TECHNICAL REPORT

ISO/TR 3313

Fourth edition
2018-03

Measurement of fluid flow in closed conduits — Guidelines on the effects of flow pulsations on flow-measurement instruments

Mesurage du débit des fluides dans les conduites fermées — Lignes directrices relatives aux effets des pulsations d'écoulement sur les instruments de mesure de débit

Reference number
ISO/TR 3313:2018(E)

ISO copyright office
CP 401 • Ch. de Blandonnet 8
CH-1214 Vernier, Geneva
Phone: +41 22 749 01 11
Fax: +41 22 749 09 47
Email: copyright@iso.org
Website: www.iso.org

Published in Switzerland

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Foreword

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This document was prepared by Technical Committee ISO/TC 30, *Measurement of fluid flow in closed conduits*, Subcommittee SC 2, *Pressure differential devices*.

This fourth edition of ISO/TR 3313:2018 is a technical revision of ISO/TR 3313:1998, which was withdrawn in 2013.

Measurement of fluid flow in closed conduits — Guidelines on the effects of flow pulsations on flow-measurement instruments

1 Scope

This document defines pulsating flow, compares it with steady flow, indicates how it can be detected, and describes the effects it has on orifice plates, nozzles or Venturi tubes, turbine and vortex flowmeters when these devices are being used to measure fluid flow in a pipe. These particular flowmeter types feature in this document because they are amongst those types most susceptible to pulsation effects. Methods for correcting the flowmeter output signal for errors produced by these effects are described for those flowmeter types for which this is possible. When correction is not possible, measures to avoid or reduce the problem are indicated. Such measures include the installation of pulsation damping devices and/or choice of a flowmeter type which is less susceptible to pulsation effects.

This document applies to flow in which the pulsations are generated at a single source which is situated either upstream or downstream of the primary element of the flowmeter. Its applicability is restricted to conditions where the flow direction does not reverse in the measuring section but there is no restriction on the waveform of the flow pulsation. The recommendations within this document apply to both liquid and gas flows although with the latter the validity might be restricted to gas flows in which the density changes in the measuring section are small as indicated for the particular type of flowmeter under discussion.

2 Normative references

There are no normative references in this document.

3 Terms and definitions

For the purposes of this document, the following terms and definitions apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at http://www.electropedia.org/

3.1
steady flow
flow in which parameters such as velocity, pressure, density and temperature do not vary significantly enough with time to prevent measurement to within the required uncertainty of measurement

3.2
pulsating flow
flow in which the flowrate in a measuring section is a function of time but has a constant mean value when averaged over a sufficiently long period of time, which depends on the regularity of the pulsation

Note 1 to entry: Pulsating flow can be divided into two categories:

— periodic pulsating flow;

— randomly fluctuating flow.

Note 2 to entry: For further amplification of what constitutes steady or pulsating flow see 5.1 and 5.2.

Note 3 to entry: Unless otherwise stated in this document the term “pulsating flow” is always used to describe periodic pulsating flow.

4 Symbols and subscripts

4.1 Symbols

A	area
A_d	area of the throat of a Venturi nozzle
A_R	turbine blade aspect ratio
a_r, b_r, c_r	amplitude of the r^{th} harmonic component in the undamped or damped pulsation
B	bf_p/q_V, dimensionless dynamic response parameter
b	turbine flowmeter dynamic response parameter
C	turbine blade chord length
C_c	contraction coefficient
C_D	discharge coefficient
C_v	velocity coefficient
c	speed of sound
D	internal diameter of the tube
d	throat bore of orifice, nozzle or Venturi tube
E_R	residual error in time-mean flowrate when calculated using the quantity $\sqrt{\Delta p}$
E_T	total error in the time-mean flowrate
f	turbine flowmeter output signal, proportional to volumetric flowrate
f_p	pulsation frequency
f_r	resonant frequency
f_v	vortex-shedding frequency
H	harmonic distortion factor
Ho	Hodgson number
I	moment of inertia
I_R, I_F	moments of inertia of turbine rotor and fluid contained in rotor envelope respectively
k/D	relative roughness of pipe wall
L	turbine blade length
L_e	effective axial length
l	impulse line length for differential pressure (DP) measurement device

$m = \beta^2$	orifice or nozzle throat to pipe area ratio
N	number of blades on turbine rotor
p	pressure (absolute)
q_m	mass flowrate
q_V	volume flowrate
R	turbine blade mean radius
Re	Reynolds number
r_h, r_t	turbine blade hub and tip radii respectively
Sr	Strouhal number
Sr_d	Strouhal number based on orifice diameter
t	time
t_b	turbine blade thickness
U	axial bulk-mean velocity
U_d	bulk-mean velocity based on orifice diameter
V	volume
X	temporal inertia term for short pulsation wavelengths
α	$U'_{RMS} / \overline{U}$
β	orifice or nozzle throat to pipe diameter ratio
γ	ratio of specific heat capacities (c_p/c_V)
Δp	differential pressure
$\Delta\varpi$	pressure loss
ε_{ss}	expansibility factor for steady flow conditions
η	blade “airfoil efficiency”
θ	phase angle
κ	isentropic exponent (= γ for a perfect gas)
μ	damping response factor (see 6.1.4.1.3)
ρ	fluid density
ρ_b	turbine blade material density
$\tau = p_2/p_1$	pressure ratio

φ	maximum allowable uncertainty in the indicated flowrate due to pulsation at the flowmeter
ψ	maximum allowable relative error
$\omega = 2\pi f_p$	angular pulsation frequency

4.2 Subscripts and superscripts

o	pulsation source
p	measured under pulsating flow conditions, possibly damped
po	measured under pulsating flow conditions before damping
RMS	root mean square
ss	measured under steady flow conditions
$\bar{\ }$ (over-bar)	the time-mean value
1,2	measuring sections
'	fluctuating component about mean value, e.g. U'

5 Description and detection of pulsating flow

5.1 Nature of pipe flows

Truly steady pipe flow is only found in laminar flow conditions which can normally only exist when the pipe Reynolds number, *Re*, is below about 2 000. Most industrial pipe flows have higher Reynolds numbers and are turbulent which means that they are only statistically steady. Such flows contain continual irregular and random fluctuations in quantities such as velocity, pressure and temperature. Nevertheless, if the conditions are similar to those which are typical of fully developed turbulent pipe flow and there is no periodic pulsation, the provisions of such standards as ISO 5167 (all parts) apply.

The magnitude of the turbulent fluctuations increases with pipe roughness, and this is one of the reasons why ISO 5167 (all parts) stipulates a maximum allowable relative roughness, k/D, of the upstream pipe for each type of primary device covered by ISO 5167 (all parts).

ISO 5167 (all parts), however, cannot be applied to flows which contain any periodic flow variation or pulsation.

5.2 Threshold between steady and pulsating flow

5.2.1 General

If the amplitude of the periodic flowrate variations is sufficiently small there should not be any error in the indicated flowrate greater than the normal measurement uncertainty. It is possible to define amplitude thresholds for both differential pressure (DP) type flowmeters and turbine flowmeters without reference to pulsation frequency. It is also possible to do this for vortex flowmeters but extreme caution is necessary if even the smallest amplitude is known to be present in the flow.

For DP-type flowmeters, the threshold is relevant when slow-response DP cells are being used. In the case of turbine flowmeters, the threshold value is relevant when there is any doubt about the ability of the rotor to respond to the periodic velocity fluctuations. In the case of a vortex flowmeter the pulsation frequency relative to the vortex-shedding frequency is a much more important parameter than the velocity pulsation amplitude.

5.2.2 Differential pressure (DP) type flowmeters

The threshold can be defined in terms of the velocity pulsation amplitude such that the flow can be treated as steady if

$$\frac{U'_{\mathrm{RMS}}}{\bar{U}} \leq 0{,}05 \tag{1}$$

where U is the instantaneous bulk-mean axial velocity such that

$$U = U' + \bar{U} \tag{2}$$

where

U' is the periodic velocity fluctuation;

$\bar{U}$ is the time-mean value.

The threshold in terms of the equivalent DP pulsation amplitude is

$$\frac{\Delta p'_{\mathrm{p,RMS}}}{\overline{\Delta p_{\mathrm{p}}}} \leq 0{,}10 \tag{3}$$

where Δp_{p} is the instantaneous differential pressure across the tappings of the primary device such that

$$\Delta p_{\mathrm{p}} = \overline{\Delta p_{\mathrm{p}}} + \Delta p'_{\mathrm{p}} \tag{4}$$

where

$\overline{\Delta p_{\mathrm{p}}}$ is the time-mean value;

$\Delta p'_{\mathrm{p}}$ is the periodic differential pressure fluctuation.

To determine the velocity pulsation amplitude it is necessary to use one of the techniques described in 5.5 such as laser Doppler or thermal anemometry. To determine the DP pulsation amplitude it is necessary to use a fast-response DP sensor and to observe the rules governing the design of the complete secondary instrumentation system as described in 6.1.3.

Theoretical considerations are covered in Annex A.

5.2.3 Turbine flowmeters

At a given velocity pulsation amplitude a turbine flowmeter tends to read high as the frequency of pulsation increases and exceeds the frequency at which the turbine rotor can respond faithfully to the velocity fluctuations. The positive systematic error reaches a plateau value depending on the amplitude and thus the threshold amplitude can be defined such that the resulting maximum systematic error is still within the general measurement uncertainty. For example, if the overall measurement uncertainty is greater than or equal to 0,5 % then it can be assumed that a systematic error due to pulsation of 0,1 % or less has negligible effect on the overall measurement uncertainty.

The velocity amplitude of sinusoidal pulsation, $U'_{\mathrm{RMS}}/\bar{U}$, that produces a systematic error of 0,1 % in a turbine flowmeter is 3,5 %. Thus the threshold for sinusoidal pulsation is given by

$$\frac{U'_{\mathrm{RMS}}}{\bar{U}} \leq 0{,}035 \tag{5}$$

Techniques such as laser Doppler and thermal anemometry can be used to determine the velocity pulsation amplitude. If the flowmeter output is a pulse train at the blade passing frequency and if the rotor inertia is known, then signal analysis can be used to determine the flow pulsation amplitude as described in 6.2.

5.2.4 Vortex flowmeters

A vortex flowmeter is subject to very large pulsation errors when the vortex-shedding process locks in to the flow pulsation. There is a danger of this happening when the pulsation frequency is near the vortex-shedding frequency. At a sufficiently low amplitude, locking-in does not occur and flow-metering errors due to pulsation are negligible. This threshold amplitude, however, is only about 3 % of the mean velocity and is comparable to the velocity turbulence amplitude. The consequences of not detecting the pulsation or erroneously assuming the amplitude is below the threshold can be very serious. This issue is discussed further in 6.3.

5.3 Causes of pulsation

Pulsation occurs commonly in industrial pipe flows. It might be generated by rotary or reciprocating positive displacement engines, compressors, blowers and pumps. Rotodynamic machines might also induce small pulsation at blade passing frequencies. Pulsation can also be produced by positive-displacement flowmeters. Vibration, particularly at resonance, of pipe runs and flow control equipment is also a potential source of flow pulsation, as are periodic actions of flow controllers, e.g. valve “hunting” and governor oscillations. Pulsation might also be generated by flow separation within pipe fittings, valves, or rotary machines (e.g. compressor surge).

Flow pulsation can also be due to hydrodynamic oscillations generated by geometrical features of the flow system and multiphase flows (e.g. slugging). Vortex shedding from bluff bodies such as thermometer wells, or trash grids, or vortex-shedding flowmeters fall into this category. Self-excited flow oscillations at tee-branch connections are another example.

