EFRC Guidelines

EFRC
EUROPEAN FORUM for RECIPROCATING COMPRESSORS

Flow Meter Errors in Pulsating Flow of Reciprocating Compressor Systems

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Contents

Foreword

The EFRC is the European Forum for Reciprocating Compressors which was founded in 1999 by Neuman & Esser, Leobersdorfer Maschinenfabrik, Hoerbiger Ventilwerke, TNO, TU Dresden, Thomassen Compression Systems, Wärtsila Compression Systems and Burckhardt Compression. The target of the EFRC is to serve as a platform to facilitate the exchange of information between vendors, operators and scientists working in the field of reciprocating compressors.
This is achieved by knowledge transfer (conferences, internet, student workshops, training and seminars), standardisation work (e.g. EFRC Guidelines, API 618, ISO 20816-8), and by joint pre-competitive research projects, aiming at improving the performance and the image of the reciprocating compressor.
In the research and standardisation projects, the forces of all interested parties are combined to solve or investigate problems which are beyond the scope of a single player. These projects are mainly carried out at research institutes or universities and in this way, they serve as the scientific arm of the reciprocating compressor community.

The research and standardisation working groups are open to all EFRC members and the annual budget is funded by participating members. The results are owned by the EFRC and the research results are disclosed to EFRC research group members only. The work of the standardization group is made publicly available via the EFRC website (www.recip.org).

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1 Terms, Definitions and Abbreviations

1.1 Definitions

1.1.1 General

Purchaser
Agency that issues the order and specification to the vendor.

Manufacturer
A person, an enterprise, or an entity that manufactures equipment.

Vendor
Vendor (OEM) or vendor's agent that supplies the equipment.

1.1.2 Pressure measurement

Differential pressure (Δp)
Difference between the (static) pressures measured at the wall pressure tapping's, one of which is on the upstream side and the other of which is on the downstream side of a primary device (or in the throat for a Venturi nozzle or a Venturi tube), inserted in a straight pipe through which flow occurs, when any difference in height between the upstream and downstream tapping's has been considered.

NOTE In ISO 5167 (all parts) the term "differential pressure" is used only if the pressure tapping's are in the positions specified for each standard primary device.

Pressure ratio
Ratio of the absolute (static) pressure at the downstream pressure tapping to the absolute (static) pressure at the upstream pressure tapping.

Static pressure of a fluid flowing through a pipeline (p)
Pressure which can be measured by connecting a pressure-measuring device to a wall pressure tapping.

NOTE Only the value of the absolute static pressure is considered in ISO 5167 (all parts).

Wall pressure tapping
Annular slot or circular hole drilled in the wall of a conduit in such a way that the edge of the hole is flush with the internal surface of the conduit.

NOTE The pressure tapping is usually a circular hole but in certain cases may be an annular slot.

1.1.3 Differential (DP) pressure meter

Nozzle
Device which consists of a convergent inlet connected to a cylindrical section generally called the "throat".

Orifice throat
Opening of minimum cross-sectional area of a primary device.

NOTE Standard primary device orifices are circular and coaxial with the pipeline.

Orifice plate
Thin plate in which a circular opening has been machined.

NOTE Standard orifice plates are described as a “thin plate” and with a “sharp square edge”, because the thickness of the plate is small compared with the diameter of the measuring section and because the upstream edge of the orifice is sharp and square.

Venturi nozzle
Device which consists of a convergent inlet which is a standardized ISA 1932 nozzle connected to a cylindrical part called the “throat” and an expanding section called the “divergent” which is conical.

Venturi tube
Device which consists of a convergent inlet which is conical connected to a cylindrical part called the “throat” and an expanding section called the “divergent” which is conical.

1.1.4 Flow

Flow disturbances
Elements in the piping such as tees, elbows, reducers, inlet and outlet of volumes, partially closed valves, butterfly and check valves, control valves, which disturb the ideal uniform velocity profile and generate additional turbulence.

Flow conditioner
Is an element placed in the flow which ensures that the “real world” environment closely resembles the “laboratory” environment for proper performance of flow meters such as orifice, turbine, Coriolis, ultrasonic etc.

Flow straightener
A flow straightener, sometimes called a honeycomb, is a device used to straighten the flow. It is a passage of ducts, laid along the axis of main gas stream to minimize the lateral velocity components caused by swirling motion in the air flow during entry.

Flowrate, rate of flow (q)
Mass or volume of fluid per unit time.

Mass flowrate, rate of mass flow (q_m)
Mass of fluid passing through the orifice (or throat) per unit time.

Reynolds number (Re)
Dimensionless parameter expressing the ratio between the inertia and viscous forces.

Steady state flow
Flow in which parameters such as velocity, pressure, density and temperature do not vary significantly enough with time to prevent measurements to within the requirement uncertainty of measurements.

Volume flowrate, rate of volume flow (q_v)
Volume of fluid passing through the orifice (or throat) per unit time.

NOTE In the case of volume flowrate, it is necessary to state the pressure and temperature at which the volume is referenced.

1.1.5 Pulsations and vibrations

Anti-node (pressure or flow)
Location where the pressure or velocity pulsation is maximal.

External pipe vibrations
Vibrations of the pipe in which the flow meter is installed which are caused by pulsation forces or by mechanical forces (e.g. unbalanced forces and moments) of the reciprocating compressor system. They can also be caused by vibrations from other rotating equipment in close vicinity of the reciprocating compressor via e.g. the foundation or pipe supports.

Helmholtz resonance
A Helmholtz resonance is the phenomenon of gas resonance in one or more cavities, such as when one blows across the top of an empty bottle.

High-frequency broadband noise
High-frequency broadband noise are pressure fluctuations occurring over a broad frequency range, with a stochastic character (e.g. turbulence-induced noise), in excess of a typical frequency of 1 kHz.

Low frequency pulsations
Pulsations with a frequency below 200 Hz.

Mechanical Natural Frequency (MNF)
Mechanical Natural frequency, also known as eigenfrequency or resonance frequency, is the frequency at which a system tends to oscillate in the absence of any driving or damping force. The motion pattern of a system oscillating at its natural frequency is called the normal mode (if all parts of the system move sinusoidally with that same frequency).

Node (pressure or flow)
Location where the pressure or velocity pulsation amplitude is minimal/zero.

Pressure pulsations
Fluctuations around the mean static *pressure* as a function of time.

Peak-to-peak amplitude
Peak-to-peak amplitude is the difference between the peak with highest amplitude and the peak with lowest amplitude, see example of the velocity pulsation U'_{pp} in Figure 1.1.

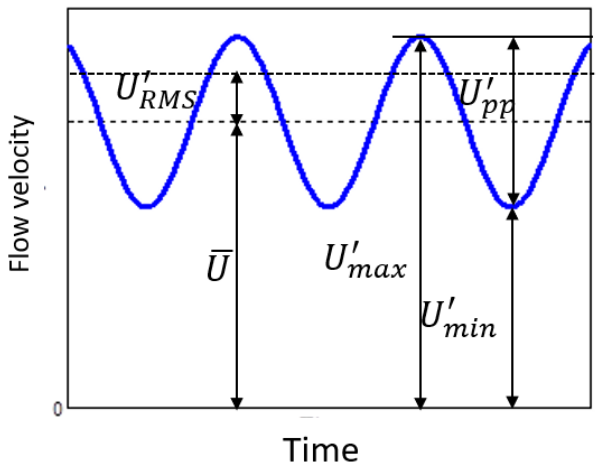


Figure 1.1 Definitions of different *velocity* velocities as a function of time, see also (1-1), (1-2) and (1-3)

Pulsation frequency f_p
Frequency of the velocity or pressure pulsations.

Root Mean Square (RMS)
RMS is defined as the square root of the mean over time and can be calculated as follows:

$$U'_{RMS} = \sqrt{\frac{1}{T}\int_0^T \acute{U}\,(t)dt} \qquad (1\text{-}1)$$

In which:
- $\acute{U}$ (t) time dependent signal (pressure of velocity pulsations)
- T sampling time
- U'_{RMS} Root Mean Square value

If the peak-to-peak values of e.g. the velocity pulsations are known, the root mean square value for *one* dominant frequency can be calculated as follows:

$$U'_{RMS} = \frac{1}{2\sqrt{2}}\, U'_{pp} \qquad (1\text{-}2)$$

In which:
U'_{RMS} root mean square value of the *velocity* pulsation [m/s]
U'_{pp} peak-to-peak of the *velocity* pulsation level [m/s], see Figure 1.1

If more components contribute to e.g. the total *velocity* pulsations, the root mean square value can be calculated as follows:

$$U'_{RMS} = \sqrt{{U'_1}^2 + {U'_2}^2 + {U'_3}^2 \ldots {U'_n}^2} \qquad (1\text{-}3)$$

In which:
U'_{RMS} root mean square value of *velocity* pulsation
$U'_1 \ldots . U'_n$ root mean square value for each dominant frequency component

Strouhal number
Is a dimensionless number describing the frequency of oscillating flow mechanisms.

Standing wave resonance
Standing waves are formed from two or more traveling waves (incident and reflecting) that collide and are “in phase” with each other in such a way that their amplitudes add (called constructive interference) or subtract (called destructive interference) in repetitive ways which is called “resonance”.

Spectrum
Description of a quantity as a function of frequency or wavelength.

Velocity pulsations
Fluctuations of the mean static *velocity* as a function of time. This is also often named *flow* pulsations. In these guidelines *velocity* pulsations are used which is according to ISO TR 3313 [2].

1.1.6 Pressure differential flow meter

Gauge line (secondary device)
Line between the orifice taps and the transmitter, flow computer or chart recorder.

Inertial error
Error which occurs when the velocity pulsations lag behind the differential pressure changes due to the inertia effect of the fluid resulting in a distortion between the relationship of pressure and flow.

Square root error
Error which occurs if the calculation of the pulsating flow is done by using the square root of the time-averaged pressure difference ($\sqrt{\overline{\Delta P}}$), instead of the time-averaged value of the square root of the instantaneous pressure difference.

1.1.7 Turbine flow meter

Blade aspect ratio
Ratio between blade length and rotor radius.

Blade chord
An imaginary straight line joining the leading edge of a turbine blade with its trailing edge.

Dynamic response parameter
Is a property of the flow meter/fluid combination, not of the flow meter alone.

1.1.8 Vortex flow meter

Kármán vortex street
Kármán vortex street (or a von Kármán vortex street) is a repeating pattern of swirling vortices, caused by a process known as vortex shedding.

*Vortex frequency or "*von Kármán" vortex *frequency f_v*
Frequency of the vortex shedding around a bluff body.

Vortex shedding
In fluid dynamics, vortex shedding is an oscillating flow that takes place when a fluid such as air or water flows past a bluff body (as opposed to streamlined body). The vortex shedding frequency *fv* depends on flow velocity and the geometry of the object and is characterized with the non-dimensional Strouhal number.

1.1.9 Coriolis flow meter

Coriolis frequency
Refers to the frequency of the vibration mode corresponding to the shape of the distortion introduced by the Coriolis forces (e.g. for a simple straight tube meter driven in its first (fundamental) mode, the Coriolis frequency would be the frequency of the second mode).

NOTE; for a meter not driven in a fundamental mode, there will be two Coriolis frequencies, an upper Coriolis frequency, above the drive frequency and a lower Coriolis frequency below the drive frequency.

Drive frequency
Refers to the frequency of the mode of the driven motion.

Drive system
A means for inducing the oscillation of the tube(s).

Flow sensor (primary device)
Mechanical assembly consisting of an oscillating tube(s), drive system measurements sensor(s), supporting structure, and housing.

Housing
Environmental protection of the flow sensor and/or transmitter.

Oscillating tube
Tube through which the fluid to be measured flows.

Transmitter (secondary device)
Electronic control system providing the drive electrical supply and transforming the signals from the flow sensor to give output(s) of measured interfered parameters.

NOTES
1. It also provides corrections derived from parameters such as temperature.
2. The transmitter is either integrally mounted (compact device) on the flow sensor (secondary device) or remotely installed away from the primary device and connected by a cable.

Sensing device
Sensor to detect the effect of Coriolis force and to measure the frequency of the tube oscillations.

Supporting structure
Mechanical support for the oscillating tube(s).

Zero offset
Indicated flow when there are zero flow conditions present at the meter.

