</tr>
</tbody>
</table>

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Venturi tubes, flow nozzles, and flow tubes, like all differential pressure producers, are based on Bernoulli’s theorem. General performance and calculations are similar to those for orifice plates. In these devices, however, there is continuous contact between the fluid flow and the surface of the primary device, in contrast to the pure line contact between the orifice plate edge and main flow. The surface finish of the devices can have some effect on the meter coefficient, although the venturi tube has a relatively constant coefficient, seldom varying more than a fraction of 1%. Modern precision manufacturing techniques allow much greater accuracy of the coefficient for venturi tubes and flow nozzles computed from dimensions, and the coefficients are only moderately less reliable than those for orifice plates. The C (meter coefficient) values for venturi tubes and flow nozzles have been well established by years of test data and are tabulated in sources such as the handbook called Fluid Meters—Their Theory and Application. In general, this is not true of the proprietary flow tubes, and flow calibration is required to establish the actual meter coefficient. Meter coefficients for venturi tubes and flow nozzles are approximately 0.98 to 0.99 and for orifice plates average about 0.62. Therefore, almost 60% (98/62) more flow can be obtained through these elements for the same differential pressure.

THE CLASSIC VENTURI

The venturi tube, as designed by Clemens Herschel in 1887 and described in Reference 1, is shown in Figure 2.29a. It consists of

- A cylindrical inlet section equal to the pipe diameter
- A converging conical section in which the cross-sectional area decreases, causing the velocity to increase with a corresponding increase in the velocity head and a decrease in the pressure head
- A cylindrical throat section where the velocity is constant so the decreased pressure head can be measured
- A diverging recovery cone where the velocity decreases and almost all of the original pressure head is recovered

The unrecovered pressure head is commonly called head loss.

The classic venturi is always manufactured with a cast-iron body and a bronze or stainless-steel throat section. At the midpoint of the throat, six to eight pressure taps connect the throat to an annular chamber so that the throat pressure is averaged. The cross-sectional area of the chamber is 1.5 times the cross-sectional area of the taps. Because there is no movement of fluid in the annular chamber, the pressure sensed is strictly static pressure. Usually, four taps from the external surface of the venturi into the annular chamber are made. These are offset from the internal pressure taps. Throat pressure is measured through these taps. This flow meter is limited to use on clean, noncorrosive liquids and gases, because it is impossible to clean out or flush out the pressure taps if they clog up with dirt or debris. The flow coefficient for the classic venturi is 0.984, with an uncertainty tolerance of ±0.75%.

SHORT-FORM VENTURIES

In the 1950s, in an effort to reduce costs and laying length, manufacturers developed the second-generation, or short-form, venturi shown in Figure 2.29b. There were two major differences in this design. The internal annular chamber was replaced by a single pressure tap or, in some cases, an external pressure averaging chamber, and the recovery cone angle was increased from 7° to 21°. The short-form venturi can be manufactured from cast iron or welded from a variety of materials as compatible with a given application. The flow coefficient for the short-form venturi is 0.985, with an uncertainty tolerance of ±1.5%.

The pressure taps are located one-quarter to one-half pipe diameter upstream of the inlet cone and at the middle of the throat section. A piezometer ring is sometimes used for differential pressure measurement. This consists of several holes
in the plane of the tap locations. Each set of holes is connected in an annulus ring to give an average pressure. Venturis with piezometer connections are unsuitable for use with purge systems used for slurries and dirty fluids, because the purging fluid tends to short circuit to the nearest tap holes. Piezometer connections are normally used only on very large tubes or where the most accurate average pressure is desired to compensate for variations in the hydraulic profile of the flowing fluid. Therefore, when it is necessary to meter dirty fluids and use piezometer taps, sealed sensors that mount flush with the pipe and throat inside wall should be used. These sensors function as independent measuring devices at each tap connection, yet they function together to read differential pressure only while automatically compensating for static pressure changes within the pipe. Single-pressure-tap venturis can be purged in the normal manner when used with dirty fluids. Because the venturi tube has no sudden changes in contour, no sharp corners, and no projections or stagnant areas, it is often used to measure slurries and dirty fluids that tend to build up on or clog other primary devices.