5.4 Occurrence of pulsating flow conditions in industrial and laboratory flowmeter installations

In industrial flows, there is often no obvious indication of the presence of pulsation, and the associated errors, because of the slow-response times and heavy damping of the pressure and flow instrumentation commonly used. Whenever factors such as those indicated in 5.3 are present, there is the possibility of flow pulsation occurring. It should also be appreciated that pulsation can travel upstream as well as downstream and thus possible pulsation sources could be on either side of the flowmeter installation. However, amplitudes might be small and, depending on the distance from pulsation source to flowmeter, might be attenuated by compressibility effects (in both liquids and gases) to undetectable levels at the flowmeter location. Pulsation frequencies range from fractions of a hertz to a few hundred hertz; pulsation amplitudes relative to mean flow vary from a few percent to 100 % or larger. At low percentage amplitudes the question arises of discrimination between pulsation and turbulence.

Flow pulsation can be expected to occur in various situations in petrochemical and process industries, natural gas distribution flows at end-user locations and internal combustion engine flow systems. Flow-metering calibration systems might also experience pulsation arising from, for example, rotodynamic pump blade passing effects and the effects of rotary positive-displacement flowmeters.

5.5 Detection of pulsation and determination of frequency, amplitude and waveform

5.5.1 General

If the presence of pulsation is suspected then there are various techniques available to determine the flow pulsation characteristics.

5.5.2 Characteristics of the ideal pulsation sensor

The ideal sensor would be non-intrusive, would measure mass flowrate, or bulk flow velocity, and would have a bandwidth from decihertz to several kilohertz. The sensor would respond to both liquids and gases and not require any supplementary flow seeding. The technique would not require optical transparency or constant fluid temperature. The sensor would be uninfluenced by pipe wall material, transparency or thickness. The device would have no moving parts, its response would be linear, its calibration reliable and unaffected by changes in ambient temperature.

5.5.3 Non-intrusive techniques

5.5.3.1 Optical: laser Doppler anemometry (LDA)

This technology is readily available, but expensive. Measurement of point velocity on the tube axis allows an estimate only of bulk flow pulsation amplitude and waveform but, for constant frequency pulsation, accurate frequency measurements can be made. Optical access to an optically transparent fluid is either by provision of a transparent tube section, or insertion of a probe with fibre-optic coupling. With the exception of detecting low frequency pulsation, supplementary seeding of the flow would probably be required to produce an adequate bandwidth. LDA characteristics are comprehensively described in Reference [2].

5.5.3.2 Acoustic: Doppler shift; transit time

Non-intrusive acoustic techniques are suitable for liquid flows only, because for gas flows there is poor acoustic-impedance match between the pipe wall and flowing gases. For the externally mounted transmitter and receiver, usually close-coupled to the tube wall, an acoustically transparent signal path is essential. The Doppler shift technique might require flow seeding to provide adequate scattering. Instruments for point velocity measurements are available which, as for the LDA, provide only an estimate of bulk flow pulsation amplitude and waveform. Moreover, Doppler-derived “instantaneous” full-velocity profile instruments[3] allow much closer estimates of bulk flow pulsation characteristics. Transit-time instruments measure an average velocity, most commonly along a diagonal path across the flow. All acoustic techniques are limited in bandwidth by the requirement that reflections from one pulse of ultrasound should decay before transmission of the next pulse. Many commercial instruments do not provide the signal processing required to resolve unsteady flow components. An investigation by Hakansson[4] on a transit time, intrusive-type ultrasonic flowmeter for gases subjected to pulsating flows showed that only small shifts in the calibration took place and that these were attributable to the changing velocity profile.

5.5.3.3 Electromagnetic flowmeters

When the existing flowmeter installation is an electromagnetic device, then, if it is of the pulsed d.c. field type (likely maximum d.c. pulse frequency a few hundred hertz), there is the capability to resolve flow pulsation up to frequencies approximately five times below the excitation frequency. This technique is only suitable for liquids with an adequate electrical conductivity. It provides a measure of bulk flow pulsation, although there is some dependence upon velocity profile shape[5].

5.5.4 Insertion devices

5.5.4.1 Thermal anemometry

The probes used measure point velocity, and relatively rugged (e.g. fibre-film) sensors are available for industrial flows. These probes generally have an adequate bandwidth, but the amplitude response is inherently non-linear. As with other point velocity techniques, pulsation amplitude and waveform can only be estimated. Estimates of pulsation velocity amplitude relative to mean velocity may be made without calibration. The RMS value of the fluctuating velocity component can be determined by using a true RMS flowmeter to measure the fluctuating component of the linearized anemometer output voltage. Mean-sensing RMS flowmeters should not be used as these only read correctly for sinusoidal waveforms. Accurate frequency measurements from spectral analysis can be made for constant frequency pulsation.

Applications are limited to clean, relatively cool, non-flammable and non-hostile fluids. Cleanness of flow is very important; even nominally clean flows can result in rapid fouling of probes with a consequent dramatic loss of response. A constant temperature flow is desirable although a slowly varying fluid temperature can be accommodated.

5.5.4.2 Other techniques

Insertion versions of both acoustic and electromagnetic flowmeters are available. Transit-time acoustic measurements can be made in gas flows when the transmitter and receiver are directly coupled to the flow[6], although this might require a permanent insertion. Again there is the limitation of a lack of commercially available instrumentation with the necessary signal processing to resolve time-varying velocity components.

Insertion electromagnetic flowmeters are not widely available and are subject to the same bandwidth limitations as the tube version, due to the maximum sampling frequency of the signal.

5.5.5 Signal analysis on existing flowmeter outputs: software tools

5.5.5.1 Orifice plate with fast-response DP sensor

A fast-response secondary measurement system is capable of correctly following the time-varying pressure difference produced by the primary instrument provided the rules given in 6.1.3.2 can be followed. In principle, a numerical solution of the pressure difference/flow relationship derived from the quasi-steady temporal inertia model, Formula (A.11), would then provide an approximation to the instantaneous flow. The square-root error would not be present, although other measurement uncertainties (e.g. C_D variations, compressibility effects) produced by the pulsation would be. Successive numerical solutions would then provide an approximation to the flow as a function of time and, hence, amplitude and waveform information. Frequency information can be determined directly from the measured pressure difference. At the time of publication of this document, there is no software tool described for this implementation.

However, the maximum probable value of $q'_{V\text{o,RMS}}$ can be approximately inferred from a measurement of $\Delta p'_{\text{po,RMS}}$ using one of the following two inequalities:

$$\frac{q'_{V\text{o,RMS}}}{\overline{q_V}} \leq \frac{1}{2}\frac{\Delta p'_{\text{po,RMS}}}{\Delta p_{\text{ss}}} \qquad (6)$$

$$\frac{q'_{V\text{o,RMS}}}{\overline{q_V}} \leq \left\{ \frac{2}{1+[1-(\Delta p'_{\text{po,RMS}} / \overline{\Delta p_{\text{po}}})]^{1/2}} - 1 \right\}^{1/2} \tag{7}$$

where

$\Delta p'_{\text{po,RMS}}$ is the r.m.s. value of the fluctuating component of the differential pressure across the primary element measured using a fast-response secondary measurement system;

Δp_{ss} is the differential pressure that would be measured across the primary element under steady flow conditions with the same time-mean flowrate;

$\overline{\Delta p_{\text{po}}}$ is the time-mean differential pressure that would be measured across the primary element under undamped pulsating flow conditions;

Δp_{po} is the instantaneous differential pressure across the primary element under undamped pulsating flow conditions where

$$\Delta p_{\text{po}} = \overline{\Delta p_{\text{po}}} + \Delta p'_{\text{po}} \tag{8}$$

NOTE 1 Reliable measurements of $\overline{\Delta p_{\text{po}}}$ and $\Delta p'_{\text{po,RMS}}$ can only be obtained if the recommendations given in 6.1.2 and 6.1.3 are strictly adhered to.

NOTE 2 If it is possible to determine Δp_{ss} Formula (6) is to be preferred. Formula (7) only gives reliable results if $\left(\Delta p'_{\text{po,RMS}} / \overline{\Delta p_{\text{po}}}\right) < 0{,}5$.

5.5.5.2 Turbine flowmeter

The raw signal from a turbine flowmeter is in the form of an approximately sinusoidal voltage with a level which varies with the flow but is usually in the range 10 mV to 1 V peak to peak. In most installations this signal is amplified and converted to a stream of pulses. The extraction of information about the amplitude and waveform of any flow pulsation from the variations in the frequency of this pulse train depends on the value of the dynamic response parameter of the flowmeter. Flowmeter manufacturers do not normally specify the response parameter for their flowmeters, and the measurements which would be necessary to determine it are unlikely to be possible on an existing flowmeter installation. However, the dependency of the parameter on the geometry of the turbine rotor and on the fluid density is discussed in 6.2.1.4, and the range of values which have been found for typical flowmeters is presented in 6.2.1.6, Table 1.

The response of a turbine flowmeter to flow pulsation is discussed in 6.2.1. It can range from the ability to follow the pulsation almost perfectly (medium to large flowmeters in liquid flows) to an almost total inability to follow the pulsation (small to medium flowmeters in gas flows with moderate to high frequencies of pulsation). This latter condition is a worst case for a turbine flowmeter installation because not only does the flowmeter output not show significant pulsation but if the flow pulsation is of significant magnitude, the apparently steady flowmeter output is not a correct representation of the mean flow. If this condition is suspected, other means of measuring the flow pulsation should be employed.

In any particular installation, the first step in an attempt to interpret a turbine flowmeter output should be to take the best available estimate of the flowmeter response parameter and using the results summarized in 6.2.1, to estimate the general nature of the flowmeter response. In the interpretation of any observable fluctuations in the turbine flowmeter output, unevenness in the spacing of the turbine blades can give the appearance of flow pulsation at the rotor frequency. Unevenness in blade spacing might be a result either of damage caused by the passing of a solid through the flowmeter or of manufacturing tolerances. Unevenness of as much as 3 % or 4 % in the blade spacing has been observed in a number of installations. A procedure for processing a turbine flowmeter output signal to remove the effect of uneven blade spacing is given in Annex C.

If preliminary estimates of the frequency of any pulsation in the flowmeter output and the general nature of the flowmeter response combine to suggest that the amplitude of pulsation in the flowmeter output is being attenuated by limited flowmeter response, it might be possible to correct the output. Two possible methods of correction have been described by Cheesewright et al.[7] and by Atkinson[8]; both are summarized in Annex C. Within the constraints of the uncertainty about the value of the flowmeter response parameter, this procedure can yield estimates of the amplitude and waveform of the flow pulsation.

5.5.5.3 Vortex flowmeter

The vortex flowmeter output can be used for instantaneous flow measurements, and hence amplitude and waveform information, in a range restricted to pulsation frequencies less than 2,5 % of the lowest mean-flow vortex-shedding frequency. Limited information can be obtained at higher pulsation frequencies but, in order to avoid the substantial flowmeter errors which can arise from the shedding frequency becoming locked-in to the pulsation frequency (see 6.3.1.3), the pulsation frequency should be less than 25 % of the mean-flow shedding frequency. The detection of pulsation frequencies substantially above the mean-flow shedding frequency can be achieved by spectral analysis. The pulsation frequency is indicated by a local peak in the power spectrum.