1.1.10 Ultrasonic flow meter

Aliasing
Is an effect that causes different signals to become indistinguishable (or aliases of one another) when sampled. It also often refers to the distortion or artefact that results when a signal reconstructed from samples is different from the original continuous signal.

1.2 Subscripts and superscripts

p	measured under pulsating flow conditions
RMS	Root Mean Square
ss	measured under steady-state flow conditions
'	fluctuating component about mean value in flow and pressure
$\bar{}$	time mean value of flow, pressure

2 Introduction

In refinery, chemical plants, natural gas transport systems and underground gas storage plants, flow meters are installed for custody transfer or pressure- or flow-control purposes.

In case of flow metering errors, severe consequences may occur. For example, the stability of the process control may be disturbed, leading to unwanted and even dangerous situations. In addition, systematic metering errors may have negative economic consequences for suppliers and consumers (custody transfer).

Various disturbance mechanisms may affect the flow metering accuracy. A well-known disturbance mechanism is distortion of the inlet flow toward the flow meter.

Another disturbing effect may be due to pulsation effects in the fluid flow. Positive displacement machinery, such as reciprocating compressors, generate pulsations (both pressure and flow) which can have a significant effect on the accuracy of different types of flow meters, even when the *pressure* pulsations are below the limits as stipulated in the API Standard 618 [1].

It has been recognised that there is a lack of knowledge on this subject, and that the importance is underestimated or not recognised during the selection of the flow meter and design of the system layout. For that reason, it was decided by the standardisation group of the European Forum for Reciprocating Compressors (EFRC) to develop these guidelines to close the gap in this area.

These EFRC guidelines establish procedures and guidelines how to determine errors in flow meters and gives guidelines how to minimise these flow meter errors to acceptable values.
Using these guidelines together with e.g. the API Standard 618, ISO TR 3313 [2], other standards, guidelines and publications as summarised in the reference list, is indispensable. In other words, these guidelines can be used as a complementary document to the available standards and guidelines.

These guidelines were specifically developed for reciprocating compressor systems but (parts of) can also be used e.g. for other positive displacement machinery such as screw compressors, reciprocating pumps, etc.

The scope of these guidelines is focussed on the following flow meters which are commonly applied in reciprocating compressor systems: pressure differential (DP), vortex, turbine, ultrasonic and Coriolis flow meters.
Its applicability is focussed on gas systems and for conditions in the measuring section where the flow direction does not reverse, where any density changes are small, where the flow remains subsonic and where the gas can be considered as single-phase.

The guideline gives a general description of the flow measurement's principle, the errors in flow reading and methods for correcting flow meter output signals produced by pulsations and external vibrations for which it is possible to keep the flow meter error within acceptable limits.

These guidelines have been developed, based on experience and a survey of the most recent literature and existing guidelines and standards. Furthermore, a limited set of interviews was conducted, with leading experts from various branches of the industry (end users, EPC contractors, compressor OEM and flow meter manufacturers).

The first 5 chapters give general back ground information which will give the reader a good overview of several flow meter types, their sensitivity to errors in a pulsating flow, threshold values of pulsations and allowable flow meter errors, and some best practices for handling issues with flow meter accuracy in reciprocating compressor systems. By reading these first 5 chapters the reader will get a good understanding on the effect and consequences of flow meter errors in pulsation flow, focussing on reciprocating compressor systems.

The remainder of the document will give detailed information for each type of flow meter, how to calculate the flow meter error for each and the possible methods to reduce the measuring error.

This document is organised in the following way:

Chapter 3 gives a short summary of the flow meters and their sensitivity to disturbing phenomena such as flow, pulsations and external vibrations, causing flow meters errors. It gives also a summary of the effect of flow meter errors in custody transfer, flow control and acceptance tests.

Chapter 4 gives an explanation of threshold values for pulsation effects and allowable limits for flow meter errors as stipulated in several different international standards for different type of flow meters.

Chapter 5 gives best-practices for handling issues with flow meter accuracy in an early system design and the selection of the optimum flow meter for reciprocating compressor systems.

Chapter 6, 7, 8, 9, and 10 give the working principle for respectively the differential pressure, turbine, vortex, Coriolis and ultrasonic meter types. It also gives a summary of the different phenomena causing a flow meter error and several methods to reduce them.

Chapter 11 gives a summary of the standards, guidelines and publications which have been used in the development of these guidelines.

Appendix A gives some background information on acoustic wave theory which can be used to understand the difference between pressure and velocity pulsations, acoustic resonances etcetera. This information is particularly important in finding the optimum methods and solutions to reduce the flow meter errors.

Appendix B gives illustrations for some practical examples of a pulsation analysis to show the optimum locations of flow meters.

Appendix C gives the rotor response parameter equations of a turbine meter.

Appendix D gives a flow chart for each flow meter with an overview of flow meter errors, calculation methods and how to reduce them for each flow meter as discussed into detail in these guidelines.

3 Overview of flow meters and their sensitivity to disturbing phenomena

3.1 Working principle of the flow meters

A brief explanation of the working principle of the flow meter included in these guidelines is as follows:

Pressure Difference (DP) meter (see section 6.1 for a more detailed description)
A pressure differential meter works on the principle of partially obstructing the flow in a pipe. This creates a difference in the static pressure between the upstream and downstream side of the device and this pressure difference is a measure of the flow rate. Differential-pressure meters are very popular due to the relatively simple design and low costs.

Turbine meter (see section 7.1 for a more detailed description)
A turbine flow meter uses a free-spinning turbine wheel to measure fluid velocity, much like a miniature windmill installed in the flow stream. The turbine blades will achieve a rotating (tip) velocity directly proportional to the linear velocity of the fluid.

Vortex meter (see section 8.1 for a more detailed description)
When the medium flows around the bluff body which is located in the main flow, an alternately arranged vortex belt is generated behind the sides of the bluff body, called the "von Kármán vortex". The frequency of these pulsations is measured and is directly proportional to the flow rate.

Coriolis meter (see section 9.1 for a more detailed description)
The basic operation of Coriolis flow meters is based on the principles of motion mechanics. As fluid moves through a vibrating tube, it is forced to accelerate as it moves towards the point of peak-amplitude vibration. Conversely, decelerating fluid moves away from the point of peak amplitude as it exits the tube. The result is a twisting reaction force and the motion of the opposite parts of the tube are no longer in phase caused by the Coriolis forces. The mass flow rate is in direct proportion to the phase difference.

Ultrasonic meter (USM) (see section 10.1 for a more detailed description)
An USM is a type of flow meter that measures the velocity of
a fluid with ultrasound to calculate the volume flow. The ultrasonic transducers measure the average velocity by measuring the transit time between the pulses of ultrasound propagating into and against the direction of the flow.

3.2 Summary of disturbing phenomena (flow distortion, pulsations and external vibrations)

Different phenomena (flow distortion, pulsations and external vibrations) can cause flow meter errors. The sensitivity to these phenomena strongly depends on the type of flow meter and an overview of the flow meters as discussed in these guidelines is given in Table 3.1. An overview of the (relative) ranking of the phenomena affecting the primary measurement principle is summarised in Table 5.1 of chapter 5.

A short explanation of the different phenomena is as follows:

Flow disturbances
Unfavourable pipe configuration can create flow disturbances that affect the velocity profile which can cause erroneous flow measurement depending on the type of flow meter. Flow disturbers include tees, elbows, and reducers, inlet and outlet of volumes, etc. Partial obstructions such as partially closed valves, butterfly valves, check valves, and control valves will also disturb the uniform flow profile. These disturbances may occur over considerable lengths downstream of the obstruction. Even a poorly installed flange gasket may create problems.
Also, measuring probes which are installed in the flow, such as e.g. a temperature or gas composition probe can also create flow disturbances. The fluid boundary layer along the probe will separate, causing abrupt transitions in the flow path and causing shear layers to form vortices behind the construction, named vortex shedding.

Vortex shedding problems, in general do not give problems in reciprocating compressor systems, as the intrinsic disturbing dynamics caused by the compressor dominate the instability effects of the flow.

Low-frequency pulsations
Low-frequency pulsations are pulsations with a frequency below 200 Hz, mostly with a clear tonal character (generally occurring at multiple orders of the rotation speed of the source). The most common examples are pulsations due to positive displacement machinery, such as reciprocating compressors and plunger pumps. However, flow-induced pulsations can also occur in the low-frequency range (e.g. due to dead-leg side branches, excited by grazing flow). Pulsations which are caused by the reciprocating action of a reciprocating compressor have a frequency below 200 Hz and are in general the most severe pulsations caused in these systems compared to other pulsation sources as explained below. These pulsations can reach a large amplitude when an acoustic resonance is excited. More detailed explanation on acoustic resonances is given in Appendix A and Appendix B.

High-frequency broadband noise
High-frequency broadband noise is typically generated at pressure let-down devices. The in-line noise generation is often accompanied by near-field and far-field noise radiation. The intense acoustic energy has a stochastic character and can cause severe pulsation and vibration issues, leading to fatigue failure within relatively short periods. In addition, it can also lead to flow meter errors. Most common examples of high-frequency broadband noise sources include:

- Pressure relief valves.
- Blowdown valves.
- Pressure reducing valves.
- Recycling valves.
- Control valves.
- Restriction orifice plates in blow-down lines or recycle lines.
- High-velocity flow in particular when facing elbows or t-joints, for example in relief headers.

External pipe vibrations

Positive displacement machinery such as reciprocating compressors generate vibrations which are caused by e.g. pulsation forces and by mechanical loads (e.g. unbalanced forces and moments) of the compressor. They can also be caused by vibrations of other equipment in close vicinity of the reciprocating compressor via e.g. the foundation or pipe supports, named structure borne vibrations. It has been shown that several type of flow meters are sensitive to external pipe vibrations. There might be cases that the flow meter indicates a flow meter reading while there is no flow in the pipe section in which the flow meter is installed.

Table 3.1 Overview of **relevant disturbances** for different type flow meters **which can cause a flow meter error**

Type of Flow meter	**Applicable chapter number**	**Relevant disturbances which can cause a flow meter error**	**Applicable standards**
DP meter (orifice, nozzle, venturi)	Chapter 6	• Flow disturbances • Low frequency pulsations • High frequency broadband pulsations	ISO 5167 ISO TR 3313 AGA Report 3 API MPMS
Turbine	Chapter 7	• Flow disturbances • Low frequency pulsations • High frequency broadband pulsations	ISO 9951 ISO TR 3313
Vortex	Chapter 8	• Flow disturbances • Low frequency pulsations • High frequency broadband pulsations • External vibrations	ISO TR 12764 ISO TR 3313
Coriolis	Chapter 9	• Low frequency pulsations • High frequency broadband pulsations • External vibrations	ISO 10790 AGA Report 10 AGA Report No. 11 API MPMS
Ultrasonic	Chapter 10	• Flow disturbance • Low frequency pulsations • High frequency broadband pulsations	ISO TR 12765 AGA Report 9

Notes
- The relevant disturbances are listed, considering the physical measurement principles of each flow meter type.
- Considerable differences in observations between different brands and instrument types have been found in the literature on this topic.
- The relevance of disturbance caused by flow disturbances is based on the fact that sufficient inlet length is used, in accordance with the applicable industry standards and guidelines.

3.3 Effect of flow meter error in custody transfer, process control and acceptance tests

If a flow meter is used for pressure or process control purposes, significant flow meter errors can lead to unwanted and even dangerous situations (unstable operation of the system) which should be avoided. Moreover, flow meters are also used to check the performance of a newly installed reciprocating compressor and errors should be kept to an acceptable level.

If a flow meter is used for custody transfer, the flow meter error shall be minimised to avoid that the buyer of the gas shall pay too much for systems with flow meters with a positive error or too little for flow meters with a negative error.

The maximum flow meter error shall be specified by the purchaser, see also section 4.5, and shall be considered during the design stage, during acceptance tests and during operation or trouble-shooting (in case of less delivered flow) of the reciprocating compressor system.

The phenomenon of flow meter errors in pulsating flow is described in detail in different standards e.g. the API Standard 688, several ISO standards such as the ISO TR 3313 [2] and in several other publications. These literature sources give, in general, a good background about the cause of the possible flow meter errors and methods to reduce the flow meter errors.