Venturis are built in several forms. These include the standard long-form or classic venturi (Figure 2.29a), a modified short form where the outlet cone is shortened (Figure 2.29b), an eccentric form (Figure 2.29c) to handle mixed phases or to minimize buildup of heavy materials, and a rectangular form (Figure 2.29d) used in ductwork. If a rectangular venturi is substantially square, it is customary to converge–diverge all four sides with angles the same as for the circular form. Where duct width differs from height, the short sides are kept parallel, with the long sides converging–diverging. A converging angle of 21° and a diverging angle of 15° give satisfactory operation. Throat length should be equal to minimum throat height or width, whichever is smaller. Tap locations are the same as for the circular form.

The angle of convergence, which may range from 19 to 23°, is the classical value established by Herschel in 1887. This angle is not particularly critical, and 21 ± 1° is commonly used. The recovery cone provides pressure recovery with its smooth flow transition. The classic long cone form is 7.5° ± 0.5° on the divergence, but up to 15° is allowed, and the sharper angle allows the short-form version to be fabricated. The 15° outlet cone sacrifices a modest amount of pressure recovery (Figure 2.29e). The venturi pressure loss of 10 to 25% is the lowest of the standard primary head measurement elements. The long-cone form develops up to 89% pressure recovery at 0.75β ratio, decreasing to 86% at 0.25β ratio. The short-cone form develops up to 85% recovery at 0.75β, decreasing to 75% at 0.25β ratio. As an example of the power savings to be obtained in an energy-short era, an added pressure recovery of 50 in. (1270 mm) H₂O differential pressure can represent a 10-HP savings in a 24-in. (610-mm) water line flowing at a velocity of 6 ft/s (1.829 m/s). For a comparison of various head-meter elements from the pressure recovery point of view, see Figure 2.29f.
Installation

A venturi tube may be installed in any position to suit the requirements of the application and piping. The only limitation is that, with liquids, the venturi is always full. In most cases, the valved pressure taps will follow the same installation guidelines as for oriﬁce plates.

Upstream piping should be as long as needed to provide a proper velocity proﬁle (Figure 2.29g). However, in most installations, shorter upstream piping is required than for orifices, nozzles, or pitot tubes, because the venturi hydraulic shape itself provides some flow conditioning. Often, the combined length of a venturi and its upstream piping is less than the overall amount of piping required for an oriﬁce or nozzle. Figure 2.29h shows typical upstream pipe diameters required for various elements at 0.7β ratio and one elbow upstream. Straightening vanes can be used upstream to reduce the inlet pipe length.

In Fluid Meters,¹ the ASME recommends the use of tubular straightening vanes (19 tubes and 2 diameters long) upstream of the venturi to reduce the inlet pipe length. The vane installation should have a minimum of 2 diameters upstream and 2 diameters downstream before entering the venturi.

There is no limitation on piping conﬁguration downstream of the venturi except that a valve should be no closer than two diameters. Valves on other devices that protrude into the ﬂow stream should not be mounted upstream of the venturi, if possible.

Flow Calculations

The American Society of Mechanical Engineers Fluid Research Committee has adopted a general coefﬁcient of discharge of 0.984 for the classic rough-cast entrance cone venturi tube from 4 in. (100 mm) through 32 in. (813 mm) and for β ratios between 0.3 and 0.75. For tubes with machined entrance cones, the general coefﬁcient is 0.995. The Reynolds number must be 200,000 or greater. Approximate ﬂow rates can be calculated from a working equation,

\[
W = 353d^2 \sqrt{\frac{hp}{1 - \beta^2}}
\]

2.29(1)
and, for approximate venturi tube design,

\[
\beta = \frac{1}{\sqrt[4]{1 + 125,000 \frac{h \rho}{W^4}}}
\]

For Reynolds numbers between 50,000 and 200,000, substitute 344 for 353. Below 50,000, reliable data are not available. It should be noted that, in contrast to an orifice, a decrease in Reynolds number results in a decrease of expansion factor, and so on, are similar to those for orifices and flow nozzles.