6 Measurement of the mean flowrate of a pulsating flow

6.1 Orifice plate, nozzle, and Venturi tube

6.1.1 Description of pulsation effects and parameters

6.1.1.1 Square-root error

For steady flow, the flowrate through a restriction such as an orifice plate is proportional to the square-root of the differential pressure measured between upstream and downstream tappings. The relationship is given by

$$q_m = C_\mathrm{D} \pi \frac{d^2}{4} \varepsilon_\mathrm{ss} \sqrt{\frac{2\rho \Delta p_\mathrm{ss}}{1-\beta^4}} \qquad (9)$$

If this relationship was assumed to apply instantaneously during pulsating flow, i.e. assuming quasi-steady conditions, the time-mean flowrate would be inferred from a measurement of the time-mean value of $\overline{\Delta p^{1/2}}$.

Any attempt to infer the time-mean flowrate from the square-root of the time-mean value of Δp would result in a square-root error, because

$$\left(\overline{\Delta p}\right)^{1/2} \neq \overline{\Delta p^{1/2}} \qquad (10)$$

In fact, the quasi-steady assumption is only valid for very low pulsation frequencies in incompressible flow. For a more complete understanding of pulsating flow behaviour of DP flowmeters it is necessary to consider temporal inertia effects, compressibility effects, and factors affecting the discharge coefficient. A brief account of these is given in 6.1.1.2 and 6.1.1.3, and further details can be found in A.4 and in Gajan et al.[9].

6.1.1.2 Temporal inertia

When the flowrate is varying rapidly there is a component of differential pressure required to generate the temporal acceleration in addition to that required for the convective acceleration of the fluid through the restriction. The flowrate-differential pressure relationship is thus

$$\Delta p_{\mathrm{p}} = K_1 \frac{\mathrm{d}q_m}{\mathrm{d}t} + K_2 q_m^2 \tag{11}$$

On the right-hand side of Formula (11), the first term is the temporal inertia term and the second term is the convective inertia term. The temporal inertia term is a function of the non-dimensional frequency known as the Strouhal number, Sr_d, with respect to the throat bore, d, of the orifice, nozzle or Venturi tube, where

$$Sr_d = \frac{f_{\mathrm{p}} d}{\overline{U}_d} \tag{12}$$

In the basic quasi-steady/temporal inertia theory the coefficients K_1 and K_2 are assumed to be constant and are defined as

$$K_1 = \frac{4L_{\mathrm{e}}}{\pi d^2 C_{\mathrm{c}}} \tag{13}$$

$$K_2 = \frac{1 - C_{\mathrm{c}}^2 \beta^4}{C_v^2 C_{\mathrm{c}}^2 (\pi d^2 / 4)^2} \frac{1}{2\rho} \tag{14}$$

or alternatively

$$K_2 = \frac{1 - \beta^4}{C_{\mathrm{D}}^2 (\pi d^2 / 4)^2} \frac{1}{2\rho} \tag{15}$$

where

C_{D} is the overall discharge coefficient;

C_{c} is the contraction coefficient;

C_v is a velocity coefficient.

The temporal inertia term is also a function of the geometry of the restriction and the axial distance between the pressure tappings, and thus the coefficient K_1 contains L_{e}, an effective axial length of the primary device.

In pulsating flow the velocity profiles upstream and through the restriction are varying cyclically and, thus, K_1 and K_2 are varying cyclically and even their time-mean values are not necessarily equal to the steady flow values, except when pulsation amplitudes and frequencies are small.

6.1.1.3 Discharge coefficients

In steady flow, the discharge coefficients of all the different types of primary device are dependent on the velocity profile of the approaching flow. The orifice plate tends to be particularly sensitive to variations in velocity profile because of the jet contraction effect. A flatter than normal velocity profile increases the contraction effect and consequently reduces the discharge coefficient. A velocity profile which is more peaked than normal has the opposite effect.

In pulsating flow, the instantaneous velocity profile is varying throughout the pulsation cycle. The degree of variation is dependent on the velocity pulsation amplitude, the waveform and the pulsation Strouhal number. As a consequence, the instantaneous discharge coefficient also depends on the phase angle in the pulsation cycle, the pulsation amplitude, the waveform and the Strouhal number. At the time of publication of this Document it is not possible to relate mathematically the instantaneous discharge coefficient to the pulsation parameters.

The practical approach to the calculation of a flowrate in pulsating flow conditions is to use a constant value of the discharge coefficient, preferably the value used in steady flow conditions. This approach gives accurate results for low amplitude and low frequency pulsation in incompressible flow and limiting values of the relevant parameters defined in 6.1.2.3. Residual errors due to temporal inertia effects and variations in discharge coefficient increase with pulsation amplitude and frequency as shown by Gajan et al.[9].

6.1.2 Flowmeters using slow-response DP sensors

6.1.2.1 Limiting conditions of applicability

For a slow-response DP sensor (upper frequency limit of about 1 Hz) then, at best, the time-mean differential pressure $\overline{\Delta p}_{\mathrm{p}}$ is indicated. The corresponding indicated mean flowrates derived from $\left(\overline{\Delta p}_{\mathrm{p}}\right)^{1/2}$ include both the square-root and temporal-inertial effect errors. Conditions limiting applicability are those which prevent the secondary measurement system producing a correct time-mean pressure signal. These include distortion of the pressure waveform or phase relationship in either of the two lines connecting tappings to the sensor. These effects arise from boundary friction, finite gas volumes and non-linear damping. In addition, the connecting line length should be restricted to prevent resonance due to this length being equal to the pulsation quarter-wavelength. This resonance occurs at a frequency, f_r, given by $f_r = c/(4l)$. In practice minimum connecting line length is restricted by physical size of primary and secondary instrumentation and associated valve assemblies. At the time of publication of this Document, it is not possible to define a threshold level of negligible pulsation applicable to all designs of secondary device. However, it is possible to recommend a number of design rules.

6.1.2.2 Design of flowmeter secondary instrumentation

For devices used to indicate the time-mean differential pressures in pulsating flow conditions, the design rules are as follows.

a) The bore of the pressure tapping should be uniform and not too small, i.e. ≥3 mm. Piezometer rings should not be used.

b) Distance between pressure tappings should be small compared with the pulsation wavelength.

c) The tube connecting the pressure tappings to the manometer should be as short as possible and of the same bore as the tappings. A tube length near the pulsation quarter-wavelength should not be used.

d) For gas-filled secondary systems, sensor cavities or other discrete volumes should be as small as possible.

e) For liquid-filled secondary systems, gas bubbles should not be trapped in the connecting tube or sensing device; thus vent points are required.

f) Damping resistances in the connecting tubes and sensing element should be linear. Throttle cocks should not be used.

g) The device time constant should be about 10 times the period of the pulsation cycle.

h) When the above rules cannot be observed, the secondary device might be effectively isolated from pulsation by the insertion of identical linear-resistance damping plugs into both connecting tubes, as close as possible to the pressure tappings.

Observance of the rules listed in items a) to h) for a slow-response device cannot eliminate the square-root error but merely reduces the error in the measurement of the time-mean differential pressure.

6.1.2.3 Estimation of correction factors and measurement uncertainties due to pulsation

The formulae given in 6.1.4 allow adequate damping to be calculated for a given maximum allowable relative error, ψ, in the indicated flowrate due to the residual damped pulsation. It is also possible to estimate the total error, E_T, directly, after measuring the pulsation amplitude at the orifice. The total error, E_T, should be less than ψ.

In theory, E_T is always a positive systematic error, but in practice there is an additional random uncertainty mostly due to pulsation effects in the secondary device. Calculations of the errors and additional uncertainty are feasible provided that the pulsation amplitudes are not too large.

Limiting pulsation amplitudes for error calculations are

$$\frac{q'_{V,\mathrm{RMS}}}{\overline{q_V}} = \frac{U'_{\mathrm{RMS}}}{\overline{U}} \leq 0{,}32 \tag{16}$$

or

$$\frac{\Delta p'_{\mathrm{p,RMS}}}{\Delta p_{\mathrm{ss}}} \leq 0{,}64 \tag{17}$$

or

$$\frac{\Delta p'_{\mathrm{p,RMS}}}{\overline{\Delta p_{\mathrm{p}}}} \leq 0{,}58 \tag{18}$$

The following equations can be used to estimate the total error, E_T, for the low amplitude pulsation:

$$E_{\mathrm{T}} = \left[1 + \left(\frac{U'_{\mathrm{RMS}}}{\overline{U}}\right)^2\right]^{1/2} - 1 \tag{19}$$

or

$$E_{\mathrm{T}} = \left[1 + \frac{1}{4}\left(\frac{\Delta p'_{\mathrm{p,RMS}}}{\Delta p_{\mathrm{ss}}}\right)^2\right]^{1/2} - 1 \tag{20}$$

or

$$E_{\mathrm{T}} = \left(\frac{1}{2}\left\{1 + \left[1 - \left(\frac{\Delta p'_{\mathrm{p,RMS}}}{\overline{\Delta p_{\mathrm{p}}}}\right)^2\right]^{1/2}\right\}\right)^{-1/2} - 1 \tag{21}$$

The practicality of using Formula (19) is limited by the requirement of an independent measurement of pulsation bulk velocity. Similarly, use of Formula (20) is restricted in practice because of the requirement to know the steady flow differential pressure. Neither Formula (19) nor Formula (20) contains terms for temporal inertia effects and they thus tend to give slightly overestimated values of E_T when the Strouhal number for the pulsation is high ($Sr_d > 0{,}02$).

The systematic error can be compensated for by multiplying the discharge coefficient by $(1 - E_T)$.

There is an additional uncertainty in the value of the discharge coefficient due to the pulsation, even after allowing for the systematic effect. There is also some additional uncertainty due to temporal inertia effects at high Strouhal numbers.

Thus the percentage additional uncertainty for pulsating flow is equal to 100 E_T and should be added to the uncertainty calculated for steady flow. If the Strouhal number for the pulsation is less than 0,02, the percentage additional uncertainty may be reduced to 50 E_T.

6.1.3 Flowmeters using fast-response DP sensors

6.1.3.1 Limiting conditions of applicability

If the time-varying difference can be faithfully followed by the secondary measurement system and the square-root of its signal averaged, then the simple square-root error is eliminated for the indicated time-mean flowrate. The conditions limiting applicability are those which prevent the secondary measurement system from producing a correct instantaneous differential pressure signal. In addition to the factors given in 6.1.2.1, there is the need to avoid dynamic distortion due to the secondary measurement system resonant frequency being too close to the pulsation frequency. For gas-filled systems, the limiting resonance is commonly that due to the connecting line length being close to the pulsation quarter-wavelength, rather than the resonant frequency of the transmitter or transducer alone. For liquid-filled systems, the limiting resonance might be either that due to the coupled stiffness of the sensor and inertial properties of the liquid-filled connecting lines, or it might be the quarter-wavelength resonant frequency, depending on the system geometry. The presence of gas bubbles in a liquid-filled system can have a dramatic effect upon the resonant frequency[10].

6.1.3.2 Design of flowmeter secondary instrumentation

For design of a fast-response secondary measurement system, a DP transmitter or transducer with very small internal volume and high natural frequency is required. Rule a) of 6.1.2.2 applies for gas-filled systems, and rules, b), d), and e) of 6.1.2.2 and the following apply.

a) The mechanical and electronic frequency limits of the secondary measurement system should be at least ten times greater than the pulsation frequency.

b) Connecting tube lengths should be as short as possible and the pressure path length from tapping to sensor should be less than 10 % of the pulsation quarter-wavelength.

c) For liquid-filled secondary systems the bore of the connecting tubes should be greater than or equal to 5 mm to avoid inertial effects reducing the resonant frequency.

d) Connecting tubes, fittings and valves should be of the same bore.

e) The secondary device should be geometrically identical on both upstream and downstream sides.

f) Secondary devices for liquids should not have any gas bubbles trapped in the connecting lines or the device itself; thus vent points are necessary.