In case that the flow meter is used for acceptance tests it is pointed out that the measured flow may deviate from the actual flow caused by flow meter errors. This is the case if the flow meter is installed at a location where pulsations are present. For those field cases it is difficult to quantify the flow meter error which is due to the fact that the flow meter error of several of the flow meter types, as discussed in these guidelines, is dependent on the *velocity* pulsations.

In contrast to *pressure* pulsations, *velocity* pulsations cannot be measured directly very easily. All techniques to measure flow pulsations require specialised equipment and knowledge and shall be done for that reason by a specialist. Probably the easiest method is to use the two-microphone method. With this method the *pressure* pulsations are measured upstream and downstream of the flow meter and the *velocity* pulsations can be calculated accordingly. More detailed information about the two-microphone method can be found in several publications [28, 29, 30].

Velocity pulsations can also be measured with other special techniques e.g. with non-intrusive techniques like laser Doppler anemometry, Doppler shift and electromagnetic flow meters. An insertion device such as thermal anemometry can also be used.
It is noted that all methods to measure the *velocity* pulsations are in general applied in laboratory applications and not in the field.

In general, it means that the most optimal way to determine the flow meter error, both during the design stage and field acceptance tests, is to calculate it with a detailed pulsation analysis as specified in the API Standard 618.

It might also happen that much tighter *pressure* pulsation limits than those defined by the API Standard 618 are defined by the purchaser for a large part of the pipe system.
However, as explained *velocity* pulsations rather than *pressure* pulsations determine the flow meter error for several flow meter types. It might even happen that, at the location of the flow meter, acceptable *pressure* pulsations can cause high *velocity* pulsations leading to an unacceptable flow meter error. Specifying tighter overall *pressure* pulsation limits is in many cases not beneficial. Only tighter *velocity* pulsation limits in the vicinity of the flow meter will reduce the measuring errors. Several examples which make this clearer are given in Appendix B.

It has been recognised that there is a lack in knowledge on the subject of flow meter errors by compressor vendors, engineering contractors, flow meter vendors and end users. This can lead to many discussions about responsibility and guarantee penalties. For that reason, these guidelines have been developed to serve as a discussion document between all parties involved to achieve an acceptable flow meter error at acceptable costs.

4 Generic guidelines for limit values related to flow meter errors

4.1 Introduction

A limited number of guidelines with threshold values for pulsation effects are available. However, the limit values only apply to certain types of flow meters. Moreover, the threshold values are not always easily identified or easily determined with calculations or measurements. The main features and limitations will be discussed below.

4.2 General guidelines for flow measurement

The ISO 5167 [3], [4], [5] consists of four different parts:

Measurement of fluid flow by means of pressure differential devices (DP) inserted in circular cross-section conduits:

- Part 1: General principles and requirements
- Part 2: Orifices
- Part 3: Nozzles
- Part 4: Venturi

These four parts cover the geometry and method of use (installation and operating conditions) of orifice plates, nozzles and venturi tubes when they are inserted in a conduit to determine the flowrate of the fluid flowing in the conduit. It also gives necessary information for calculating the flowrate and its associated uncertainty. It is applicable only to pressure differential devices in which the flow remains subsonic throughout the measuring section and where the fluid can be considered as single-phase.

However, due to the fact that the ISO 5167 cannot be applied to flows which contains periodic flow or pressure pulsations, a reference for the subject of pulsations was made in the four parts to the 1998 edition of the ISO TR 3313, "*Measurement of fluid flow in closed conduits — Guidelines on the effects of flow pulsations on flow-measurement instruments*".

Due to the fact that the topics on pulsations of the 1998 edition of the ISO TR 3313 were incomplete and outdated, a new revision of this standard was developed. The 2018 edition [2] includes more information on the effect of pulsations for the pressure differential (DP), vortex and turbine flow meters. It still lacks information on the Coriolis and ultrasonic meters with respect to pulsations.

The practical applicability is limited due to the fact that the ISO TR 3313 is rather complex to understand. Moreover, the threshold values for pulsating flow are very strict and only proposed for the pressure differential (DP), vortex and turbine flow meter. These strict values are not always feasible to achieve and are also rather difficult to validate. Furthermore, the equations to calculate the flow meter errors are only developed for the pressure differential (DP) and turbine flow meter types. Besides that, detailed information from the meter vendor may be required to calculate the flow measurement error.

In section 4.3, the allowable flow meter error is given as stipulated in the API Standard 618. In section 4.4, a brief summary is given of the threshold values of a pulsating flow as stipulated in the ISO TR 3313 for the pressure differential (DP), vortex and turbine flow meter types.

4.3 Limit values of flow meter errors in the API Standard 618

The 5th edition of the API Standards 618 (section 7.9.4.2.5.3.3), is a procurement document. This means that it is to be used during the design of the system, for reciprocating compressors and stipulates for flow meters the following:

Unless otherwise specified, for flow meters located in the specified piping system, the maximum flow measurement error caused by pulsation shall not exceed the following:

- *Non-custody transfer applications: 1%*
- *Custody transfer applications: 0.125%*

It is noted that the allowable value of 0.125% for custody transfer is retrieved from the threshold value for a DP meter which was first defined in the AGA Report Number 3.1. [7].

The API 618 does not make any distinction between the type of flow meter (unlike ISO-TR-3313).

It is noted that the pulsation limits in API 618 (formulated for *pressure* pulsations) are dedicated to structural integrity and not to ensure acceptable flow metering accuracy. Generally, the criterion for flow meter accuracy is much more restrictive than the 'default' *pressure* pulsation limits.

The API 618 standard does not provide a calculation method to compute the flow metering error, even if the *pressure* and *velocity* fluctuations are available from a detailed numerical analysis. This limits its practical use during detailed design. However, in section 6.2.1 and 7.2 of these guidelines a summary is given how to calculate the flow measurement error for respectively a pressure difference (DP) and a turbine meter.

4.4 Threshold values in ISO TR-3313

The ISO TR 3313 states that if the amplitude of the periodic flowrate variations is sufficiently small, there should not be any significant error in the indicated flowrate greater than the normal measurement uncertainty. It is possible to define an amplitude threshold for the differential pressure (DP) and turbine flow meter without reference to pulsation frequency.
For vortex flow meters the critical flow pulsation *amplitude* is that *amplitude* which is just sufficient to cause lock-in. Defining a threshold value for which an error will not occur is also possible for vortex flow meters. However, extreme caution is required because even for a very small pulsation amplitude a large error can occur if a so-called lock-in of the pulsation frequency with the vortex will occur. A more detailed explanation will be given in section 9.1.

For DP-type flow meters, the threshold is relevant when slow-response DP cells are being used. In the case of turbine flow meters, the threshold value is relevant when there is any doubt about the ability of the rotor to respond to the periodic velocity fluctuations (also named *flow* pulsations). In the case of a vortex flow meter the pulsation frequency relative to the vortex shedding frequency is a much more important parameter than the velocity pulsation.

DP flow meter
According to the ISO TR 3313, the threshold value for a DP flow meter can be defined in terms of *velocity* pulsations such that the flow can be treated as steady if:

$$(U'_{RMS}/\bar{U}) \leq 0.05 \qquad (4\text{-}1)$$

In which
U'_{RMS} Root Mean Square value of the *velocity* pulsations
$\bar{U}$ flow time-mean value

Turbine flow meter
The threshold value for a turbine flow meter can be defined in terms of *velocity* pulsations such that the flow can be treated as steady if:

$$(U'_{RMS}/\bar{U}) \leq 0.035 \qquad (4\text{-}2)$$

In which
U'_{RMS} Root Mean Square value of the *velocity* pulsations
$\bar{U}$ flow time-mean value

Vortex meters
According to section 6.3.1.2. of the ISO TR 3313, the flow can be assumed steady state if:

$$(U'_{RMS}/\bar{U}) \leq 0.03 \qquad (4\text{-}3)$$

If the steady state requirement is fulfilled it can be assumed that *velocity* pulsations will not cause a flow meter error.
If the steady state requirement according to equation (4-3) cannot be fulfilled it shall be assumed that the flow meter is liable to locking-in (see section 8.2 of these guidelines) if the frequency of the *velocity* pulsations is in the range 0.25-2 times the maximum vortex frequency. In that cause a flow meter error will occur.

Coriolis and Ultrasonic meters
For these meters a threshold value is not defined in any standard which makes it impossible to do a first assessment of the risks. Furthermore, it is also explained in section 4.2 that it is not possible to calculate the flow meter error for these meters. However, it is possible to avoid a large flow meter error by the methods as described in section 9.3 for the Coriolis meter and in section 10.3 for the ultrasonic meter.

4.5 Summary allowable flow meter errors

4.5.1 Introduction

The allowable flow meter error shall be specified by the purchaser of the compressor system. The standards described above give allowable metering errors and allowable *velocity* pulsations for several types of flow meters. A summary is given in Table 4.1. and can be used by the responsible company.

The lack of absolute threshold limits and the fact that the error cannot be calculated for all types of meter, implies that often only a *relative* reduction of the flow meter error can be achieved, by lowering the pulsation amplitudes. Methods to reduce the flow meter error for these meters are given in the applicable sections for these meters.

Table 4.1 Summary of the allowable flow meter errors and allowable *velocity* pulsations.

Type of flow meter	Flow measurement error (%)	Velocity pulsations** $\frac{U'_{RMS}}{\bar{U}}$ [−]
DP	Non-custody transfer: 1%* Custody transfer: 0.125%*	0.05
Turbine		0.035
Vortex		0.03
Coriolis		-
Ultrasonic		-

*Values according to the API Standard 618, 5th edition. The standard does not discriminate between different types of flow meters.
**Values according to the ISO TR 3313

4.5.2 Flow meter errors for existing systems or during acceptance tests

Often, during commissioning a performance test is done, which evaluates (amongst others) the compressor flow. As discussed before, the effect of pulsations may lead to significant metering errors. Moreover, the errors are often systematic (constant negative or positive offset in the reading).

In case the flow meter is used for existing systems or for acceptance tests it is pointed out that the measured flow may deviate from the actual flow caused by flow meter errors. This is the case if the flow meter is installed at a location where pulsations are present. For those field cases it is difficult to quantify the flow meter error which is due to the fact that the flow meter error of several of the flow meter types discussed in these guidelines, is dependent on the *velocity* pulsations.
From Table 4.1 it is shown that most allowable levels are based on the ratio of velocity pulsations and mean static flow. In contrast to *pressure* pulsations, *velocity* pulsations cannot be measured directly very easily.

All techniques to measure *velocity* pulsations require specialised equipment and knowledge and shall be carried out for that reason by a specialist. There are several methods to measure the *velocity* pulsations. One of them is the two-microphone method. With this method the *pressure* pulsations are measured upstream and downstream of the flow meter and the *velocity* pulsations can be calculated accordingly. More detailed information about the two-microphone method can be found in several publications [27, 28, 29]. *Velocity* pulsations can also be measured with other special techniques e.g. with non-intrusive techniques like laser Doppler anemometry, Doppler shift and electromagnetic flow meters. An insertion device such as thermal anemometry can also be used. However, all methods to measure the *velocity* pulsations are very difficult and not always feasible to be used in the field. For that reason, they are limited to laboratory applications.

In general, it means that the most optimum way to determine the flow meter error, both during the design stage and field acceptance tests, is to calculate the *velocity* pulsations with a detailed pulsation analysis as specified in the API Standard 618. The calculated values can be compared with the allowable values as given in Table 4.1.

In section 5.5 some guidance is given if the actual flow needs to be measured during commissioning of a compressor and which actions are required in case the measured flow deviates from the guaranteed flow.

5 Best-practice for handling issues with flow meter accuracy in reciprocating compressor systems

5.1 Introduction

Flow meters are essential components in process installations, for the purpose of control and custody transfer. In general, flow meters are robust process equipment. However, it is known that the accuracy of the flow meters can suffer from disturbances associated with the process (flow disturbance, pulsations, vibrations, turbulence etcetera). Moreover, the detection of a potential issue with flowmeter accuracy is very difficult, because these issues are normally not associated with ‘common’ observations such as high vibrations and failures of equipment. The primary parameter that determines the metering error (velocity pulsation amplitude) is not easily measured in real installations.