**FLOW TUBES**

There are several proprietary primary-head-type devices that have a higher ratio of pressure developed to pressure lost than a venturi tube (Figure 2.29i). They are all considerably more compact than the classical venturi tube, with its long recovery cone, although the short-from venturi can come close to some types of these tubes.

These designs are available in cast iron, can be welded from various materials, and in some cases can have insert-type units in fiberglass-reinforced plastic or metal. The flow coefficient ranges from 0.9797 for an all-static-tap “near venturi” design to 0.75 for an all-corner-tap “flow tube” design. All of these proprietary units are available in the United States except the Dall tube, which was developed in England. All of these tubes vary in contour used, tap locations, and differential pressure and pressure loss for a given flow. All have a laying length less than 4 diameters long. The shortest are the corner tap designs, with lengths equaling 2 to 2.5 diameters.

A flow tube is broadly defined by the ASME as any differential-pressure-producing primary whose design differs from the classic venturi. Flow tubes fall into three main classes, depending on the hydraulic position of the inlet and throat pressure tap. Type 1 has static pressure taps at both the inlet and outlet, Type 2 has a corner tap in the inlet and a static tap in the throat, and Type 3 has a corner tap at both the inlet and outlet.

The classic venturi had static pressure taps that provided a section in which the velocity is not changing direction and is parallel to the pipe wall. A corner tap senses pressure in a section where the velocity is changing direction and is not parallel to the pipe wall. Figure 2.29i shows examples of several flow tubes.

Type 3 flow tubes can be useful in larger sizes because of their shorter lay length, but they may also require longer upstream pipe runs for proper performance. They can be subject to coefficient change due to variations in Reynolds number, line size, and beta ratio; manufacturers can provide data on these effects.

The B.I.F. Universal Venturi is the product (Type 1) that most closely approaches the Herschel design classic venturi. The inlet cone has two vena contracta angles that condition the fluid as it enters the throat. This is claimed to reduce the sensitivity to upstream piping configuration and give higher accuracy. Also claimed are a stable coefficient (0.9797) that is unaffected by internal surface roughness, lower Reynolds number application (90,000), low head loss (4 to 18%), and extensive documentation including expansion factors.

Whereas flow tubes can be useful in larger sizes because of their shorter lay length, they may also require longer upstream pipe runs than the venturi for proper performance and thus lose any real advantage. They can be subject to coefficient change with viscosity and Reynolds number; manufacturers can provide data on these effects. None has the smooth contour and resistance to clogging of the venturi meter; however, some are claimed to operate satisfactorily on wastewater and sewage flow measurement.

In general, these devices are available in 4-in. (100-mm) and larger sizes up to 48 in. (1219 mm). There is little justification for their use in small-flow, small-pipe applications. In the larger sizes, their installed cost may be less than that of the venturi tube. Accuracy depends basically on the manufacturer’s calibration data. Derivation of the flow coefficient by extrapolation from theory and tests on smaller sizes is much less direct than in the simple structure of the venturi tube; actual flow calibration, particularly in sizes above 24 in. (610 mm), can be difficult and expensive. Although these devices generally have a better pressure recovery than the venturi (expressed as a percentage of the differential), most flow tubes have a lower coefficient of discharge (less efficient). As a result, there is often very little difference in the actual head loss.

**FIG. 2.29i**

Proprietary flow tubes.
In selecting a primary flow element, the possible advantages of slightly lower pressure loss and shorter laying length of the flow tubes should be carefully weighed against the metering accuracy and established flow data available on the Herschel-form venturi. The ASME recommends that, if a proprietary flow tube is used, it should be calibrated with the piping section in which it is to be used and over the full range of flows to which it will be subjected. The only possible exception to this is the Universal Venturi, and it must be carefully evaluated. The background of extensive tests under a wide range of conditions that support orifice meters does not exist for these proprietary devices.