6.1.3.3 Estimation of correction factors and measurement uncertainties due to pulsation

Assuming that the fast-response DP sensor is used in conjunction with a signal processor which generates an output proportional to the square-root of the instantaneous differential pressure, the systematic error due to the “square-root effect” is eliminated. There are some additional uncertainties due to pulsation, however, which can be estimated as follows.

Provided that the frequency response of the entire secondary system, including the connecting tubes, fittings and DP sensor, can be proved by experiment to be flat from 0 to 10 f_p (where f_p is the fundamental pulsation frequency), the additional percentage uncertainty may be taken as

25 E_T if $Sr_d < 0,02$

or

50 E_T if $Sr_d \geq 0,02$

As indicated in 6.1.2, E_T can be estimated using Formula (19), (20) or (21) provided that the flow can be regarded as incompressible. This is the case if $\varepsilon_{SS} \geq 0{,}99$.

Systematic errors in the secondary measurement system arise from pressure wave distortion effects in connecting lines and from resonance effects due either to connecting line quarter-wavelength resonance or connecting line plus sensor cavity resonance. Because these errors are strongly dependent upon secondary system geometry and the fluid, it is not possible to give general results for error estimation. Details of these effects and suggested correction procedures are given by Clark[10] and Botros et al.[11].

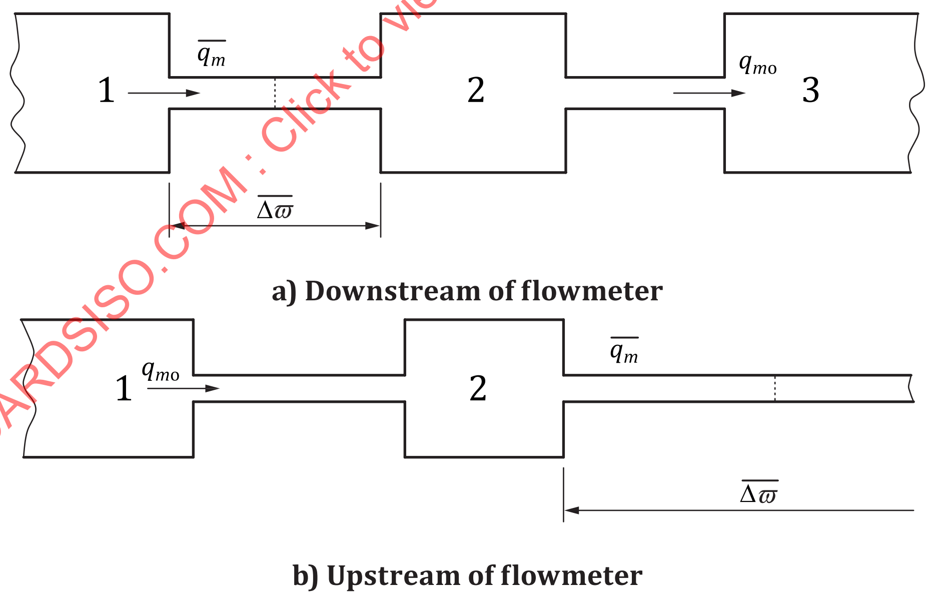
6.1.4 Pulsation damping

6.1.4.1 Adequate damping criteria for gas flow

6.1.4.1.1 General

Pulsation in gases or vapours can be damped by a combination of volumetric capacity and throttling placed between the pulsation source and the flowmeter (see Figure 1). The volumetric capacity can include the volume of any receivers and the pipeline itself, provided that the axial lengths involved are short compared with the pulsation wavelength. The throttling can be provided by the flowmeter itself and can be augmented by valves and other fittings. The frictional pressure losses in the pipeline can also contribute towards the throttling effect. The straight pipe length between the flowmeter and any additional throttling device should be in accordance with the installation requirements of the appropriate part of ISO 5167. Care should be taken that the selected throttling device does not create hydrodynamic oscillations. Further work on pulsation damping criteria is given in Annex B.

In this Document more attention is paid to single-receiver damping systems than to divided-receiver systems because it is possible to present a simple mathematical representation of an adequate damping criterion for the former but not for the latter.



Key

1 source of supply at constant pressure

2 receiver

3 source of pulsations (for example: reciprocating engine)

Figure 1 — Flowmeter installations including damping receivers for pulsating gas flows

6.1.4.1.2 Single-receiver damping system

Figure 1 shows the installation arrangements of the flowmeter and the damping system for both upstream and downstream pulsation sources.

A parameter which can be used as a criterion of adequate damping in gas flow is the Hodgson number, *Ho*, defined by

$$Ho = \frac{V}{q_V / f_\mathrm{p}} \frac{\Delta \varpi}{p} \tag{22}$$

where

V	is the volume of the receiver and the pipework between pulsation source and flowmeter;
q_V/f_p	is the time-mean volume flow per pulsation cycle;
$\Delta\varpi$	is the overall time-mean pressure loss between the receiver and the source of supply (or discharge) at constant pressure;
p	is the mean absolute static pressure in the receiver.

Damping is adequate provided that the following condition is fulfilled:

$$\frac{Ho}{\kappa} \geq \frac{1}{4\pi\sqrt{2}} \frac{1}{\sqrt{\varphi}} \frac{q'_{m\mathrm{o,RMS}}}{\overline{q_m}} \tag{23}$$

where

κ	is the isentropic exponent of the gas or vapour (for an ideal gas, $\kappa = \gamma$, the ratio of the specific heat capacities);
$q'_{m\mathrm{o,RMS}}$	is the root-mean-square value of the fluctuating component of the mass flowrate measured at the pulsation source;
$\overline{q_m}$	is the time-mean value of the mass flowrate;
φ	is the maximum allowable uncertainty in the indicated flowrate due to pulsation at the flowmeter.

The pulsation amplitude ratio can be expressed in terms of mass or volume flowrate or bulk-mean velocity.

Thus,

$$\frac{q'_{m\mathrm{o,RMS}}}{\overline{q_m}} = \frac{q'_{V\mathrm{o,RMS}}}{\overline{q_V}} = \frac{U'_{\mathrm{o,RMS}}}{\overline{U}} \tag{24}$$

The derivation of the criteria for adequate damping is only valid when the dimensions of the damping chamber and the lengths of pipe between the damping chamber and the flowmeter are short compared with the pulsation wavelength.

The following guidelines may be used:

a) the length of the damping receiver, L_1, should not be greater than 1/10 of the pulsation wavelength; thus, the frequency limit is given by

$$f_p < \frac{c}{10L_1} \tag{25}$$

where c is the speed of sound;

b) the length of the pipe, L_2, between the damping receiver and the flowmeter should not be greater than 1/5 of the pulsation wavelength; thus, the limiting frequency should also fulfil the condition

$$f_p < \frac{c}{5L_2} \tag{26}$$

The criterion of adequate damping depends on having an accurate value of the undamped pulsation amplitude, $U'_{RMS}/\overline{U}$. The undamped pulsation amplitude should be determined very close to the damping receiver on the side nearest the pulsation source. If the amplitude has to be estimated it is recommended that a safety factor of at least 2 be applied to the calculated Hodgson number.

6.1.4.1.3 Divided-receiver damper with choke tube

This kind of damper is shown in Figure 2 and can be analysed by means of an electrical/acoustic analogy as described by Chilton and Handley[12], Wallace[13] and Mottram[14]. The damping system is considered to have lumped impedances and the damping response factor, μ, can be shown to be given by

$$\mu = \frac{(U'_{RMS}/\overline{U})_{damped}}{(U'_{RMS}/\overline{U})_{undamped}} = \frac{1}{\left\{\left[1-3\left(\frac{\omega}{\omega_o}\right)^2+\left(\frac{\omega}{\omega_o}\right)^4\right]^2+\left(\frac{2\pi Ho}{\kappa}\right)^2\left[2-\left(\frac{\omega}{\omega_o}\right)^2\right]^2\right\}^{1/2}} \tag{27}$$

where

$\omega = 2\pi f_p$ is the angular pulsation frequency;

ω_o is the angular resonant frequency for one half of the divided receiver

where

$$\omega_o = \frac{1}{(LC)^{1/2}}$$

where

L is the inductance and equal to $\rho l_c/A_c$;

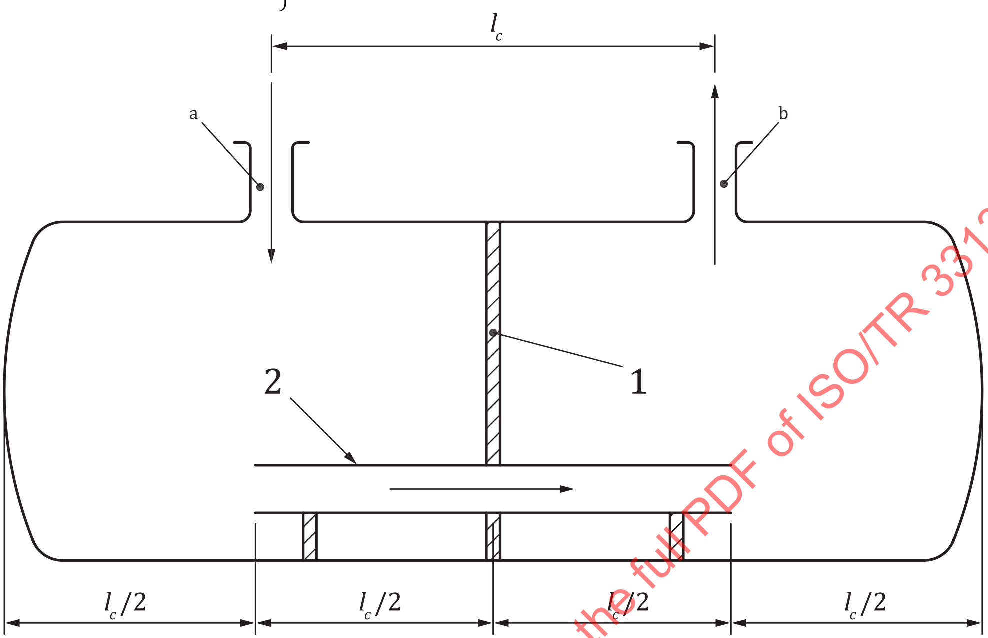
l_c is the length of the choke tube;

A_c is its cross-sectional area;

C is the capacitance and is represented by $V/2$, half the total volume of the receiver.

The equivalent damping response equation for a single-receiver damper as described in 6.1.4.1.2 is

$$\mu = \frac{1}{\left\{ \left[1 - \left(\frac{\omega}{\omega_o} \right)^2 \right]^2 + \left(\frac{4\pi H o}{\kappa} \right)^2 \right\}^{1/2}} \quad (28)$$



Key

1 partition (baffle)

2 restrictor (choke tube)

a In.

b Out.

Figure 2 — Divided damping receiver with internal choke tube

The validity of this analysis and the resulting Formulae (27) and (28) are limited to conditions where the dimensions of the receivers and the lengths of the pipework between the damper and the pulsation source and flowmeter are all short compared with the pulsation wavelength.