Reciprocating compressors are inherent sources of unsteady flow, leading to pulsations and vibrations. To ensure a safe and reliable system layout, the API 618 standard stipulates in detail how pulsation analyses shall be executed for new designs. However, the philosophy of API 618 is mainly dedicated to *structural integrity* and less so to the accuracy of flow meters. Moreover, the guidelines formulated in the API 618 standard for the maximum allowable errors of flow meters are very strict. Also, the simplified guidelines in ISO TR 3313 [2] for flow pulsation amplitudes are very strict and may be difficult to achieve with practical and economical means. If not explicitly specified, these very strict guidelines may be used as ‘default’, without a clear definition of the responsibilities for the various partners. Often, during the selection of flow meters (in the early design), the effect of flow meter disturbances is not fully taken into account. This is caused by the fact that not all relevant parties are included in this evaluation and selection. In case a potential issue with flow meter accuracy is identified late in the detailed design, often complex discussions arise on optimal system design and on responsibilities for the implementation. Even though various standards are available on this topic, there is a strong need for extra, practical, guidance.

5.2 Early system design and selection of flow meter

As a first step, during the bidding phase, it shall be clearly agreed which specification shall apply to the flowmeter accuracy, including the individual responsibilities of the partners in the project.

The impact of process disturbances on flow meter accuracy shall be assessed in the early stage of the design of the system. Ideally this shall be a *joint effort* between the end user, the compressor vendor/manufacturer, the designer of the pipe system, the pulsation consultant and the flowmeter manufacturer.

During initial design of the system e.g. in the FEED study, the risks for disturbances on flow meters shall be identified and the most suitable type of flow meter shall be chosen, the one which is least sensitive to errors caused by the various disturbing mechanisms. This evaluation can be done based on the results of this EFRC guideline. If unacceptable flow measuring errors are expected, consultation of the expertise of the flow meter manufacturer shall be considered. This may result in a dedicated, state-of-the-art design for critical cases.

To make a preliminary judgement of significant disturbing effects on the different types of flow meters, Table 5.1 may be used. Note that the tentative ranking is made, considering the physical measurement principles of each flow meter type.

Table 5.1 Preliminary ranking of disturbance affecting the primary measurement principle

Flow meter	**Flow disturbance***	**Low frequency pulsations**	**External Vibrations**	**High frequency broadband noise**
DP meter (orifice, nozzle venturi)				
Turbine				
Vortex				
Coriolis				
Ultrasonic				

General note: there may be considerable differences in observations between different brands and instrument types that have been found in the literature on this topic.
* Provided that sufficient inlet length is used, in accordance with industry guidelines.

5.3 Pulsation analysis

During detailed design, a pulsation analysis shall be done to provide detailed information on velocity pulsation levels within the system. The pulsation study shall be done in accordance with accepted industrial guidelines such as the API 618 standard. It is important to include all operating scenarios as input for the analysis. For example, process conditions, part load conditions or minimum operating speed of the compressor may be the most critical cases for flow meter accuracy (low mean flow). The pulsation consultant shall advise on the optimal system layout and the optimal location for the flow meter and shall make a preliminary judgment on the risk on large flow metering errors. The results of the pulsation analysis (residual pulsation amplitudes and frequency near the flow meter) shall be shared with the flow meter vendor for approval/confirmation of the specified meter accuracy. If the situation is judged as unsatisfactory, additional modifications can be evaluated after joint discussion such as re-location of the flow meter, extra damping, extra volume, re-design of pulsation dampers, optimal flow meter type etc. Detailed discussion for the different types of flow meters can be found in chapters 6 to10.

5.4 Mechanical response analysis

In the detailed mechanical response analysis, the vibration levels on the piping near the flow meter shall be evaluated. In case the flow meter is susceptible to piping vibrations, the support layout in the vicinity of the flow meter shall be carefully evaluated. The results of the mechanical response analysis (vibration levels, frequency) shall be shared with the flow meter vendor for approval/confirmation of the specified meter accuracy and the ability of e.g. the electronics to withstand the vibrations.

5.5 Commissioning

If, during commissioning, the flow meters are used for performance testing of the compressor, the reading of the flow meters may be disturbed by flow pulsations and/or vibration effects. Moreover, often, the error is systematic (always too high or always too low). In such cases, it is recommended to do the performance test with at least a duplicate measurement. Preferably, an alternative meter type shall be used, to identify and eliminate potential errors. During the performance test, a pulsation and vibration measurement at relevant locations (minimum at the meter location) shall be done. This enables a calibration of the pulsation and vibration simulation models to the actual conditions during performance testing. The calibrated simulation models can be used to quantify the velocity pulsations near the meter and determine the flow metering error for that type of meter where the relation between velocity pulsations and flow meter is known.

6 Differential Pressure (DP) flow meters

6.1 Working principle

The orifice plate is one method in a group known as head loss devices or differential pressure (DP) flow meters. In simple terms, the pipe line fluid passes through a restriction and the pressure differential is measured across that restriction as shown in Figure 6.1. The flange connection including the pressure tapping's is named the primary element and the system from the pressure tapping's up to the DP meter is named the secondary element.

Other flow meters in the differential group include nozzle and venturi nozzles as shown in Figure 6.2. These meters can be used for certain applications to minimize undesired effects such as excessive pressure loss or abrasion of components due to solids carried in the fluid. Nozzles with a rounded, funnel-like inlet clearly reduce turbulence and thus create less pressure loss. Turbulence reduction is even greater with Venturi nozzles and finally with Venturi tubes where the restriction is created by longer, conical constrictions in the pipe wall.
All these designs can be supplied with different restriction diameter sizes so that the pressure loss and the differential pressure generated can be optimized for the process conditions.

The relationship between the pressure loss by the DP flow meter and the flow of the fluid passing through the pressure differential flow meters is given by equation (6-1) as follows:

$$\Delta P = K\, {q_v}^2 \qquad (6\text{-}1)$$

In which:

- ΔP differential pressure across the orifice plate
- K proportionality factor (pressure loss coefficient)
- q_v volume flow rate

The flow can be calculated by the square root of the pressure loss as follows:

$$q_v = \sqrt{\Delta P / K} \qquad (6\text{-}2)$$

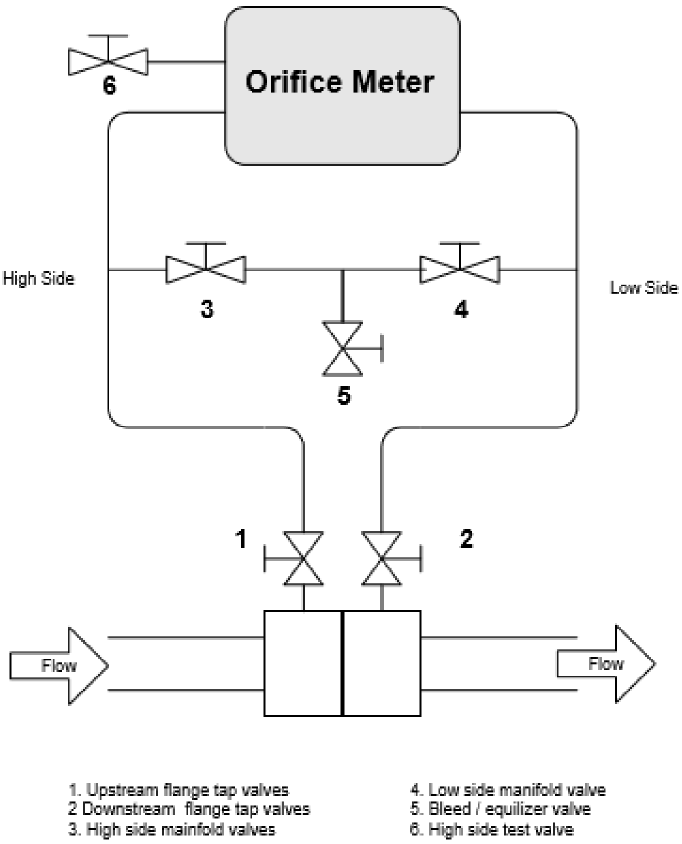


Figure 6.1 Principle of an orifice plate flow meter (source: Coastal Flow)

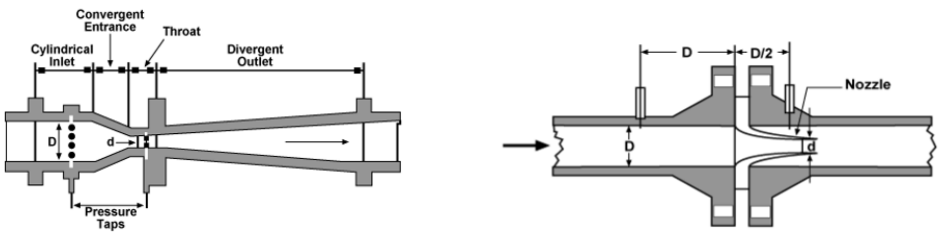


Figure 6.2 Principle of venturi nozzle (left) and flow nozzle (right) flow meters

6.2 Flow meter errors

6.2.1 Primary element errors

Square root error

According to equation (6-2), the square root of the (instantaneous) pressure difference shall be averaged over time ($\overline{\Delta P^{1/2}}$) for a proper flow rate determination. However, slow response DP flow meters are not capable to measure the *velocity* pulsations and the most fundamental error at an orifice plate is the square root error. Instead, the equipment will measure the time-mean pressure difference ($\overline{\Delta P}$) before the square root is taken and means that the *measured* pressure difference ($\sqrt{\overline{\Delta P}}$) is not equal the *actual* pressure difference

($\overline{\Delta P^{1/2}}$). The difference is causing the square root error and can be calculated by equation (6-3). The square root error is inherent for this type of flow meter due to the data processing and is always positive and increases with increasing pulsation amplitude.

$$E_{sqrt} = \frac{\sqrt{\overline{\Delta P}} - \overline{\Delta P^{1/2}}}{\overline{\Delta P^{1/2}}} \tag{6-3}$$

In which:

E_{sqrt}	square root error of differential pressure transmitter reading
$\sqrt{\overline{\Delta P}}$	square root of the time-mean pressure difference
$\overline{\Delta P^{1/2}}$	time-mean value of the square root of the instantaneous pressure difference

Figure 6.3 shows the square law relation between the volumetric flow (q_v) and the differential pressure (ΔP) across the orifice plate. It can be seen that the square law creates the distortion since a larger portion of the time-mean pressure differential ($\overline{\Delta P}$) wave occurs above the time-mean flow $\overline{q_v}$.

It is noted that the square root error (SQRT) is always positive which means that the meter will indicate a too high flow.

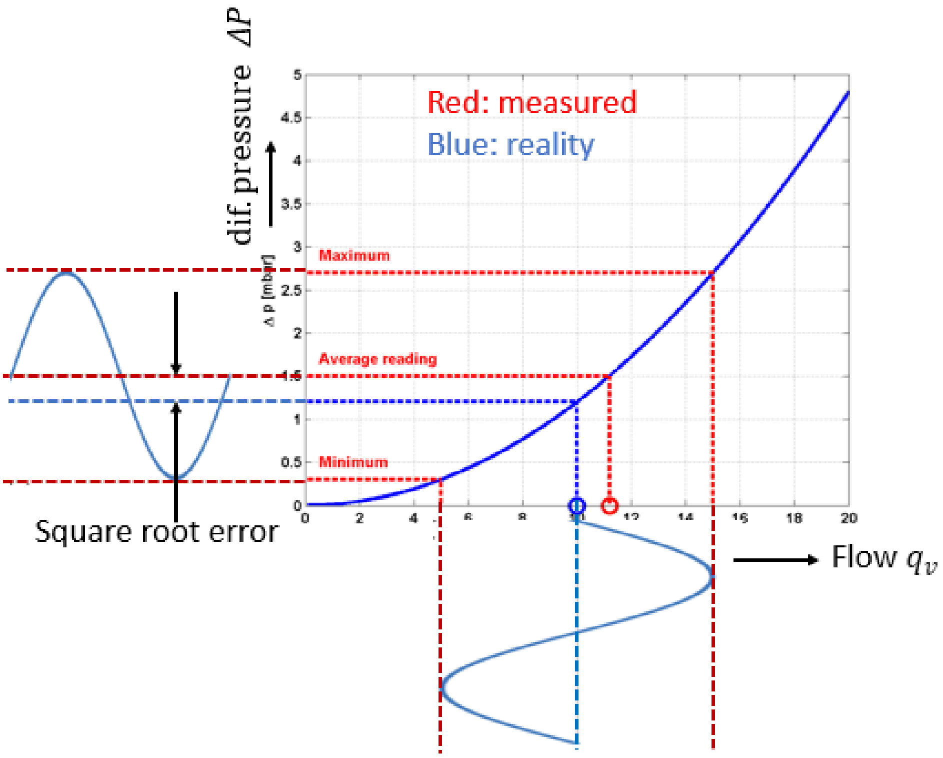


Figure 6.3 Square root error resulting from pulsation effect on flow meters.