**FLOW NOZZLES**

There are two types of flow nozzles. The 1932 ISA nozzle is a European design that has not seen use in the United States. A special variation known as a venturi nozzle is a hybrid combination of a 1932 ISA nozzle inlet profile combined with the divergent cone of a venturi tube. The common nozzle used in the United States is the so-called long-radius or ASME flow nozzle. This nozzle comes in two versions, known as low-beta-ratio and high-beta-ratio designs. This flow nozzle, shown in Figure 2.29j, is a metering primary whose shape consists of a quarter ellipse convergence section and a cylindrical throat section. In the United States, the nozzle generally used is the long-radius ASME flow nozzle. The ASME Fluid Meters Research Committee has investigated various configurations and has developed the geometry for these nozzles based on the required beta ratio for the application. High-beta nozzles are recommended for diameter ratios between 0.45 and 0.80. Low-beta nozzles are recommended for diameter ratios between 0.20 and 0.50. For beta values between 0.25 and 0.5, either design may be used.

The difference between the two nozzles is basically a flattening of the ellipse in the high-beta-ratio version. The power test code, PTC-6, requires that the low-beta-ratio version be used in their test section for turbine acceptance.

Both types of nozzles may be either welded in the pipeline or provided with a holding ring for mounting between flanges. The latter design, shown in Figure 2.29k, is preferred when frequent inspection of the nozzle is required.

Nozzles may be manufactured from any material that can be machined; typically, they are fabricated from aluminum, fiberglass, stainless steel, or chrome-moly steel. Modern manufacturing methods and fluid contact surface finishes on the order of 6 to 10 μin result in more predictable nozzle coefficients and highly repeatable data. The standard surface finish is 16 RMS. Flow nozzle inaccuracy of ±1% is standard with ±0.25% flow calibrated. The standard coefficient, as published in Reference 1, is 0.9962 with correction factors for beta ratio and throat Reynolds number. ASME gives an uncertainty of ±2% for nozzles having a beta ratio between 0.2 and 0.8 and throat Reynolds numbers between $1 \times 10^5$ and $2.5 \times 10^6$.

**FIG. 2.29j**

ASME nozzle construction.
Most ASME nozzles are calibrated as meter sections 20 pipe diameters long. As with venturi tubes, the uncertainty of calibrated units depends on the uncertainty of the hydraulic laboratory. Generally, one could expect ±0.25% uncertainty on the calibration.

The outlet or discharge side of the nozzle is normally beveled and is one of the more critical points of manufacture. Where the 10° back angle meets the throat bore, the edge must be sharp. Particular care must be taken to avoid taper and out-of-roundness of the throat.

Flow nozzles are made in various configurations. The most common is the flange type (Figure 2.29k), but others are the holding-ring type, the weld-in type, and the throat-tap type. Differential pressure measurement taps are commonly located one pipe diameter upstream and one-half pipe diameter downstream from the inlet face (U.S. practice), except for the throat-tap type, which has a special downstream construction. The PTC-6 nozzle uses throat taps to sense the low pressure and standard pipe wall taps for the high pressure.

**Application Considerations**

Tap installation precautions are the same as for orifice plates. The preferred installation position for flow nozzles is horizontal, but they can be installed in any position. However, a vertical downflow position is preferred for wet steam or for gases and liquids with suspended solids. In general, upstream and downstream piping requirements are similar to those required for orifices. Because of the width, nozzles installed between flanges are difficult to remove. Common practice is to provide a flange in the downstream piping to allow the nozzle to be removed as part of a spool section for inspection at regular intervals. Sometimes, inspection openings are placed just upstream of the nozzle so that frequent inspections can be made without removing the nozzle from service.