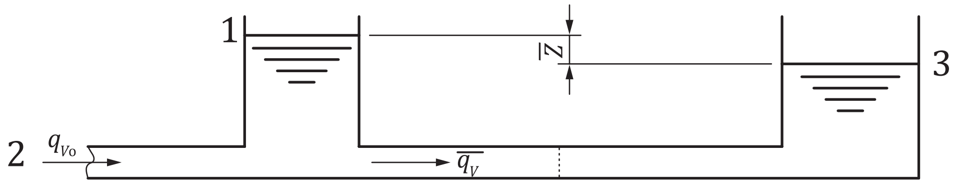
When this condition is not fulfilled a more rigorous analysis, involving the equations to describe the propagation of acoustic waves through the damping system, is required. Such analysis, applied to a variety of damping system configurations, is described by Blodgett[15].

6.1.4.2 Adequate damping criteria for liquid flow

6.1.4.2.1 General

Pulsating liquid flow can be damped by placing either a surge chamber or an air vessel between the pulsation source and the flowmeter (see Figures 3 and 4).

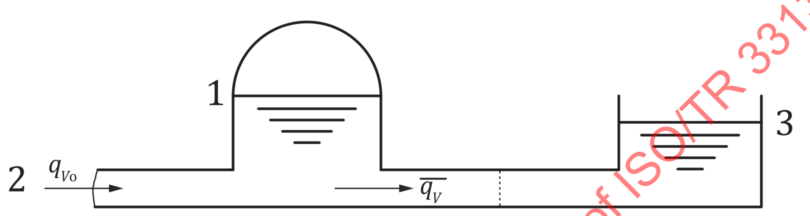
In Figures 3 and 4, the flow is shown with the pulsation source upstream. It is equally possible, however, to have a flow with a constant head source upstream and pulsation source downstream. In this case, the surge chamber or air vessel should be placed downstream of the flowmeter.



Key

1 surge chamber
2 pulsating flow
3 constant head

Figure 3 — Use of a surge chamber in pulsating liquid flows



Key

1 air vessel
2 pulsating flow
3 constant head

Figure 4 — Use of an air vessel in pulsating liquid flows

6.1.4.2.2 Use of a surge chamber

The criterion for adequate damping is

$$\frac{\bar{Z}A}{\bar{q}_V / f_\mathrm{p}} \geq \frac{1}{4\pi\sqrt{2}} \frac{1}{\sqrt{\varphi}} \frac{q'_{Vo,\mathrm{RMS}}}{\bar{q}_V} \tag{29}$$

where

$\bar{Z}$ is the time-mean value of the difference in liquid level between the surge chamber and the constant head vessel;

A is the cross-sectional area of the surge chamber.

6.1.4.2.3 Use of an air vessel

The criterion for adequate damping is

$$\frac{1}{\kappa} \frac{V_0}{\bar{q}_V / f_\mathrm{p}} \frac{\overline{\Delta\varpi}}{p_\mathrm{o}} \frac{1}{\left(1+\dfrac{V_0 \rho g}{p_\mathrm{o} \kappa A}\right)} \geq \frac{1}{4\pi\sqrt{2}} \frac{1}{\sqrt{\varphi}} \frac{q'_{Vo,\mathrm{RMS}}}{\bar{q}_V} \tag{30}$$

where

V_0 is the volume of air in the air vessel;

κ is the isentropic exponent for air;

ρ is the density of the liquid;

g is the acceleration due to gravity;

A is the free surface area of the liquid in the air vessel;

$\overline{\Delta\varpi}$ is the mean value of the difference in pressure between the air vessel and the constant head vessel.

6.2 Turbine flowmeters

6.2.1 Description of pulsation effects and parameters

6.2.1.1 General

A turbine flowmeter comprises a free-running, axial flow, turbine rotor, which for steady flow has a closely linear relationship between the rotational speed and the volume flowrate. The rotational speed is usually obtained by detecting the passage of blade tips past a pickup mounted on the casing. Some flowmeters use the pickup from only one of the blades per revolution but the additional resolution obtained by using the signal from all the blades is often important when there are pulsation effects. When subjected to a time-dependent flow, the inertia of the rotor (and possibly of the fluid contained within the rotor envelope) can cause the rotor speed to lag behind the steady state condition in an accelerating flow and to exceed it in a decelerating flow. The influence of a decelerating flow is greater than that of an accelerating one so that the mean speed of a flowmeter subjected to pulsation can be greater than that corresponding to the mean flowrate. In extreme cases the error can be more than 25 % of the indicated flow.

6.2.1.2 General relationship between metering error and pulsation parameters

The flowmeter error depends on the amplitude and waveform of the pulsation, the mean flowrate, the density of the fluid and the design characteristics of the turbine rotor, including its moment of inertia. The transient behaviour of the flowmeter can be described by

$$b\frac{\mathrm{d}f}{\mathrm{d}t} = q_V{}^2 - q_V f + b\left(\frac{I_F}{I_R + I_F}\right)\frac{\mathrm{d}q_V}{\mathrm{d}t} \tag{31}$$

where

q_V is the actual instantaneous flowrate;

f is the instantaneous flowrate indicated by the flowmeter;

t is the time;

b is the dynamic response parameter;

I_R, I_F are the moments of inertia of the turbine rotor and of the fluid contained within the rotor envelope respectively.

NOTE The dynamic response parameter, b, is a property of the flowmeter/fluid combination, not of the flowmeter alone. Details of the dependency are discussed in 6.2.1.4.

For a given mean flowrate the quantity $b/\overline{q_V}$ has the characteristic of a reference time interval or time constant. For flowmeters in the range of 25,4 mm (1 in) to 101,6 mm (4 in) in diameter, for measuring gases flowing at near atmospheric pressure, typical time constants are of the order of 1 s. For flowmeters in the range from 19,1 mm (3/4 in) to 50,8 mm (2 in) in diameter, for measuring water, typical time constants are of the order of 1 ms. It is thus apparent that errors due to pulsation are much more likely to be significant with flowmeters measuring gases (at atmospheric pressure) than with flowmeters measuring liquids. For gases at atmospheric pressure, the inertia of the fluid contained within the turbine rotor is normally negligible compared with the inertia of the rotor so that the last term in Formula (31) can be neglected.

With reference to Formula (31), the error at any instant is given by

$$\frac{f-q_V}{q_V}=\frac{b}{{q_V}^2}\left(-\frac{\mathrm{d}f}{\mathrm{d}t}+\frac{I_\mathrm{F}}{I_\mathrm{R}+I_\mathrm{F}}\frac{\mathrm{d}q_V}{\mathrm{d}t}\right) \tag{32}$$

In the limit of vanishingly small error, $\mathrm{d}q_V/\mathrm{d}t \approx \mathrm{d}f/\mathrm{d}t$ so that the error is approximated by

$$\frac{f-q_V}{q_V}=-\frac{b}{{q_V}^2}\frac{\mathrm{d}q_V}{\mathrm{d}t}\left(\frac{I_\mathrm{R}}{I_\mathrm{R}+I_\mathrm{F}}\right) \tag{33}$$

and the condition for negligible error is that the right-hand side of Formula (33) should be negligible.

For the special case of sinusoidal pulsation at a frequency f_p and an amplitude $\alpha\left[=\left(q_{V_\mathrm{max}}-q_{V_\mathrm{min}}\right)/\left(2\overline{q_V}\right)\right]$, the error can be written as

$$\frac{f-q_V}{q_V}=-\frac{2\pi f_\mathrm{p}\alpha b}{\overline{q_V}}\frac{I_\mathrm{R}}{I_\mathrm{R}+I_\mathrm{F}}G(\alpha) \tag{34}$$

where $G(\alpha)$ varies smoothly from a value of 1 at $\alpha = 0$ to a value of 1,6 at $\alpha = 0,5$.

For some applications the above criteria might be unduly conservative because the quantity of interest is the mean flow rather than the instantaneous flow. The assumption $\mathrm{d}q_V/\mathrm{d}t \approx \mathrm{d}f/\mathrm{d}t$, which is made above, removes the asymmetry between accelerating and decelerating flows so that Formulae (33) and (34) cannot be used to estimate the error in the mean flow. There is no simple analytical expression for the error in the mean flow. For sinusoidal pulsation in a (low pressure) gas flow Atkinson[8] has made extensive computations of the flowmeter response and has confirmed his results by experimental measurements. A summary of this work is given for information in Annex C where Figure C.1 shows the relative mean-flow error as a function of α and $B(=b f_\mathrm{p}/\overline{q_V})$.

6.2.1.3 Determination of the dynamic response parameter

There are two main ways in which the value of the response parameter can be obtained; one is by experiment via a step response test and the other is by the use of one of the many analytic formulae which have appeared in the literature. The latter are discussed in 6.2.1.4.

Formula (31) shows that the response of a flowmeter to a step change in the flow can be divided into two parts. Firstly, there will be a change in the flowmeter speed to accommodate any influence of the fluid inertia term. Secondly, there will be first order (exponential) transition to the new condition. For gas flows, the first of these parts is small and occurs very rapidly compared with the second. For liquid flows, the rapidity of change which is required to constitute a step (≤100 µs) is such that there have not been any step response tests made with liquids. It seems likely, however, that if such tests were made, the two parts of the response might overlap.

If a flowmeter is subjected to a step change in a gas flow from $q_{V_0} + \Delta q_V$ to q_{V_0} (where Δq_V might be positive or negative) the response is described by

$$f = q_{V_0} + \Delta q_V e^{-q_{V0} t/b} \quad (35)$$

Thus a graph of $\ln(q_{V_0} - f)$ against t should be a straight line with a gradient equal to $-q_{V_0}/b$. Step response tests on a variety of different flowmeters have been made at the University of Surrey and the above behaviour has been confirmed.

6.2.1.4 Dynamic response parameter by calculation

The most sophisticated of the numerous published formulae for b are those by Grey[16] and by Jepson[17]. Grey's derivation applies specifically to flowmeters with short rectangular blades and yields

$$b = \frac{2I\left[1 + 2\eta / A_R\right]}{N\rho\eta LR} \quad (36)$$

where

I is the moment of inertia of the rotor;

η is the blade "airfoil efficiency"; Grey suggests that η = 0,9;

A_R is the blade aspect ratio;

N is the number of blades;

ρ is the density of the fluid;

L is the length of the blades;

R is the mean radius of the blades.

Jepson's derivation applies to helical blades and leads to

$$b = \frac{2I\left[1 + 2\eta / A_R\right]R\left(r_\text{t} - r_\text{h}\right)}{N\rho\eta \int_{r_\text{h}}^{r_\text{t}} Lr^2 \text{d}r} \quad (37)$$

where r_h and r_t are the blade hub and tip radii respectively, and Jepson suggests that η = 1.

It is notable that, in common with almost all the other published formulae, the above expressions suggest that the response parameter is inversely proportional to the density of the fluid. This factor has been used to obtain values of the response parameter for flowmeters designed for liquid flows. It is assumed that the flows in step response tests would also be similar so that the response parameters obtained from step response tests in air can be scaled by the air and the liquid densities to yield the response parameter for a liquid flow.

6.2.1.5 Moment of inertia

If the inertia of the rotor hub is negligible compared with that of the blades then

$$I = N\rho_\text{b} \int_{r_\text{h}}^{r_\text{t}} t_\text{b} C r^2 \text{d}r \quad (38)$$

where

ρ_b is the density of the blade material;

t_b and C are the thickness and the chord of the blade respectively.

For t_b and C independent of r, this simplifies to

$$I = N\rho_b t_b C\left(r_t^{\,3} - r_h^{\,3}\right)/3 \tag{39}$$

6.2.1.6 Representative values of the response parameter

Table 1 gives values of the response parameter which have been measured for a range of different flowmeters.