Inertial effects
Equation (6-3) is developed for a steady state flow. Developing an orifice equation from the time dependent unsteady flow considerations, including the unsteady continuity and momentum equations, results in an additional inertial term which is proportional with the acceleration of the flow. This proportionality factor depends on certain dynamic properties of the orifice and is also time varying.
As a result of this inertia, the flow velocity changes lag behind differential pressure changes and the simple relationship between the measured pressure difference and flow is distorted.

Inertial effects distort the differential pressure but do not increase the average pressure difference. One feature of the inertial effect is that when the measured pressure difference is averaged over time with a slow response sensor, the inertial effect error is eliminated and only the square root error will remain. However, if the square root of the instantaneous differential pressure across the orifice is recorded correctly with a fast response sensor only the inertial effect error is left.
The inertial effects are in general small with respect to the square root error unless the amplitude and frequency become relatively high. In the paper of Robert J. McKee [14] it is indicated that inertial effects become important if the peak-to-peak *velocity* pulsation is larger than approximately 80% for frequencies above 100 Hz. It is noted that the error caused by the inertial effect is always negative. In general, the dominant frequencies of pulsations for reciprocating compressor systems are below 100 Hz which means that the inertial effects can be neglected. In case the dominant frequency is above 100 Hz, it is recommended to the consult the ISO TR 3313.

It is noted that the inertial error is always negative which means that the meter will indicate a too low flow.

Orifice pressure loss coefficient shift
In a static flow calculation, the orifice coefficient K from equation (6-1) is assumed to be constant. However, swirl and flow distortions caused by pulsations are known to cause a change in orifice coefficient K [31].
However, for orifice meters with flange taps the change in the orifice coefficient only causes an error at extreme pulsation amplitudes which will not happen if all pulsation levels do not exceed the allowable levels as stipulated in the API 618.
The orifice coefficient for other tap locations, e.g. pipe taps, is affected much more by pulsations. Pipe taps for a DP meter are therefore not recommended in pulsating conditions.

It is noted that the orifice pressure loss coefficient shift is always positive which means that the meter will indicate a too high flow.

Calculation of total error
If a pulsation analysis is carried out according to the API Standard 618, the *velocity* pulsations can be calculated on each location of the system. The estimated total error can be calculated according ISO TR 3313 with equation (6-4) below. It is noted that the total error does not contain the inertial term as discussed before and in section 6.1.2.3. of the ISO TR 3313. An explanation and calculation of U'_{RMS} can be found in chapter 1.

$$E_T = \sqrt{\left[1 + \left(\frac{U'_{RMS}}{\bar{U}}\right)^2\right]} \text{ -1} \tag{6-4}$$

In which:

E_T	Total error [-]
$\bar{U}$	Average gas velocity based on the average actual volumetric flow rate [*m/s*],
U'_{RMS}	Root Mean Square value of *velocity* pulsations [*m/s*],

Time domain versus frequency transfer method for pulsation calculation
Most programs which are used nowadays for the calculation of pulsations are based on time domain calculations. For those programs the root mean square value of the *velocity* and *pressure* pulsations can be calculated from the time signal. Other programs are based on the frequency transfer method and the velocity and pressure pulsations are then only available for discrete frequency components which means that the root mean square value cannot be calculated accurately. However, the largest flow meter errors will generally occur for acoustic resonance conditions which means that in general the dominant pulsations will occur at one frequency.
If the pulsations contain one or more discrete frequencies, the root mean square value can be calculated with equation (1-2) and (1-3) of chapter 1.

It is noted that the total error is always positive which means that the meter will indicate a too high flow.

Important remarks to calculate the maximum square root error
Pressure and velocity pulsations strongly depend on several parameters such as the load condition of the compressor (100%, 50% etc.), type of gas, density, pressures, speed of sound, acoustic impedance, etc. It is possible that large *velocity* pulsations leading to large flow meter errors can occur even in the case that the *pressure* pulsations are far below the allowable levels according to the API Standard 618.

To explain this effect, it is needed to discuss the acoustic impedance, which is in fact the acoustic response of the pipe system for all possible frequencies. Acoustic impedance has the symbol Z, and is defined as the ratio of pressure pulsation P'_p to flow pulsations U'_G according to equation (6-5) as follows:

$$P'_p = Z\, U'_G \tag{6-5}$$

In which

P'_p	*pressure* pulsation level [%]
Z	acoustic impedance [kg/m^3]
U'_G	*velocity* pulsations [%]

A low acoustic impedance means that a small *pressure* pulsation causes a large *velocity* pulsation, leading to a high flow meter error. In the theoretical case of an infinitely long, undisturbed pipe, the relation between pressure and velocity pulsations can be described by an analytic relation, using the characteristic impedance of the gas (depending only on the process conditions). However, in real systems, reflections upon obstacles (diameter transition, side branches etcetera) will cause acoustic resonances. In such case, the impedance cannot be described with simplified analytical relations, and numerical models are required. In such cases, the relation between *pressure* and *velocity* pulsations depends strongly on the location within the pipe system. This means that for a given pipe system, locations for the flow meter may be optimized, making it less susceptible for metering errors.

Furthermore, it is known that large *velocity* pulsations can occur for unloaded conditions of the compressor due to the fact that the acoustic damping of the pipe e.g. by means of an orifice plate is lower due to the reduced flow. Two effects play a role; first the lower mean flow (increasing the *relative* effect of the flow pulsations) and second the lower acoustic damping which will increase the amplitudes upon resonance conditions. It is therefore important that the maximum flow meter error will be calculated for all possible operating and process conditions with a detailed pulsation analysis according to the API Standard 618.

Several examples on the above items are given in Appendix B.

6.2.2 Secondary element errors

Velocity pulsation is associated with *pressure* oscillation at the orifice plate taps. The actual pressure signals are usually distorted by the length of the tube connecting the pressure tapping's to the manometer, named gauge line, and the internal volume of the differential pressure (DP) transmitter, resulting in inaccurate flow rate metering. Furthermore, the gauge line fittings and the valve manifold as shown in Figure 6.1, contain abrupt changes in the internal diameter altering the mean differential pressure. This secondary system can amplify or attenuate the actual oscillating pressure signal measured across the orifice as shown in Figure 6.4 which shows a typical frequency response function (FRF) of a gauge line with an acoustic resonance.

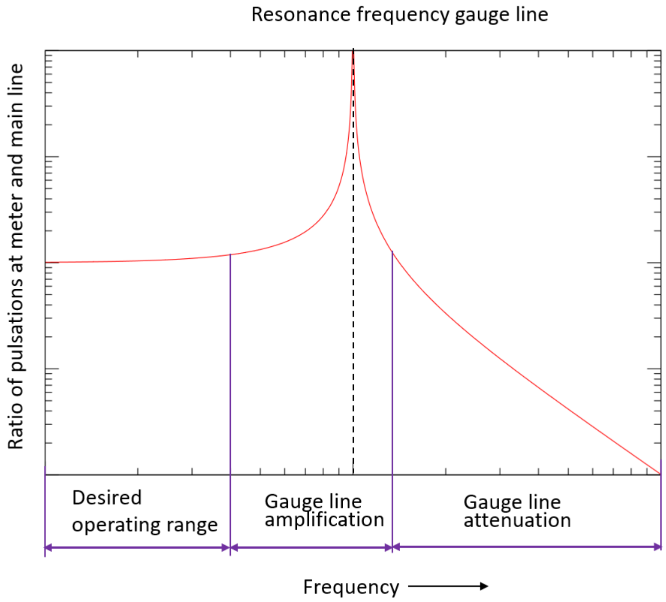


Figure 6.4 Typical Frequency Response Function (FRF) of a gauge line

Gauge line amplification
Gauge line amplification occurs when the frequency of the pulsations in the main line coincides or is close to the standing wave resonance in the gauge line. When amplification occurs, a too high reading of the pressure signal will occur. In particular when the two gauge lines (upstream and downstream of the orifice) are of different lengths, deviations to the correct pressure reading are expected.
Gauge line amplification is usually a result of a quarter lambda ($\frac{1}{4}\lambda$) standing wave type occurring in this line as shown in Figure 6.5. The quarter lambda standing wave resonance can be calculated with equation (6-6).

$$\lambda = \frac{c}{4f} \tag{6-6}$$

In which:
- λ pulsation wave length [m]
- c speed of sound [m/s]
- f frequency [Hz]

Depending on the length of the gauge line, the frequency of the pulsations and the speed of sound, resonances at 3/4 lambda, 5/4 lambda, etc. can also occur especially for compressors with a higher speed and for gases with a low speed of sound e.g. carbon dioxide (CO_2). It is desirable that the pressure pulsation at the meter and the pressure tap is the same (ratio of 1). This is in general the case for frequencies which are minimum -20% lower than the acoustic resonance condition, as shown in Figure 6.4.

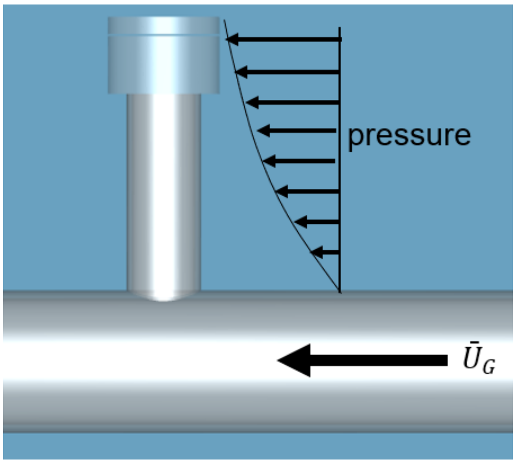


Figure 6.5 Pulsations in the main and gauge line at a ¼ lambda standing wave

Gauge line attenuation
Gauge line attenuation occurs when the frequency of the measured pulsation is higher than approximately +20% of the frequency of the standing wave as shown in Figure 6.4. Also, when attenuation occurs in the gauge lines, deviations in the measured pressure differential may occur, in particular when the gauge lines are of different length or layout.

It is therefore recommended to keep the gauge line as short as possible so that the lowest standing wave resonance is far enough from the frequency of the dominant pulsations in the main line. It is recommended that the resonance frequency is minimum 2 times the frequency of the dominant pulsations in the main line. Furthermore, it is recommended to install constant-diameter tubing, fittings and valve manifolds, together with a DP transmitter having a small chamber and high-frequency response.

It is noted that the secondary element error can be positive or negative which means that the meter will indicate a too high (if the pulsation frequency is below the gauge line standing wave frequency) or a too low flow (if the pulsation frequency is above the gauge standing wave frequency).

6.3 Guidelines how to reduce flow meter errors

6.3.1 Introduction

One of the disadvantages of a slow response pressure differential meter is that it has an intrinsic feature of a positive flow meter error when it is used in a pulsation flow. This flow meter error shall not exceed the allowable limits as defined by the purchaser of the system.
There are several methods how to reduce the flow meter error such as damping of *velocity* pulsations, relocating the meter to a location with the lowest velocity pulsations, acoustic separation from the excitation source, shifting of the frequency of pulsations, etc. Achieving the largest reduction strongly depends on the system layout, on the (varying) process and the compressor load conditions during the life time of the system. It is shown from experience that in general the most optimum reduction methods are achieved by a combination of several methods as given in these guidelines. The guidelines as given in this section can be used during the preliminary and detailed design of the system (piping layout, acoustic dampers/filters, etc.) but also during a root cause analysis if there is a suspicion that the flow is not measured correctly.