Flow nozzles are particularly suited for measurement of steam flow and other high-velocity fluids, fluids with some solids, wet gases, and similar materials. Because the exact contour is not critical, the flow nozzle can be expected to retain good calibration for a long time under erosion or other hostile conditions. Because of its streamlined contour, it tends to sweep solids or moisture through the throat and is far superior to orifice plates in these services. Because tap geometry and contour is critical to maintaining calibration, it is not recommended to use flow nozzles on slurries or dirty fluids.

A flow nozzle will pass about 60% more flow than an orifice plate of the same diameter and differential pressure. It also has the advantage of operating acceptably over a wide beta ratio range of 0.2 to 0.8. For the same flow and differential pressure, the flow nozzle has a similar but slightly lower pressure loss than an orifice plate. This becomes apparent when it is recognized that the area of the throat and the velocity in the throat of a flow nozzle must be approximately the same as the area of flow and velocity at the vena contracta following an orifice so as to develop the same differential pressure from the same flow. The slightly lower pressure loss of the flow nozzle is due to its streamlined entrance.

On the other hand, because the ASME flow nozzle does not utilize a recovery cone, the permanent head loss can still be as much as 40% of the differential pressure (Figure 2.29f). In an effort to reduce these losses, particularly in applications for PTC-6 testing of turbines, a recovery cone may be added. In this design, the permanent head losses can be substantially reduced. The actual amount of reduction should be determined through testing.

Although nozzles should be used at Reynolds numbers of 50,000 or above, data are available for Re down to 6000, so it is possible to use nozzles with more viscous fluids. Still, work published by the ASME Fluid Research Committee suggests that the most stable flow coefficients are seen at throat Reynolds number of $1 \times 10^5$. For throat Reynolds numbers below $1 \times 10^5$, the shift in flow coefficient can be as high as 6%.

Flow nozzles have very high coefficients of discharge, typically 0.99 or greater. Using a typical value of 0.993, approximate flow rates can be calculated from a working equation,

$$ W = 358d^2 \sqrt{\frac{hp}{1-\beta^4}} $$.  \hspace{1cm} 2.29(3)

and, for approximate flow nozzle design,

$$ \beta = \frac{1}{\sqrt{4+\frac{128,000hpD^4}{W^2}}} $$.  \hspace{1cm} 2.29(4)

**Critical-velocity Venturi Nozzles**

One of the most accurate ways to measure gas flow is to cause “choked” flow (sonic velocity flow) through a venturi nozzle. Critical-velocity venturi nozzles are also used as secondary flow standards in calibrating other flowmeters. The nozzle can
be the ASME long-radius, elliptical inlet, wall-tap nozzle; the ISA 1932 nozzle; or the ASME throat-tap nozzle used in steam turbine testing.

One of the highest rangeability and most accurate gas flowmeters has been devised by combining the sonic venturi nozzles with the digital control valve (Figure 2.1j).

**ACCURACY**

Operation and calibration of venturi tubes over a period of many years has resulted in extensive documentation. As a result, most manufacturers will guarantee a standard design inaccuracy of ±0.75% of actual flow. This can be reduced to ±0.25% by calibration at a recognized hydraulics laboratory. Modern manufacturing techniques have led to predictable discharge coefficients and a repeatability of ±0.2% for venturi tubes of the same size and design.

For very small (<4 in. or 100 mm) and very large (>32 in. or 813 mm) venturi tubes, and for very high (>2,000,000) or very low (<150,000) Reynolds numbers, flow calculations for venturi tubes have about a 50% greater uncertainty than a corresponding sharp-edged orifice plate. However, fluid flow calibration, particularly when made under conditions closely approximating service values, can provide a coefficient with practically the same accuracy as that of the calibration facilities.

The error contribution of the d/p-generating flow sensor is defined as the uncertainty tolerance of the flow coefficient. The inaccuracy values can range from as low as 0.25% of rate for calibrated units to 1.5% of rate for uncalibrated welded units and can be expected to hold true only for a limited range of Reynolds numbers (see Figure 2.1f) and beta ratios.