Table 1 — Representative values of the response parameter

Flowmeter			Design fluid	b
Diameter in	N	Blade material		(per m³ in air at atmospheric pressure unless otherwise specified)
2	12	plastic	gas	0,004 6
3	16	metal	gas	0,023
4	12	plastic	gas	0,016
4	16	plastic	gas	0,017
4	16	metal	gas	0,035
4	16	metal	gas	0,065 gas density 0,935 kg/m³
6	20	metal	gas	0,156 gas density 0,935 kg/m³
6	20	metal	gas	0,183 gas density 0,935 kg/m³
8	20	metal	gas	0,261 gas density 1,105 kg/m³
2	10	stainless steel	water	0,009
1	6	stainless steel	water	0,001 2
7/8	6	metal	kerosene	0,002
3/4	6	metal	water	0,001

6.2.2 Estimation of pulsation correction factors and measurement uncertainties

The estimation of a correction factor might be required either when the nature of the pulsation in the true flow is known or, more commonly, when checks of the type described in 6.2.1 have indicated that there is likely to be an error and the estimation should be made from the characteristics of the flowmeter signal. There are no “simple” analytical relationships for the correction factor in either case. If the pulsations are known and are sinusoidal then the work of Atkinson[8] can be used as described in 6.2.1.2 and in more detail in Annex C. The accuracy of a correction factor obtained in this way would be limited by the need to scale data from Figure C.1 (Annex C) and by the accuracy with which the flowmeter response parameter was known. If the pulsation is known but is significantly non-sinusoidal in character there are no a priori estimates of the correction factor available. In principle it would be possible to evaluate the q_V terms in Formula (31) and then solve that equation for f (for example, by using a Runge-Kutta integration), from which a correction factor could be estimated. Since such a calculation would need to be repeated for each different pulsation amplitude, even when the waveform remained constant, the waveform would need to be very reproducible to justify the effort involved.

The estimation of correction factors from the flowmeter signal has been considered by Atkinson[8] and by Cheesewright et al.[7] The former assumed that the pulsation was sinusoidal, obtained the amplitude and frequency from the flowmeter signal and then used a lookup table based on the same information as presented in Figure C.1 to give the correction factor. Correction factors obtained by this method were checked experimentally and it was found that, provided the amplitude of the pulsation in the flowmeter output signal was at least 0,5 % and the amplitude of the true pulsation was less than 50 %, then the correction factors gave the true flow to an accuracy of 1 %. Further details of this method and of the computer program involved are given in Annex C.

If the pulsation is significantly non-sinusoidal and/or of large amplitude, the method reported by Cheesewright et al.[7] is the only available method of obtaining correction factors. The method is based on the estimation of the time histories of f and $\mathrm{d}f/\mathrm{d}t$ from the flowmeter signal and then the solution of Formula (31) for the time history of q_V.

For some conditions the problem is complicated by Formula (31) having multiple roots. The procedure has been checked against independent measurements for cases of extreme pulsation and correction factors as significant as 0,8 were found which gave a true flow accurate to 1 %. Further details of the procedure are given in Annex C.

Neither of the above methods has been applied to liquid flows where the fluid inertia is likely to be significant. It should also be noted that none of the above work on turbine flowmeters would be accurately applicable to a situation in which the flow switched on and off (at relevant frequencies).

6.3 Vortex flowmeters

6.3.1 Pulsation effects

6.3.1.1 General

A vortex flowmeter consists of two main components: a bluff body for generating vortices and a sensor for detecting the vortices. Under steady flow conditions vortices are shed alternately from each side of the bluff body at a frequency proportional to the upstream flow velocity over a wide range of the Reynolds number.

The relationship for the flow velocity under steady flow condition is

$$U = \frac{f_V d}{Sr} \tag{40}$$

where

- f_V is the vortex-shedding frequency;
- d is the bluff body diameter;
- Sr is the Strouhal number which is constant over the linear range.

Under pulsating flow conditions the crucial parameter is the ratio of the pulsation frequency to the vortex-shedding frequency. When this ratio is small compared with unity quasi-steady behaviour is observed[18][19] and the vortex-shedding frequency follows the velocity variations with no change in Strouhal number or calibration constant.

When the pulsation frequency becomes comparable with the vortex-shedding frequency there is a strong tendency for the vortex-shedding cycle to “lock-in” to the pulsation cycle at the same ($f_V = f_p$) or half ($f_V = f_p/2$) the pulsation frequency. Under locking-in conditions the flowmeter output ceases to be a reliable indicator of flowrate. Errors in the indicated flowrate can be as high as ±80 %.

The flowrate range over which the vortex-shedding frequency remains locked to a fixed frequency flow pulsation depends on the amplitude of the pulsation, the bluff body shape and the blockage ratio. Work carried out in a wind tunnel by Al Asmi and Castro[20] indicates that the bluff body shapes associated with the largest locking-in flowrate range are those normally selected by flowmeter manufacturers to give the strongest and most regular vortex signals. It was also found that the locking-in flowrate range tended to increase as blockage ratio was increased to the values typically found in commercial vortex flowmeters.

When the pulsation frequency is much higher than the vortex-shedding frequency there is no obvious locking-in but shifts in the Strouhal number, and the resulting departure from the steady flow calibration, can still be of the order of ±10 %.

6.3.1.2 Threshold pulsation amplitudes

The critical flow pulsation amplitude is that which is just sufficient to cause locking-in to occur when the pulsation frequency is around twice the vortex-shedding frequency. If the flow pulsation amplitude is less than this critical or threshold value there is no significant shift in Strouhal number, and therefore no significant deviation from the steady-flow calibration.

Available data both from tests in a low turbulence intensity wind-tunnel[20] and from experiments in a pipe flow rig[21] indicate that the critical or threshold flow pulsation amplitude is in the region of 3 % of the mean-flow velocity. These data indicate that the threshold value is independent of bluff body shape and blockage ratio (bluff body frontal area as a fraction of the pipe cross-sectional area).

As the critical velocity pulsation amplitude is of the same order as the velocity turbulence amplitude sophisticated techniques are necessary to detect the pulsation and measure its amplitude. It is suggested that a fast-response velocity sensor such as a hot wire or hot film probe, inserted in the pipe immediately upstream of the vortex flowmeter location, should be used to determine whether or not there is any pulsation present at a frequency and amplitude which might cause locking-in. It is necessary to apply band-pass filtering to exclude turbulence components outside the vortex-shedding frequency range. If no filtering is employed, there is a danger that any pulsation might be masked by the velocity turbulence noise.

Unless it can be demonstrated by the above or similar technique that the velocity pulsation amplitude $U'_{\mathrm{RMS}}/\overline{U}$ is less than 3 %, it should be assumed that a vortex flowmeter is liable to locking-in if the pulsation frequency is in a range from 25 % of the lowest to at least twice the maximum vortex-shedding frequency.

6.3.1.3 Frequency limit for quasi-steady behaviour

The available experimental data are restricted to conditions where the velocity pulsation amplitude $U'_{\mathrm{RMS}}/\overline{U}$ does not exceed 20 %. These data indicate that quasi-steady behaviour can be assumed provided that the pulsation frequency is less than 25 % of the vortex-shedding frequency at the lowest flow velocity, i.e. quasi-steady behaviour if $U'_{\mathrm{RMS}}/\overline{U} \leq 0{,}2$ and $f_v/f_p < 0{,}25$.

6.3.2 Minimizing pulsation effects

The sensitivity of a vortex flowmeter to pulsation is such that it would only be sensible to select this flowmeter type if there was either no pulsation present or if the pulsation frequency was less than 25 % of the lowest vortex-shedding frequency. The vortex-shedding frequency is inversely proportional to the bluff body diameter. Thus, it might be possible to select a vortex flowmeter with a small bluff body diameter (low blockage ratio) to raise the shedding frequency. Insertion vortex flowmeters have very much smaller bluff bodies than full line-size flowmeters.

6.3.3 Estimation of measurement uncertainties

It is only possible to make an estimate of measurement uncertainty if vortex-shedding frequencies are derived from spectral analysis of the raw signal from the vortex sensor and when:

a) quasi-steady conditions exist, $f_v/f_p < 0{,}25$;

b) pulsation frequencies are much higher than twice the maximum shedding frequency.

In the first case data only exist at velocity pulsation amplitudes less than 20 % and these indicate a measurement uncertainty of about 1 %.

In the second case, even if there is no obvious locking-in, experimental data exist at velocity pulsation amplitudes between 10 % and 20 % that indicate that there could still be errors in the indicated flowrate of the order of 10 %.

Annex A
(informative)

Orifice plates, nozzles and Venturi tubes — Theoretical considerations

A.1 General

Most of the recommendations in this Document concerning differential pressure-type flowmeters are based on the fact that, unless the amplitude of the pulsation in a pulsating flow is reduced to an acceptable level, an accurate value of time-mean flowrate is not obtained by substituting the time-mean differential pressure in the ordinary steady flow equation. At sufficiently small pulsation amplitudes it is possible to relate the error in the indicated flowrate to pulsation amplitude and this has been done in the derivation of the equations for adequate damping.

The above approach has been adopted partly because at significant pulsation amplitudes the flowmeter installation would need to be fitted with specially designed fast-response secondary instrumentation and partly because at the time of publication of this Document there is insufficient understanding of the flowmeter behaviour in pulsating flows to allow accurate flow measurement.

Clauses A.2, A.3 and A.4 outline the theoretical basis for the equations relating flow-metering errors, due to pulsation in flowrates computed using time-mean differential pressures, with pulsation amplitudes and temporal inertia parameters. Clause A.5 describes an approach using CFD techniques which might eventually lead to a more complete understanding of flowmeter behaviour in pulsating flows.

A.2 Derivation of equations relating error in indicated flowrate to pulsation amplitude

These equations are derived from the assumption that the fluid can be regarded as incompressible. Experimental results obtained from placing orifice flowmeters in pulsating gas flows indicate that this assumption is only valid if:

— the expansibility factor is very nearly unity, i.e. if $\varepsilon \geq 0{,}99$, and

— the amplitude of the fluctuations in upstream density are small compared with the mean value, i.e. if $\rho'_{\mathrm{RMS}} / \bar{\rho} \leq 1/40$.

It is also assumed that the throat of the flowmeter is small compared with the pulsation quarter-wave length.

A.3 Total error in the indicated flowrate

The definition of the total error, E_{T}, is based on the fact that the indicated flowrate during pulsating flow is calculated using the measured time-mean differential pressure in the steady flow equation. It is assumed that there are no secondary-device errors.

Hence,

$$E_{\mathrm{T}} = \sqrt{\frac{\overline{\Delta p_{\mathrm{p}}}}{\Delta p_{\mathrm{ss}}}} - 1 \tag{A.1}$$

where

$\overline{\Delta p_{\mathrm{p}}}$ is the time-mean differential pressure measured in pulsating flow conditions;

Δp_{ss} is the differential pressure that would have been measured under steady flow conditions at a flowrate equal to the time-mean flowrate of the pulsating flow.