An optimisation to achieve acceptable flow meter errors, considering investment and operation costs, can only be achieved with a detailed pulsation analysis according to the API Standards 618. The results of the analysis shall always be discussed with the compressor and flow meter vendor, the engineering contractor and purchaser of the system to achieve the most efficient solution with an acceptable flow meter error. If for example a too strict allowable error is specified, the consequence is that more stringent reduction measures are requested which are not always necessary, feasible or economically achievable. In many cases a compromise is required between acceptable flow meter errors and total costs.

Appendix D.1 gives a flow chart with an overview of flow meter errors, the calculation methods and how to reduce them.

6.3.2 Installation of flow conditioners and flow straighteners

It is well known that flow disturbances such as pulsations, disturbed flow and flow noise, also named installation effects, may affect the performance of all flow meters and will add consequently an additional flow meter error. This effect is difficult to quantify and for that reason these disturbances shall be avoided as much as possible. This effect can be minimised by installing the flow meter far enough from any disturbances such as bends, tee joints, valves with a large pressure loss, reducers, inlet and outlet of volumes (pulsation dampers, separators, etc.).

The primary device of the flow meter shall be installed in the pipeline at a position such that the flow conditions immediately upstream of the primary device approximate to those of swirl-free, fully developed pipe flow. Conditions meeting this requirement are specified in section 7.3. of the ISO 5167-1 [3].

The required minimum upstream and downstream straight lengths required for differential flow meters are specified in Clause 6 of ISO 5167-2 [4].
However, a flow conditioner as described in 7.4 of ISO 5167-1 [3] will permit the use of much shorter upstream pipe lengths. Such a flow conditioner shall be installed upstream of the primary device where sufficient straight length, to achieve the desired level of uncertainty, is not available.

6.3.3 Additional damping

Acoustic resonances such as standing waves and Helmholtz type resonances, will amplify the pressure and velocity pulsations in the system leading to unacceptable flow meter errors. These acoustic resonances can be mitigated e.g. by the installation of additional damping devices such as an orifice plate.
Damping is most effective if the device is installed at a location with the highest *velocity* pulsations because the pressure loss is proportional with the square of the velocity. In general, the locations with the highest *velocity* pulsations are at the inlet and outlet of volumes (velocity anti-node) such as pulsation dampers, separators, etc. See also Figure 6.1 and the explanation of this in Appendix A. It is noted that additional damping means an increase of energy consumption and the power of the compressor driver must be large enough. If installed at the suction side, the compressor capacity will be (slightly) reduced by installation of extra pressure loss due to orifices.
It shall be kept in mind that the pressure drop of the damping device will reduce when the flow is reduced and goes with the square of the flow ratio, e.g. a factor 4 when the flow is reduced with 50%. This means that the pressure and velocity pulsations will increase, leading to a higher flow meter error. The damping device(s) shall be optimised for that reason for all possible flow and process conditions and for all different gases, e.g. Nitrogen duty for start-up gas purposes of Hydrogen systems.
It may also be possible to install several orifice plates at different locations to achieve an optimal effect at minimum pressure loss. This is especially the case for a combination of more than one resonance.

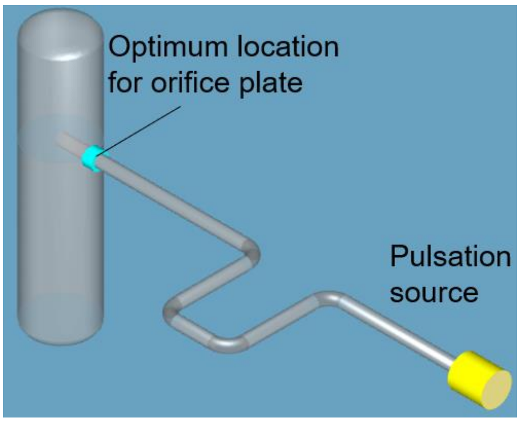


Figure 6.6 Optimum location (velocity anti-node) for an orifice plate.

6.3.4 Detuning acoustic resonances by changing pipe lengths

A change in pipe length can be very effective in detuning/shifting the frequency of a standing wave or Helmholtz type resonance so that it does not coincide anymore with the frequency of the dominant pulsations. Some do's don't' s and engineering rules on this are given in section of A.5 of Appendix A.

6.3.5 Optimum (re)location of the flow meter

One of the most efficient methods in reducing the flow meter error is the (re)location of the flow meter to a location with the lowest *velocity* pulsations of which several examples are shown in the next sections.

It is noted that the meter shall not be (re)located in close vicinity of locations with large flow disturbances such as bends, tee joints, valves with a large pressure loss, reducers, etc. The primary device of the flow meter shall be installed in the pipeline at a position such that the flow conditions immediately upstream of the primary device approximate those of swirl-free, fully developed pipe flow. Conditions meeting this requirement are specified in section 7.3. of the ISO 5167-1 [3]. The required minimum upstream and downstream straight lengths required for differential flow meters are specified in Clause 6 of ISO 5167-2 [4]. However, a flow conditioner as described in 7.4 of ISO 5167-1 [3[] will permit the use of much shorter upstream pipe lengths. Such a flow conditioner shall be installed upstream of the primary device where sufficient straight length, to achieve the desired level of uncertainty, is not available.

6.3.5.1 Installation at a location with little or no velocity pulsations

From equation (6-4) in section 6.2.1 it is shown that the flow meter error is maximal at a location where the *velocity* pulsations are maximal. For that reason, the most effective measure in achieving the lowest flow meter error is to install the flow meter at a location where the velocity pulsation is minimum which is at a *velocity* node, of which an example is shown in Figure 6.7 for a standing wave type resonance. More examples are shown in Appendix A.

It is noted that in many cases, the location of the *velocity* node is at *pressure* anti-node (node with highest levels). This depends on the specific features of the acoustic response (resonances) in the pipe system. More explanation on nodes and anti-nodes is given in section A.3 and A.4 of Appendix A.

The wave length and also the location of the *velocity* node of a standing wave type resonance depends on the speed of sound (c) and frequency (f) of the pulsations which is calculated with the following equation: $\lambda = c/f$. An example of the location of a *velocity* anti-node is the middle of two (large) enough volumes for a ½ lambda standing wave resonance between these volumes.

For a Helmholtz resonator, the *velocity* pulsations are constant (see also section A.4) between the two volumes which means that a relocation of the meter will not help and other methods shall be investigated in that case.

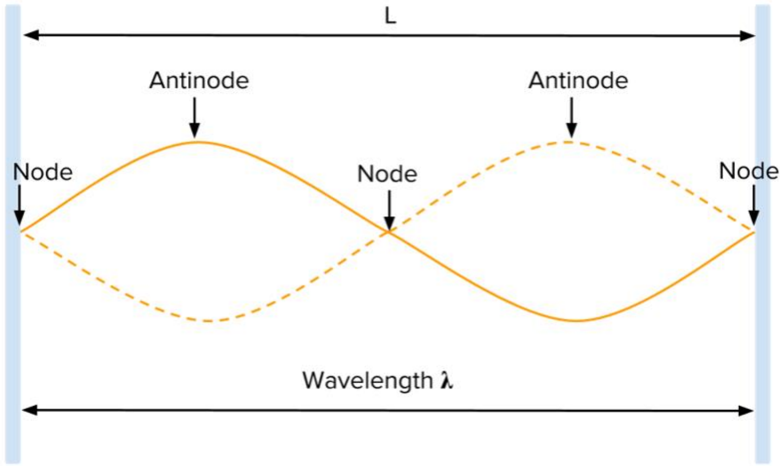


Figure 6.7 Example of nodes and anti-node of a *velocity* pulsation for a fixed wall boundary condition on both sides

It is noted that the wave length may change for other compressor loads (e.g. 100% versus 50% load), other process conditions (pressure and temperature) and type of gas (e.g. CO_2 versus Hydrogen). Especially, for high speed compressors and/or a gas with a low speed of sound (e.g. CO_2) it is sometimes challenging to find an optimum location due to the short wavelength. A small change in location can already lead to much higher velocity pulsations in those cases.

Furthermore, the velocity (and pressure) node location will change with varying compressor load and process conditions and makes it sometimes very difficult to find an optimum location for all possible conditions. This is even more challenging and, in many cases, even not possible for compressors with a variable speed. For those cases it means in general that a combination of other reduction methods are required to achieve acceptable flow meter errors.

As explained before, the optimum location can only be determined with a detailed pulsation analysis.

6.3.5.2 Installation of flow meter far enough from the compressor

One of the possibilities to reduce the *flow* pulsations is to (re)locate the flow meter by a significant distance from the compressor, the reduction caused by the fact that longer pipe systems have a larger damping. This is not always effective because pulsations can travel a long distance before they are damped out sufficiently. This is especially the case for a gas with a high speed of sound which has a rather long wavelength e.g. Hydrogen. A general rule is that pulsations are damped out sufficiently for a length of approximately 5-10 times the wave length between the flow meter and compressor. Sometimes, this is a very challenging requirement. For example, for a Hydrogen compressor with a speed of 300 rpm and a speed of sound of 1200 m/s, it means a distance of 1200 meter for 5 times the wavelength!

6.3.5.3 Acoustic separation

A good solution to achieve an acceptable flow meter error is to separate the flow meter from the reciprocating compressor which can be achieved by installing/relocating the flow meter upstream (suction side) or downstream (discharge side) of a large enough volume as shown in the example in Figure 6.8 The minimum required volume to achieve this acoustic separation strongly depends on the type of gas. This is caused by the fact that the acoustic effectiveness is inversely proportional to the square of the speed of sound (effectiveness ~ V/c^2) For a gas with a high speed of sound (e.g. Hydrogen), a much larger volume is required than for a gas with a low speed of sound (e.g. CO_2). A general rule for the minimum required gas volume of the separator is 5-10 times the volume of the pulsation damper, to ensure full separation.

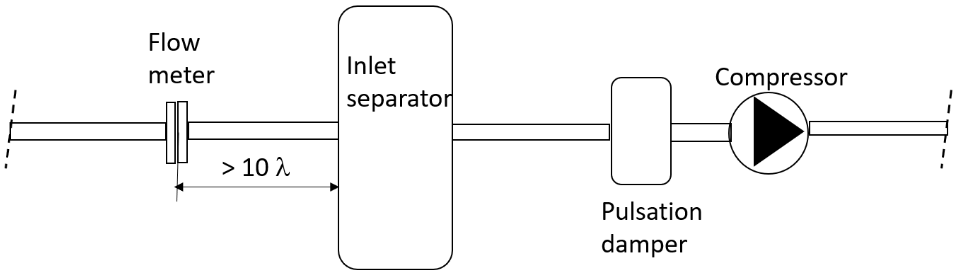


Figure 6.8 Example of an acoustic separation between the pulsation source and flow meter at the suction side of a reciprocating compressor

An acoustic separation can also be achieved by large a large diameter change, e.g. large headers. A general rule is that the header diameter shall be minimum 4 times the diameter of the connected pipe. However, as explained before this is strongly depending on the type of gas and process and compressor load conditions. Also, for this case, the header diameter shall in general be larger for a gas with a high speed of sound, e.g. Hydrogen.

6.3.6 Changing pulsation damper volumes

Pulsations dampers are installed at the suction and discharge side of each stage of a reciprocating compressor to mitigate the pulsations which are transferred to the piping. The design of a pulsation damper volume is based on acceptable pulsations as stipulated in e.g. the API Standard 618 to avoid too high vibrations of the damper and pipe system.

Despite the fact that a pulsation damper volume may be sufficiently large to mitigate the pulsations according to the allowable levels as stipulated in the API 618, a large increase may be necessary to achieve acceptable flow meter errors. Larger volumes mean in general larger production costs especially for high pressure applications and in most of the cases there are more effective reduction methods at lower costs. Furthermore, an increase in damper volume will also mean an increase in mass which can lead to unacceptable compressor cylinder loads and higher vibrations.

6.3.7 Acoustic filter

More effective than a larger volume of the pulsation damper as discussed in the former section is an acoustic filter which reduces both the amplitude and frequency of the pulsation if designed well. The filter can consist of two or more volumes separated by baffle plates and choke tubes. In general, this will result in a smaller damper volume with a lower mass compared with an empty bottle as discussed in section 6.3.6. However, the production costs of an acoustic filter are in general much higher than an empty bottle. Designing an acoustic filter for varying process conditions (process, flow, variable speed, etc.) may be very challenging and is not always effective for gases with a high speed of sound, e.g. Hydrogen. Furthermore, damper internals such as baffle plates and choke tubes are not always allowed by the end-user for reasons of corrosion and fatigue failures of these parts. For those cases, a combination of several other reduction methods as discussed in this chapter are more effective.