The overall performance of the total flow measurement system therefore will be the sum of the transmitter and sensor errors. This sum will hold true only over the flow range between the maximum flow and the flow rate corresponding to the minimum Reynolds number for which the sensor error is still guaranteed. This minimum Reynolds number for venturies and flow tubes is around 100,000, and for flow nozzles it is over 1,000,000. Consequently, the rangeability of these devices, if defined in terms of actual flow error, can be rather low.

**DIFFERENTIAL PRESSURE MEASUREMENT**

The differential pressure generated by these primary devices (venturi, flow tubes, and flow nozzles) can be measured by manometers, gauges, and electronic pressure transmitters. The accuracy of analog electronic transmitters varies from ±0.1 to ±0.5% of calibrated span. When square-root circuitry is added, there is usually a ±0.05% increase in the error. Microprocessor-based (“smart”) transmitters have an inherent accuracy of ±0.1% of calibrated span or less, regardless of whether the square-root function is used.

Overall accuracy of the entire flowmeter system is a function of the transmitter and other instruments in the loop. The most common method for determining accuracy is to root mean square (RMS) the errors to calculate the total error. Let’s look at an example with only the primary device and an electronic transmitter. Recognize that transmitter accuracy is specified as a percent of calibrated span. Thus, at 25% of span, the error will be four times the error at full scale. The venturi, flow tube, and flow nozzle accuracy is specified as a percent of rate. Thus, the sensor accuracy is the same throughout its usable range.

Assume the venturi and the electronic transmitter are both set up such that there is a 100-in. H2O differential pressure at full scale flow. Assume that the venturi has an accuracy of 0.75% of rate, and the transmitter has an accuracy of 0.25% of calibrated span. At full-scale flow, the total RMS uncertainty will be \( \sqrt{0.75^2 + 0.25^2} = 0.79\% \). At 50% flow, the total RMS uncertainty will be \( \sqrt{0.75^2 + 0.50^2} = 0.90\% \). At 25% of flow, the total RMS uncertainty will be \( \sqrt{0.75^2 + 0.75^2} = 1.25\% \). At 10% flow, the total RMS uncertainty will be \( \sqrt{0.75^2 + 1.50^2} = 2.61\% \). Therefore, the transmitter can contribute significantly to the total error of the system when used over a wide range, even though the primary device maintains its accuracy over that range.

Some manufacturers of smart transmitters have routines that reduce the full-scale value of the transmitter as the differential pressure signal from the primary decreases so as to increase the total accuracy. This requires that the transmitter communicate digitally to the receiver so the reduced full scale and the measured differential pressure can be transmitted to the receiver. The total error in a flow measurement is the sum of two errors: that of the sensor and that of the transmitting or readout device. The error contribution of the best d/p transmitters is about 0.1% of span. To cover a flow range of 10:1, d/p range of 100:1 needs to be covered, which requires either an extremely wide-range d/p cell or, more likely, two d/p transmitters (a high-span and a low-span one). If such a dual-transmitter configuration is used, and if the transmitters are switched as needed, the actual error contribution of the transmitter can be limited to 1% of actual flow.

**CONCLUSION**

The main limitation of venturi tubes is cost, both of the tube itself and often of the piping layout required for the length necessary in the larger sizes. However, the energy-cost savings attributable to venturi tubes’ higher pressure recovery and reduced pressure loss usually justifies their use in larger pipes.
Another limitation is the relatively high minimum Reynolds number required to maintain accuracy. For venturies and flow tubes, this minimum is around 100,000, whereas, for flow nozzles, it is greater than 1,000,000. Naturally, correction data are available for Reynolds numbers below these limits, but measurement performance will suffer.