A.4 Quasi-steady temporal inertia theory

The one-dimensional flow momentum equation is

$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + \frac{1}{\rho}\frac{\partial p}{\partial x} = 0 \tag{A.2}$$

and the continuity equation is

$$A\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u A)}{\partial x} = 0 \tag{A.3}$$

If it is assumed that the flow is incompressible, these two equations can be combined and integrated with respect to x between the pressure tappings to give an equation for the instantaneous differential pressure, Δp_{p},

$$\Delta p_{\mathrm{p}} = \frac{q_m^2 (1 - C_{\mathrm{c}}^2 \beta^4)}{2\rho\left(C_{\mathrm{c}} \pi d^2 / 4\right)^2} + \rho \int_1^2 \frac{\partial u}{\partial t} \mathrm{d}x \tag{A.4}$$

The first term on the right-hand side of the equation is the differential pressure associated with the convective acceleration of the fluid through the restriction. It is identical to the expression for the differential pressure generated by a steady flow with a mass flowrate equal to instantaneous mass flowrate, q_m; thus,

$$\Delta p_{\mathrm{p}} = \Delta p_{\mathrm{ss}} + \rho \int_1^2 \frac{\partial u}{\partial t} \mathrm{d}x \tag{A.5}$$

The second term is the differential pressure associated with the temporal acceleration of the fluid. Its magnitude increases with pulsation frequency and is zero for steady flow. The integral cannot be evaluated exactly but can be replaced by an equivalent term:

$$\frac{L_{\mathrm{e}}}{\left(C_{\mathrm{c}} \pi d^2 / 4\right)} \frac{\mathrm{d}q_m}{\mathrm{d}t} \tag{A.6}$$

where L_{e} is the effective axial length of the restriction.

The instantaneous mass flowrate, q_m, can be expressed by

$$q_m = \overline{q_m}\left[1+\phi(t)\right] \tag{A.7}$$

where

$$\phi(t) = \sum_{r=1}^{\infty} a_r \sin(r\omega t + \theta_r) \tag{A.8}$$

and

$$\frac{\mathrm{d}q_m}{\mathrm{d}t} = \overline{q_m}\sum_{r=1}^{\infty} r\omega a_r \cos(r\omega t + \theta_r) \tag{A.9}$$

For a steady flow:

$$\Delta p_{\mathrm{ss}} = \frac{\overline{q_m}^2\left(1-C_\mathrm{c}^2\beta^4\right)}{2\rho\left(C_\mathrm{c}\pi d^2/4\right)^2} \tag{A.10}$$

and hence for a pulsating flow:

$$\Delta p_\mathrm{p} = \Delta p_{\mathrm{ss}}\left\{\left[1+\phi(t)\right]^2 + \frac{2J\phi'(t)}{\omega}\right\} \tag{A.11}$$

where

$$J = \frac{2\pi C_\mathrm{c}}{\left(1-C_\mathrm{c}^2\beta^4\right)}\frac{L_\mathrm{e}}{d}\frac{fd}{\overline{U}_d} \tag{A.12}$$

and where C_c is the contraction coefficient for an orifice (C_c = 1 for a nozzle or Venturi tube). By integrating Formula (A.11) with respect to time

$$\overline{\Delta p_\mathrm{p}} = \Delta p_{\mathrm{ss}}(1+\overline{\phi^2}) \tag{A.13}$$

and thus

$$E_\mathrm{T} = \sqrt{\frac{\overline{\Delta p_\mathrm{p}}}{\Delta p_{\mathrm{ss}}}} - 1 = \left[1+\left(U'_{\mathrm{RMS}}/\overline{U}\right)^2\right]^{1/2} - 1 \tag{A.14}$$

where U'_{RMS} is the root-mean-square amplitude of the fluctuating component (i.e the a.c. component) of velocity U and is defined as

$$U'_{\mathrm{RMS}} = \overline{U}\left(\sum_{r=1}^{\infty}\frac{a_r^2}{2}\right)^{1/2} \tag{A.15}$$

The ratio $(U'_{\mathrm{RMS}}/\overline{U})$ is equal to $(q'_{m,\mathrm{RMS}}/\overline{q_m})$ for incompressible flow and is the velocity pulsation amplitude ratio.

Amplitudes of DP pulsation might be easier to measure than those of velocity pulsation, however, and it is, therefore, useful to define

$$\Delta p'_{\mathrm{p,RMS}} = \sqrt{\overline{\left(\Delta p_{\mathrm{p}} - \overline{\Delta p}_{\mathrm{p}}\right)^2}} \tag{A.16}$$

From Formula (A.11) and neglecting terms of secondary importance

$$\frac{\Delta p'_{\mathrm{p,RMS}}}{\Delta p_{\mathrm{ss}}} = \sqrt{\left(4\overline{\phi^2} + \frac{4J^2}{\omega^2}\overline{\phi'^2}\right)} \tag{A.17}$$

The harmonic distortion factor, H, is introduced

where

$$H = \left(\frac{\sum_{r=1}^{\infty} r^2 a_r^2}{\sum_{r=1}^{\infty} a_r^2}\right)^{1/2} \tag{A.18}$$

and where a_r is the amplitude of the rth harmonic in the Fourier series defined in Formula (A.8).

For a sine wave, H has a value of 1; it is greater than 1 for all other waveforms.

Thus,

$$\frac{\Delta p'_{\mathrm{p,RMS}}}{\Delta p_{\mathrm{ss}}} = 2\sqrt{\overline{\phi^2}\left(1 + H^2 J^2\right)} \tag{A.19}$$

Hence, from Formula (A.14) the following alternative equation for total error, E_{T}, may be obtained:

$$E_{\mathrm{T}} = \left[1 + \frac{1}{4}\left(\Delta p'_{\mathrm{p,RMS}} / \Delta p_{\mathrm{ss}}\right)^2 / \left(1 + H^2 J^2\right)\right]^{1/2} - 1 \tag{A.20}$$

And also from Formula (A.15)

$$E_{\mathrm{T}} = \frac{1}{2}\left\{1 + \left[1 - \left(\Delta p'_{\mathrm{p,RMS}} / \overline{\Delta p}_{\mathrm{p}}\right)^2 / \left(1 + H^2 J^2\right)\right]^{1/2}\right\}^{-1/2} - 1 \tag{A.21}$$

Both Formulae (A.20) and (A.21) involve the inertia term H^2J^2, where J is defined by Formula (A.12) and is proportional to the Strouhal number $fL_{\mathrm{e}} / \overline{U}_d$.

The effective length, L_{e}, cannot be measured directly but it is likely that $L_{\mathrm{e}} \approx d$.

Temporal inertia effects are negligible when

$$H^2 J^2 << 1$$

When this is so, the total error, E_{T}, is theoretically entirely due to the square-root effect and is only dependent on the pulsation amplitude.

In the derivation of all equations from Formula (A.4) it was assumed that the coefficient of discharge and the expansibility factor behave as constants with their steady flow values. It was also assumed that the upstream density remained constant and that U was the axial velocity in the upstream section.

A review of experimental data for orifice flow-metering error in pulsating flow conditions is given by Gajan et al.[9]. In general, the experimental data show good agreement at low pulsation amplitudes and Strouhal numbers.

A.5 Use of Computational Fluid Dynamics (CFD) to predict discharge characteristics in pulsating flow

A.5.1 Basic elements of CFD

The first step in any numerical prediction of a fluid flow is to define a region or "solution domain" within which the flow is to be calculated. The boundaries of the domain should coincide as far as possible with identifiable boundaries of the flow, e.g. walls or symmetry planes, or where values of the flow variables are known, so that physically realistic boundary conditions can be specified.

The second step is to define a grid or mesh within the domain. This might be a structured mesh, for example, a Cartesian mesh or a distorted form of Cartesian mesh, or a completely unstructured mesh, i.e. of the finite element type.

In the third step, the equations representing the conservation of mass, momentum and scalar variables (e.g. enthalpy or temperature) are integrated numerically over the individual cells to give a set of algebraic equations.

In the final step, the algebraic equations are solved by some form of numerical algorithm. The process of solving the equations invariably involves a certain amount of iteration in order that non-linearities in the equations and linkages between the equations can be removed through the use of previous iteration step values.

A.5.2 Application to pulsating flow through an orifice plate

The configuration of an orifice plate in a pipe is geometrically simple, and the two-dimensional, axi-symmetric nature of the flow means that computations can be carried out with relatively modest computing resources. Even in the most demanding case in which the flow is turbulent and pulsating, it appears feasible to calculate ensemble averaged values of the velocity and pressure fields on desk-top computers (Atkinson[22]).

Numerous workers have reported computations of the steady, turbulent flow case, including, for example, Langsholt and Thomassen[23], Reader-Harris[24] and Atkinson[22]. All of these workers employ a simple rectangular solution domain with boundaries at the centreline and pipe wall with rectangular grid cells which expand in size away from the orifice plate. Both Atkinson[22] and Reader-Harris[24] show that it is perfectly feasible to treat the orifice plate as a "thin" surface located at the boundary between adjacent rows of cells. In all cases, the turbulence has been modelled with the k-ε model in conjunction with logarithmic wall functions.

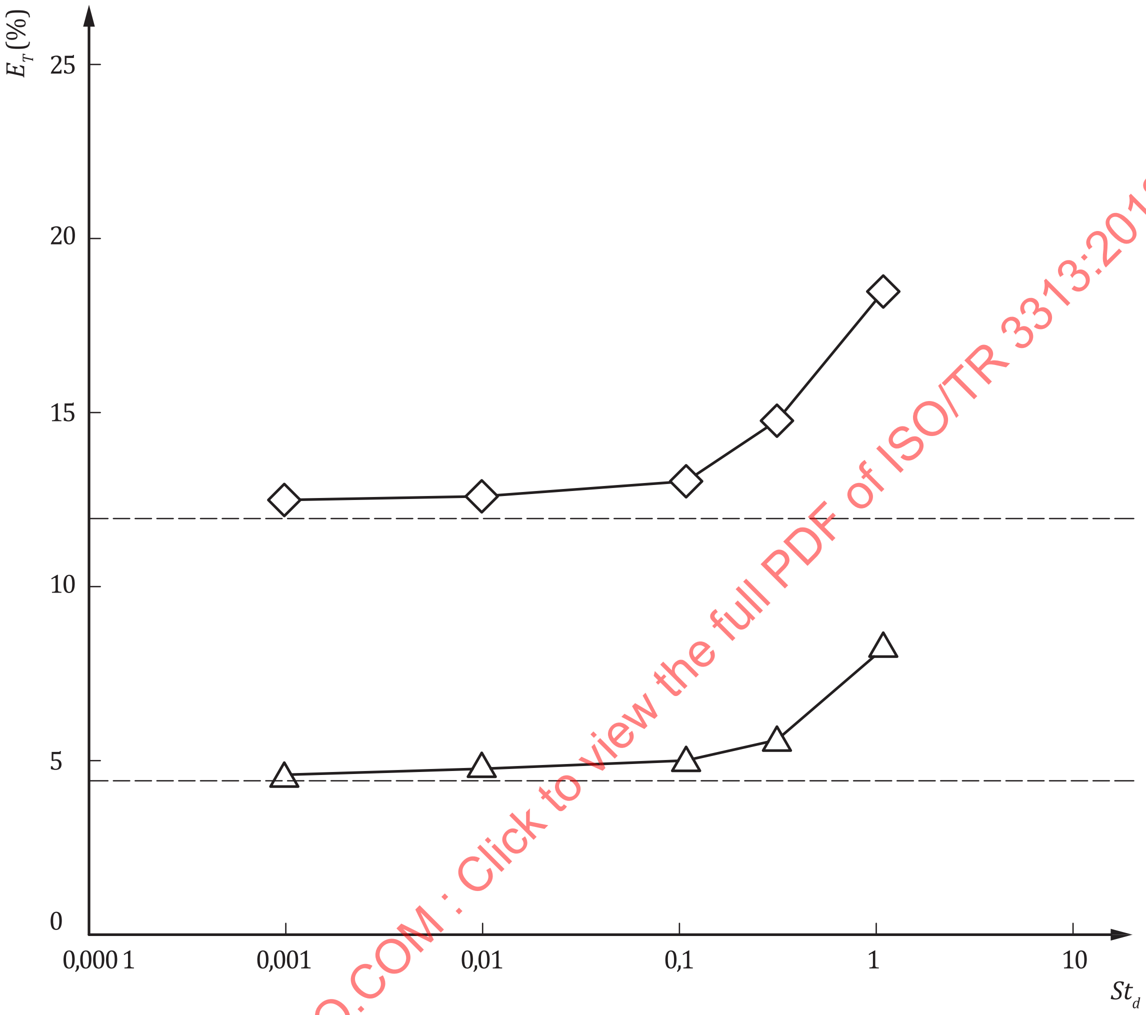
One of the major benefits of CFD is that it allows field values of all of the major variables to be obtained. This makes it possible to compute the discharge coefficient for any configuration of pressure tappings, or to compute the flow streamlines and hence determine the location of the vena-contracta and the size of the contraction coefficient.