In the API 688 [6], the API Standard 618 [1] and in ISO TR 3313 [2] some guidelines are given for the design of acoustic filters. However, an optimum acoustic filter for all possible process and flow conditions can only be achieved with a detailed pulsation analysis according to the API 618.

6.3.8 Optimization of the flow meter gauge line

Recommendations according to the ISO TR 331 to reduce the gauge line errors are summarised as follows:

a. The bore of the pressure tapping shall be uniform and shall not be too small, i.e. ≥3 mm.
b. The distance between the pressure tapping's shall be small compared with the shortest wavelength of the dominant pulsations. This is to avoid that different pulsation levels are measured at the pressure tapping's.
c. The tube connecting the pressure tapping's to the DP-transmitter shall have the same bore as the pressure tapping's.
d. The length of the tube connecting the pressure tapping's to the DP-transmitter shall be as short as possible to avoid a standing wave in this connecting line. The resonance frequency of the ¼ lambda standing wave shall be far enough from the frequency of the dominant pulsations at the pressure tapping's, see also Figure 6.4. The lengths of the two gauge lines shall be chosen of equal length.

When the above recommendations cannot be fulfilled, the gauge line might be effectively isolated from the pulsations at the pressure tapping's, by the insertion of identical (linear) damping plugs (orifice plate) into both connection tubes, as close as possible to the pressure tapping's. It is also possible to correct for the pulsations at the DP-transmitter, but this requires an accurate measurement or acoustic calculations of the pulsations at the pressure tapping's and DP-transmitter.

6.3.9 Installation of a meter which is less sensitive for flow meter errors

In many cases the flow meters are already ordered before the pulsation and vibration analysis is carried out. If rigorous system changes are required to achieve acceptable flow meter errors, it is sometimes more cost-efficient to consider another type of flow meter which is less sensitive to one or more of the disturbances for the flow meter of interest.

7 Turbine Flow Meters

7.1 Working principle

A turbine flow meter is used for volumetric total flow and/or flow rate measurement and has a relatively simple working principle. As fluid flows through the turbine meter, it impinges upon turbine blades that are free to rotate about an axis along the centre line of the turbine housing, see Figure 7.1. The angular (rotational) velocity of the turbine rotor is directly proportional to the fluid velocity flowing through the turbine. The resulting output (frequency) is taken by an electric sensor(s) mounted on the flow meter body.

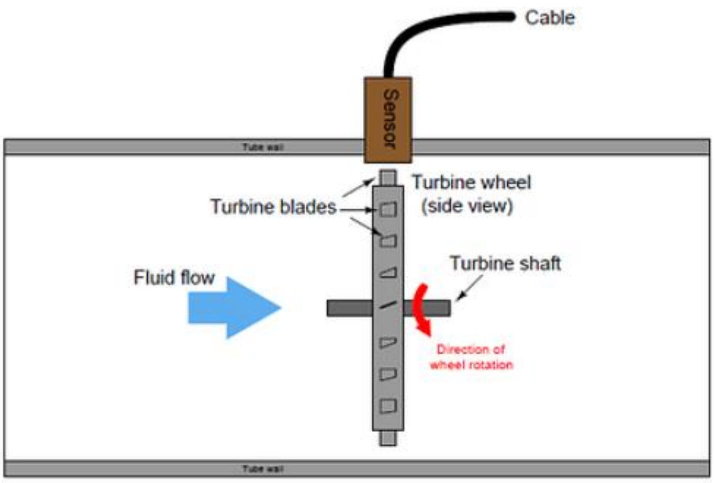


Figure 7.1 Working principle of a turbine meter

7.2 Flow meter errors

According to ISO 9951 [13], turbine flow meters shall not be used where strong velocity and pressure pulsations are present. When subjected to pulsating flow, they will cause a flow meter error caused by the inertia of the rotor and the fluid inertia. The rotor can lag behind the steady state situation in an accelerating flow and can exceed it in a deceleration flow which can even be larger than for an accelerating flow. According to the ISO TR 3313 the systematic flow meter error is positive, and the value depends on the mean flow, *velocity* pulsation amplitude and frequency, and the rotor response parameter.
According to the API 688, a sinusoidal *velocity* pulsation of 5% peak-to-peak results in a maximum error of less than 0.1% which is approximately the allowable level for custody transfer. The criterion of a *velocity* pulsation level of 5% peak-to-peak is therefore an acceptable limit for most cases. For large *velocity* pulsation amplitudes, positive errors up to 25 % or even more can occur.

Much work has been carried out by several researchers (Atkinson [8], Cheesewright [9], Grey [10] and Jepson [11]) which were used in the development of the ISO TR 3313. It is shown that no general prediction for the error in mean flow is possible to define. However, Atkinson has made extensive calculations of the flow meter response for a sinusoidal pulsation and has confirmed his results with measurements. Figure 7.2. shows the relative mean-flow error which is a function of α and β. It is noted that if the pulsations are significantly non-sinusoidal, there are no simple ways for the calculation of the error. However, the pulsations are highest at resonance conditions and in that case, they have in general a sinusoidal pattern.

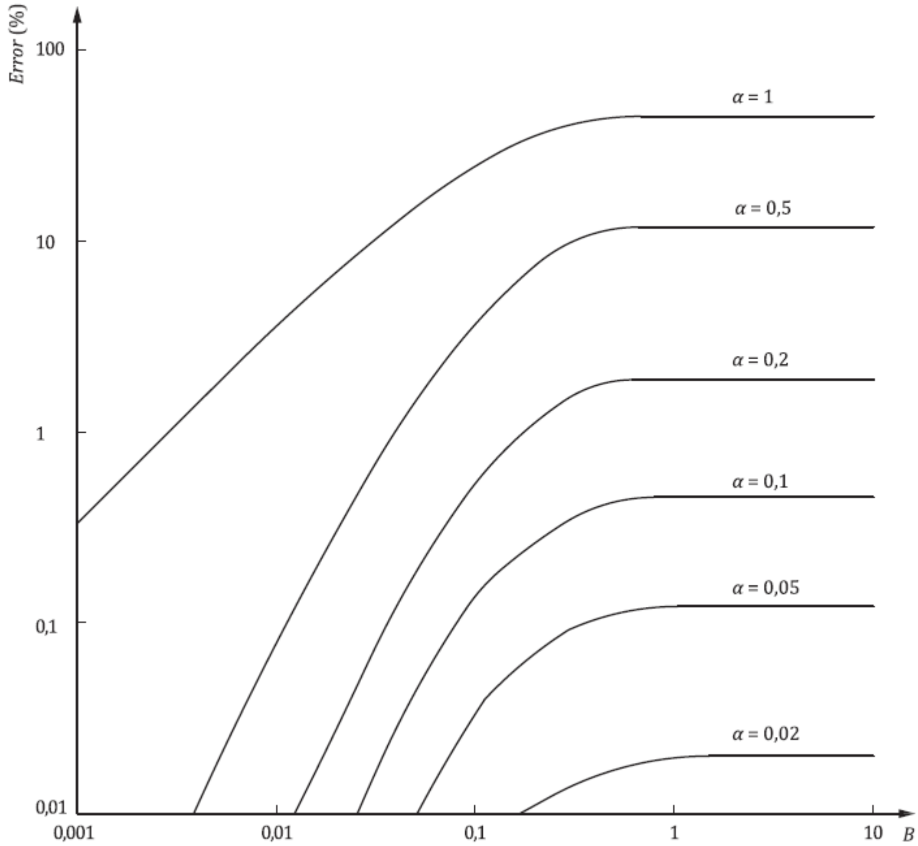


Figure 7.2 Error as a function of α and B (source: Figure C.1 from ISO TR 3313)

The parameters *α* and *B* from Figure 7.2 can be calculated as follows:

$$\alpha = (U'_{max} - U'_{min})/2\bar{U}_G \qquad (7\text{-}1)$$

In which:

α	dimensionless ratio between *velocity* pulsations and mean flow velocity
U'_{max}	maximum value of *velocity* pulsation
U'_{min}	minimum value of *velocity* pulsation
$\bar{U}_G$	mean static flow

The term $(U'_{max} - U'_{min})$ is the peak-to-peak *velocity* pulsation as shown in Figure 1.1 and can be calculated with a detailed pulsation analysis. Term *B* can be calculated as follows:

$$B = bf/\bar{U}_G \qquad (7\text{-}2)$$

In which:

B	dimensionless dynamic rotor response parameter
b	rotor response parameter
f_p	frequency of the flow *velocity* pulsation
$\bar{U}_G$	mean static flow velocity

The frequency of the *velocity* pulsation can be measured in the field or can be calculated with a pulsation analysis during the design stage.

The dynamic rotor response parameter *(b)* shall be given by the meter vendor. The equations how to calculate the response parameter are given in Appendix C.

It is noted that the error is always positive which means that the meter will indicate a too high flow.

7.3 Guidelines how to reduce flow meter errors

7.3.1 Installation of flow conditioners and flow straighteners

For some general information see the first paragraph of section 6.3.2.
The required minimum upstream and downstream straight lengths required for turbine flow meters are specified in Annex E of the ISO 9951 [13].
However, a flow conditioner as described in section 7.4 and Annex C of ISO 5167-1 [3], and Annex E of the ISO 9951 specifically for turbine flow meters, will permit the use of much shorter upstream pipe lengths. Such a flow conditioner shall be installed upstream of the primary device where sufficient straight length, to achieve the desired level of uncertainty, is not available.

7.3.2 General methods to reduce velocity pulsations

The flow meter error is a function of the *velocity* pulsations (see equation (7-1) and (7-2) and the frequency of the pulsations (see equation (7-2) and the rotor parameters (see appendix C). It is obvious that a reduction of the *velocity* pulsations will lead to a reduction of the flow meter error. The reduction methods as described in sections 6.3.3 through 6.3.7 can also be applied for the turbine flow meter.

7.3.3 Changing flow meter parameters

The general methods for reducing the *velocity* pulsations as discussed in the former sections can have a large impact on the installation and operational costs. It is sometimes more efficient to change the turbine flow meter parameters as summarised in Appendix C such as e.g. by decreasing the moment of inertia. However, in general, this means that another turbine flow meter is required and in that case a comparison shall be made of all costs involved. In some cases, it might be cheaper to install another meter instead of changing the pipe system, dampers, etc.

7.3.4 Installation of a meter which is less sensitive for flow meter errors

See section 6.3.9.

8 Vortex Flow Meters

8.1 Working principle

Vortex flow meters operate under the vortex shedding principle, where oscillating vortices occur when a fluid such as water or gas flows past a bluff (as opposed to streamlined) body with a square, rectangular, t-shaped or trapezoidal geometry. The frequency of the vortex shedding depends on the size and shape of the body and the fluid velocity. This metering principle is ideal for applications where low maintenance costs are important. Industrial size vortex meters are custom-built and require appropriate sizing for specific applications. An example is shown in Figure 8.1.

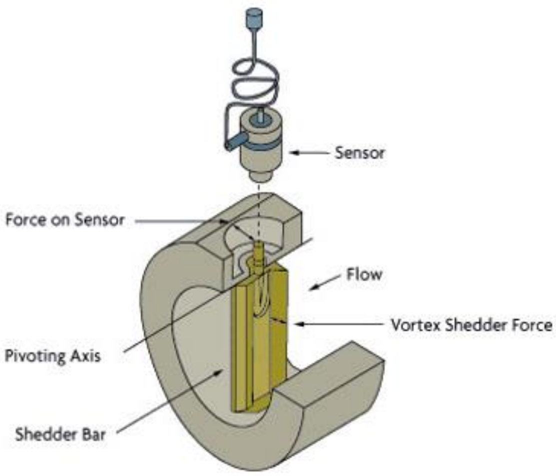


Figure 8.1 Example of a vortex flow meter (source: Omega engineering)

When the medium flows around the bluff body at a certain speed, an alternating vortex street is generated behind the sides of the bluff body, called the "von Kármán vortex", see Figure 8.2.
Since both sides of the vortex generator alternately generate the vortex, the pressure pulsation is generated on both sides, resulting in a fluctuating force perpendicular to the flow direction of which the frequency is proportional to the upstream velocity over a wide range of Reynolds number.