Cavitation can also be a problem. At the high flow velocities (corresponding to the required high Reynolds numbers) at the vena contracta, the static pressure will be low. When it drops below the vapor pressure of the flowing fluid, cavitation occurs. This, if present, will destroy the throat section of the tube, as no material can stand up to cavitation. The possible ways to eliminate cavitation include relocating the meter to a point in the process where the pressure is higher and the temperature is lower, reducing the pressure drop across the sensor, and replacing the sensor with one that has less pressure recovery.

As a result of their construction, venturies, flow tubes, and flow nozzles are relatively difficult to inspect. This problem can be solved by providing an inspection port on the outlet cone near the throat section. This can be an important factor when metering dirty (erosive) gases, slurries, or corrosive fluids. On dirty services where the pressure ports are likely to plug, the pressure taps on the flow tube can be filled with chemical seals having stainless-steel diaphragms that are installed flush with the tube interior (Figure 2.29l).

The main advantages of these sensors include their relatively high accuracy, good rangeability (on high Reynolds number applications), and energy-conserving high-pressure recovery. For these reasons, in higher-velocity flows and in larger pipelines (and ducts), the venturies are still favored by many users in spite of their high costs. Their hydraulic shape also contributes to greater dimensional reliability and therefore to better flow-coefficient stability than that of the orifice-type sensors, which depend on the sharp edge of the orifice for their flow coefficient.

The accuracy of a flow sensor is defined as the uncertainty tolerance of the flow coefficient. Accuracy can be improved by calibration. Table 2.29m gives some accuracy data in percentage of actual flow as reported by various manufacturers. These values are likely to hold true only for the stated ranges of beta ratios and Reynolds numbers, and they do not include the added error of the readout device or d/p transmitter.

![FIG. 2.29l](image_url)

*A variable capacitance flow transmitter can be mounted integrally to the flow tube and provided with chemical seals for protection against plugging or corrosion. (Courtesy of BIF Inc.)*

<table>
<thead>
<tr>
<th>Flow Sensor</th>
<th>Line size, inches (1 in. = 25.4 mm)</th>
<th>Beta Ratio</th>
<th>Pipe Reynolds Number Range for Stated Accuracy</th>
<th>Inaccuracy, Percent of Actual Flow</th>
</tr>
</thead>
<tbody>
<tr>
<td>Herschel standard Cast¹</td>
<td>4–32</td>
<td>0.30–0.75</td>
<td>$2 \times 10^6$ to $1 \times 10^8$</td>
<td>±0.75%</td>
</tr>
<tr>
<td>Herschel standard Welded</td>
<td>8–48</td>
<td>0.40–0.70</td>
<td>$2 \times 10^6$ to $2 \times 10^8$</td>
<td>±1.5%</td>
</tr>
<tr>
<td>Proprietary true venturi</td>
<td>Cast¹</td>
<td>0.30–0.75</td>
<td>$8 \times 10^4$ to $8 \times 10^6$</td>
<td>±0.5%</td>
</tr>
<tr>
<td>Proprietary true venturi</td>
<td>Welded</td>
<td>0.25–0.80</td>
<td>$8 \times 10^4$ to $8 \times 10^6$</td>
<td>±1.0%</td>
</tr>
<tr>
<td>Proprietary flow tube</td>
<td>Cast¹</td>
<td>0.35–0.85</td>
<td>$8 \times 10^4$ to $1 \times 10^6$</td>
<td>±1.0%</td>
</tr>
<tr>
<td>ASME flow nozzles³</td>
<td>1–48</td>
<td>0.20–0.80</td>
<td>$7 \times 10^4$ to $4 \times 10^4$</td>
<td>±1.0%</td>
</tr>
</tbody>
</table>

¹No longer manufactured because of long laying length and high cost.
²Badger Meter Inc.; BIF Products; Fluidic Techniques Inc.; Primary Flow Signal Inc.; Tri-Flow Inc.
³ABB Instrumentation; Badger Meter Inc.; BIF Products; Preso Industries.
⁴BIF Products; Daniel Measurement and Control.
2.29 Venturi Tubes, Flow Tubes, and Flow Nozzles

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