The computations of the steady flow case appear to be very encouraging. They show, for example, that the discharge coefficient can be calculated to within about 2 % of the experimentally measured values in ISO 5167-2. In addition, details of the velocity and turbulence fields, such as the length of the separation region and the maximum value of the turbulence intensity, can also be calculated reasonably well. This suggests that the k-ε model represents the turbulence field adequately, at least for the steady flow case, and provides some grounds for using it in the pulsating case also.

The only examples of calculations of pulsating flow through an orifice plate appear to be those of Atkinson[22]. Nevertheless, they demonstrate that it is possible to obtain realistic values of both the total error and residual error of the flowmeter by recourse to CFD. The calculations are for fully turbulent flow with sinusoidal pulsation, and for different values of β and pulsation amplitude and frequency.

Figures A.1 and A.2 show the effect of frequency on the total error, E_T, and residual error, E_R, of a flowmeter with corner tappings. The total error is constant at first but increases once the flow ceases to be quasi-steady, whereas the residual error, which is negligible at first, becomes strongly negative. These trends are in good agreement with measurements by Downing[25].

Of course the same calculations also yield the total error and residual error for flange and *D*–*D*/2 tappings for which measurements are more scarce. Also, since it appears that sinusoidal pulsating flows can be calculated accurately, there seems no reason to suppose that flows with more complex waveforms cannot be calculated equally well. Perhaps the main advantage of CFD is that it can be used to calculate flows which would be difficult to set up in the laboratory and time-consuming to measure.



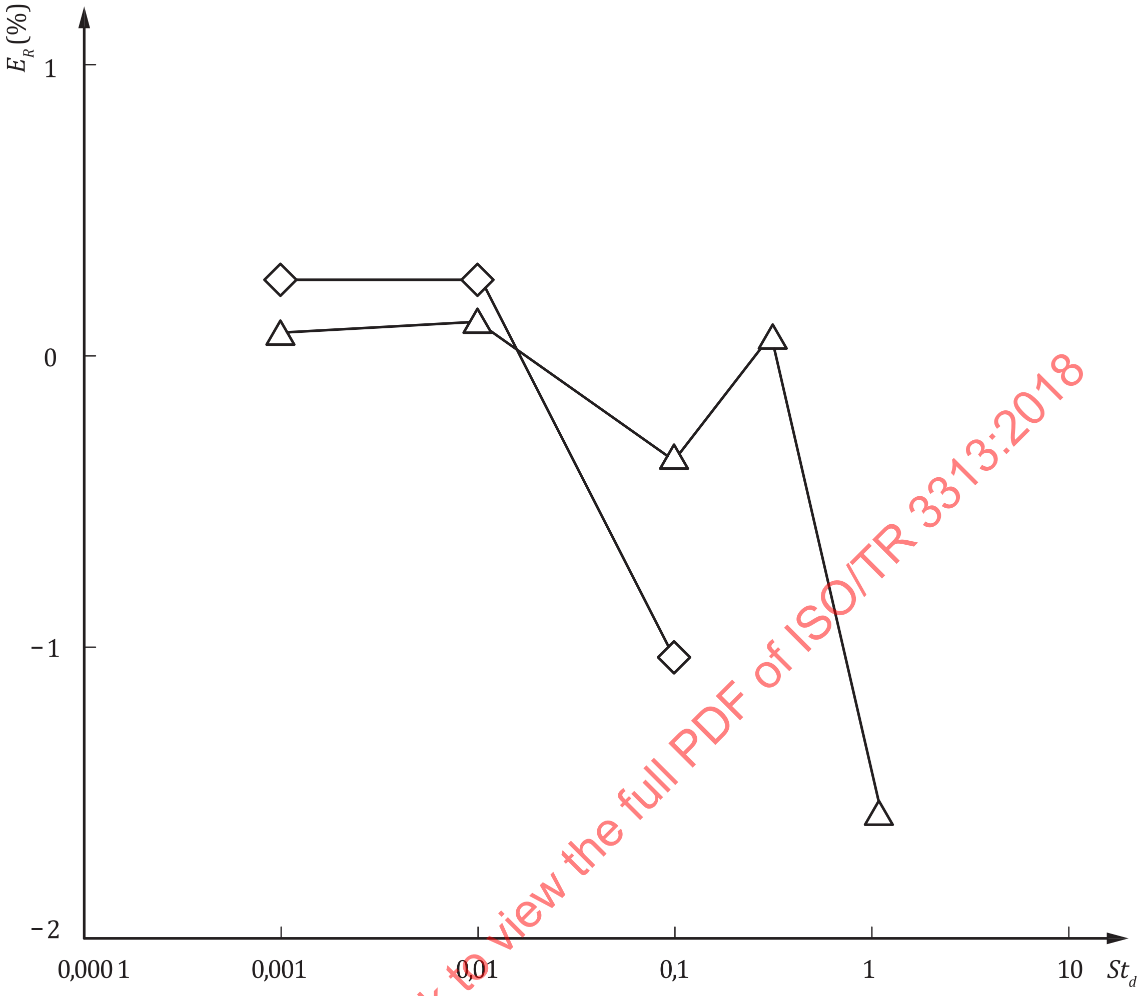
Key

◇ $\alpha = 0{,}5$, $\beta = 0{,}7$

△ $\alpha = 0{,}3$, $\beta = 0{,}7$

---- quasi-steady theory, $\beta = 0{,}7$

Figure A.1 — Total error for an orifice plate in pulsating flow with corner tappings



Key

◇ α = 0,5, β = 0,7

△ α = 0,3, β = 0,7

Figure A.2 — Calculated residual error, E_R, for an orifice plate in pulsating flow with corner tappings

Annex B
(informative)

Orifice plates, nozzles and Venturi tubes — Pulsation damping criteria

B.1 General

In the 1920s J. L. Hodgson published work[26][27] demonstrating that a dimensionless number could be used as a criterion for adequate damping of pulsating gas flow. Later this criterion was slightly modified by Ruppel[28] and the number which has become known universally as the Hodgson number is defined as follows:

$$Ho = \frac{V}{\left(\overline{q_V} / f_p\right)} \frac{\overline{\Delta\varpi}}{\overline{p}} \tag{B.1}$$

where

- V is the volume of the receiver and the pipework between source and flowmeter;
- $\overline{q_V}$ is the time-mean volumetric flowrate at the mean density in the receiver;
- f_p is the fundamental pulsation frequency;
- $\overline{\Delta\varpi}$ is the overall time-mean pressure loss between the receiver and the source of supply (or discharge) at constant pressure;
- $\overline{p}$ is the mean absolute static pressure in the receiver.

Hodgson himself related his criterion to certain specific pulsation amplitudes and waveforms typical of slow-speed reciprocating steam engines.

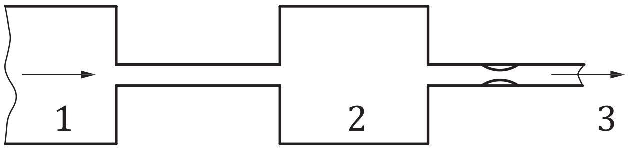
Later Lutz[29], Herning and Schmidt[30], and Kastner[31] presented work concerning the measurement of airflow inducted by internal combustion engines. This work broadly confirmed the validity of Hodgson's damping criterion. More recently, Fortier[32] presented a theoretical analysis in which values of the Hodgson number required for adequate damping could be predicted for both sinusoidal and rectangular waveforms.

Mottram[33] presented an analysis similar to Fortier's but made it applicable to pulsation of any waveform. A summary of these analyses, slightly modified, is presented in B.2. Only the single-receiver damping system is considered in B.2. Further information on the theoretical background to divided-receiver damping systems can be found elsewhere[12–15].

B.2 Theoretical analysis for adequate damping — Subsonic flow in the throttling device

The following analysis is based on the "lumped element" theory, i.e. the pipe lengths are assumed to be short compared with pulsation wavelengths. Damping vessel design based on a more sophisticated "distributed element" analysis requires knowledge of the acoustic characteristics of each particular flow system.

Consider a system in which a pulsating gas flow passes through a receiver to a throttling device including an orifice flowmeter located in a length of pipe terminating in a constant pressure reservoir, i.e. the atmosphere (see Figure B.1).



Key

1 pulsation source, instantaneous flowrate q_{mo}

2 damping vessel

3 flowmeter/throttle, instantaneous flowrate q_m

Figure B.1 — Diagram of a system with a damping vessel

The undamped and damped mass flowrates can be represented by a Fourier series and hence

$$q_{mo} = \overline{q_m}\left[1+\sum_{r=1}^{\infty} b_r \sin(r\omega t+\theta_{or})\right] \tag{B.2}$$

and

$$q_m = \overline{q_m}\left[1+\sum_{r=1}^{\infty} a_r \sin(r\omega t+\theta_r)\right] = \overline{q_m}\left[1+\phi(t)\right] \tag{B.3}$$

where

b_r and a_r are the amplitudes of the rth harmonic components in the series representing the undamped and damped flows respectively;

θ_{or} and θ_r are phase angles which might be different for high harmonic components.

The instantaneous differential pressure, Δp_p, across the throttling device is always given by an equation which has the general form

$$\Delta p_p = \Delta p_{ss}\left\{\left[1+\phi(t)\right]^n + B\phi'(t)\right\} \tag{B.4}$$

where

Δp_{ss} is the differential pressure for a steady flow of mass flowrate $\overline{q_m}$ through the throttling device;

n is an exponent which depends on the structure of the throttling device but is generally equal to 2;

$$B = \frac{1}{l_e}\frac{\overline{q_m}}{\Delta p_{ss}} \tag{B.5}$$

where l_e is an effective length of the throttling device.

Since the dimensions of the damping vessel are assumed to be small compared with the pulsation wavelength, continuity of mass flow is preserved if

$$V\frac{\mathrm{d}\rho}{\mathrm{d}t}+q_m=q_{mo} \qquad \text{(B.6)}$$

If the processes in the damping chamber can be assumed to be isentropic, then

$$\frac{\mathrm{d}\rho}{\mathrm{d}t}=\frac{1}{c^2}\frac{\mathrm{d}p}{\mathrm{d}t} \qquad \text{(B.7)}$$

where c is the speed of sound (= $\sqrt{\kappa p/\rho}$ for an ideal gas).

The term dp/dt can be evaluated from Formula (B.4) if it is further assumed that

$$\Delta p_\mathrm{p}=p-p_\mathrm{a} \qquad \text{(B.8)}$$

where

p is the instantaneous pressure in the damping vessel;

p_a is a constant pressure.