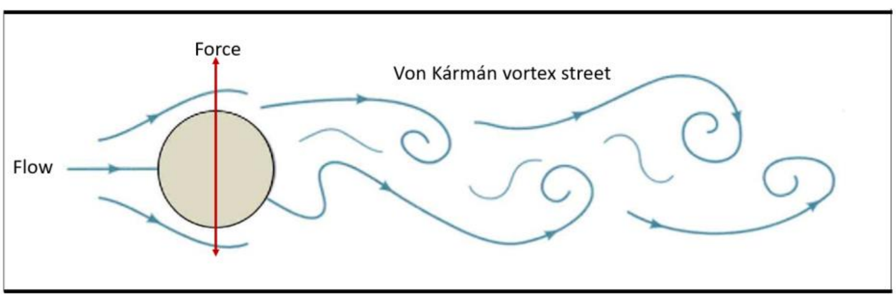


Figure 8.2 Von Kármán vortex street

The majority of vortex meters uses piezoelectric or capacitance-type sensors to detect the pressure oscillation around the bluff body. These detectors respond to the pressure oscillation with a low voltage output signal which has the same frequency as the oscillation. The relationship for the flow velocity under steady flow conditions is given in equation (8-1).

$$\bar{U}_G = \frac{f_v\, d}{S_R} \qquad (8\text{-}1)$$

In which:

- $\bar{U}_G$ mean static flow velocity [m/s]
- f_v vortex shedding frequency [Hz]
- d bluff body diameter, or width in case of a triangular shape [m]
- S_r Strouhal number [-]

The Strouhal number depends on the geometry of the bluff body but typical values are between 0.2-0.3. For a specific type of flow meter, the exact value shall be calibrated accurately in a test bed environment

8.2 Flow meter errors

8.2.1 Errors caused by velocity pulsations

According to the API 688 [6] the vortex signal is influenced by the following two effects in a pulsating flow:

1. The amplitude of the signal which is modified by the *velocity* pulsations.
2. The vortex frequency f_v, which is influenced by the pulsation frequency f_p.

The first effect results in additional frequencies in the vortex signal, corresponding to $f_v - f_p$ and $f_v + f_p$, next to the original vortex frequency f_v. Because of this effect, a number of pulses can be missed, which results in a deviating vortex frequency f_v, indicated by the flow meter, and resulting in a systematic error.
The second effect leads to lock-in which is explained in the next section.

Lock-in effect
Under pulsating flow conditions, the crucial parameter for vortex flow meters is the ratio of the frequency of the pulsations (f_p) to the vortex shedding frequency (f_v). When the ratio f_p/f_v is small which is explained further, there is no disturbance in the flow reading. However, if the pulsation frequency is very close or coincides with the vortex frequency, there will be a strong tendency that the vortex shedding phenomenon will “lock-in” with the pulsation frequency (f_p=f_v). This can happen for ratios of f_p/f_v =0.25, 0.5, 1, 1.5, 2, etc. For those lock-in situations the flow meter will give gross errors in flow readings which can go up to 80%.
When the pulsation frequency is much higher than the vortex-shedding frequency, in general for ratios of f_p/f_v > 2, there will not be an obvious lock-in.

According to the ISO TR 3313, the flow rate range over which the vortex-shedding remains locked-in with the fixed frequency of the pulsations depends on:

- amplitude of the pulsation;
- bluff body shape;
- blockage ratio (bluff body frontal area as a fraction of the pipe-cross area)

Threshold value of pulsation amplitudes and threshold value for lock-in
According to the ISO TR 3313 section 6.3.1.1., the critical *velocity* pulsation amplitude is that amplitude which is just sufficient to cause lock-in for a ratio around f_p/f_v =2. 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-state flow condition. From the literature, which is referenced by the ISO TR 3313, it is concluded that this threshold value of the velocity pulsation is as follows:

$$(U'_{RMS}/\bar{U}) \leq 0.03 \qquad (8\text{-}2)$$

In which:

U'_{RMS} root mean square value of *velocity* pulsation [m/s]
$\bar{U}_G$ mean static flow velocity [m/s]

The ISO TR 3313 further states that unless it is demonstrated that the threshold value is not exceeded, it shall be assumed that there will be a lock-in for f_p/f_v ratios in a range from 0.25 times the vortex shedding frequency and higher.

From measurements carried out by TNO [15] it is shown that, depending on the frequency and amplitudes of the pulsations, error readings of -10% can already occur for a ratio of f_p/f_v =0.25 and can be -50% or even larger for a ratio of f_p/f_v =0.5. Depending on the *velocity* pulsation amplitude and frequency it is further shown that errors smaller than 3% occur for f_p/f_v ratios >2-3 which confirms the statement as made by the ISO TR 3313.

It is noted that the error due to lock-in is mainly negative but also positive errors may occur if the vortex frequency is close to a pulsation frequency [15].

8.2.2 Errors caused by external vibrations

Vortex meters are sensitive to mechanical vibrations and the ISO 12764 [16] states: “Vibration of the vortex meter and associated piping should be within the levels recommended by the vendor”.
Allowable vibrations for the piping system of reciprocating compressors are defined in the ISO 20816-8 *“Mechanical vibration — Measurement and evaluation of machine vibration —Part 8: Reciprocating compressor systems”,* first edition 2018-8 [17]. The allowable piping vibration level for Evaluation zone B/C of this ISO is 19 mm/s RMS. TNO [15] has carried out measurements to investigate the effect of pipe vibrations on the reading of vortex flow meters.
The system was excited up to the allowable ISO 20816-8 level of 19 mm/s RMS and it was shown that large flow reading errors were measured in three directions from 5-100% depending on the frequency and excitation direction. That means that the vibration levels at the location of the vortex flow meter shall be much lower than the value as specified in the ISO 20816-8 to avoid gross errors.

It is noted that the sign of error due to external vibrations can be positive or negative.

8.3 Guidelines how to reduce flow meter errors

Appendix D.3 gives a flow chart with an overview of flow meter errors, the calculation methods and how to reduce them but a detailed description is given in the following sections.

8.3.1 Installation of flow conditioners and flow straighteners

For some general information see the first paragraph of section 6.3.2.
The required minimum upstream and downstream straight lengths required for vortex flow meters are specified in Annex E of the ISO 9951 [13].
However, a flow conditioner as described in section 7.4 and Annex C of ISO 5167-1 [3], and in Annex E of the ISO 9951, specifically for vortex flow meters, will permit the use of much shorter upstream pipe lengths. Such a flow conditioner shall be installed upstream of the primary device where sufficient straight length, to achieve the desired level of uncertainty, is not available.

Reduction of pulsation amplitude or avoiding lock-in with vortex frequency. The flow meter error caused by *velocity* pulsations can be reduced by either a reduction of the amplitude of the *velocity* pulsations or by avoiding a lock-in of the excitation frequency with the vortex frequency. Both reduction methods are described more into detail as follows:

Reduction of velocity amplitude
According to section 6.3.1.2. of the ISO TR 3313, the threshold value for a steady state flow condition is as follows:

$$(U'_{RMS}/\bar{U}) \leq 0.03 \qquad (8\text{-}3)$$

If the steady state requirement is fulfilled it can be assumed that *velocity* pulsations will not cause a flow meter error.
However, if the steady state requirement according to equation (8-3) cannot be fulfilled it shall be assumed that the flow meter is liable to locking-in if the frequency of the *velocity* pulsation is in a range of 0.25 times the lowest to at least twice the maximum vortex frequency (<0.25 f_p/f_v > 2) The methods as described in section 6.3.2 through 6.3.7 can be applied to reduce the amplitude of the *velocity* pulsations to fulfil the requirement of the threshold value.

Avoiding lock-in with vortex frequency
It is shown in section 8.2.1 that if the ratio of the pulsation and vortex frequency is smaller than 0.25 (f_p/f_v <0.25), or larger than 2 (f_p/f_v >2), lock-in will not occur. Furthermore, it is shown that if the ratio is large enough (*f_p/f_v>2-3)* errors of smaller than 3% can occur. To be on the safe side, an additional margin of 20% is recommended and are as follows:

- *$f_p/f_v < 0.2$ or,*
- *$f_p/f_v > 3.6$*

The requirement is that the meter vendor shall provide the vortex frequency *f_p.*

It is noted that if the meter is used for custody, the margin for the ratio $f_p/f_v > 3$ may be too low. In this case the additional margin shall be discussed and agreed upon with the purchaser and meter vendor.

The methods as described in sections 6.3.3 through 6.3.7 can be applied to find the optimal ratio of f_p/f_v for all possible operation conditions.

8.3.2 Installation of other type of flow meter

As explained in section 8.3.1, the flow meter error depends on the construction of the flow meter (bluff body shape, blockage ratio). That would mean that the optimum vortex flow meter could be designed for the application of interest. However, this can only be achieved by detailed CFD calculations or measurements assuming that the flow pulsation amplitude and frequency are known which makes it very expensive. Furthermore, there is not an equation to calculate the flow meter error for vortex flow meters which makes it difficult to quantify if the calculated or measured *velocity* pulsations are acceptable.

In many cases the flow meters are already ordered before the pulsation and vibration analysis is carried out. If rigorous system changes are required to achieve acceptable flow meter errors, it is sometimes more cost-efficient to consider another type of flow meter which is less sensitive to one or more of the disturbances for the flow meter of interest.

8.3.3 External vibrations

Vortex meters are sensitive to mechanical vibrations and the ISO 12764 [16] states: “Vibration of the vortex meter and associated piping should be within the levels recommended by the vendor”.
Allowable vibrations for the piping system of reciprocating compressors are defined in the ISO 20816-8 [17]. As explained in section 8.2.2, the pipe vibrations at the location of the flow meter shall be much lower than the 19 mm/s RMS in the three perpendicular directions.

This can be achieved in two ways:

1. Separation of excitation frequency with Vortex frequency

The structure on which the pipe support/flow meter is mounted shall be stiff enough to avoid that the dominant frequency of the external pipe vibrations of the section in which the flow meter is mounted, shall not coincide or is close to the vortex frequency (f_v). The external vibrations can be caused by pulsations in the pipe or by external vibration sources in the vicinity of the section in which the meter is mounted. This separation criterion is as follows:

- $f_{vpipe}/f_v < 0.2$ *or,*
- $f_{vpipe}/f_v > 3.6$

In which:

f_{vpipe} is the frequency of the external pipe vibrations

Notes:

- It is noted that in general the pulsations in the pipe will be the dominant excitation source and in that case the frequency of the external pipe vibrations is the same of the dominant pulsation frequency ($f_{vpipe} = f_p$).
- To achieve the above requirement, the stiffness of the support structure on which the flow meter is installed shall fulfil the above requirement. This might lead to either a reduced stiffness or an increased stiffness. In general, the safest solution is to shift the ratio to a value $f_{vpipe}/f_v > 3.6$.
- The pipe vibration frequency will change for different loading conditions of the compressor and it might be difficult to fulfill the above requirement for a compressor with a variable speed.

The calculation of the dominant pulsations and pipe vibrations can be carried out with a mechanical response analysis according to the API Standards 618.

2. Achieving acceptable vibration levels

If it is not possible to meet the requirement of the separation margin the target shall be to keep the vibrations at the location of the flow meter as low as possible in all three perpendicular directions. This can be carried out with a mechanical response analysis according to the API Standards 618.

The allowable vibration level for the flow meter shall be defined at a very early stage of the design of the system and shall be given by the flow meter vendor. However, an allowable vibration level for the Vortex flow meter is not defined in any standard, guideline or publication.

If an allowable level is not specified, it is recommended to use a level of 25-50% according to evaluation zone A/B (12.7 mm/s RMS) of the ISO 20816-8 of the piping, which gives a value of 3.2-6.4 mm/s RMS. This proposed value shall be applied only at the flow meter location. This acceptable level shall be discussed and agreed upon with the compressor vendor, purchaser, meter vendor and other relevant parties.
An acceptable level can be achieved by e.g. stiffening the structure on which the flow meter is installed and an example is shown in Figure 8.3.

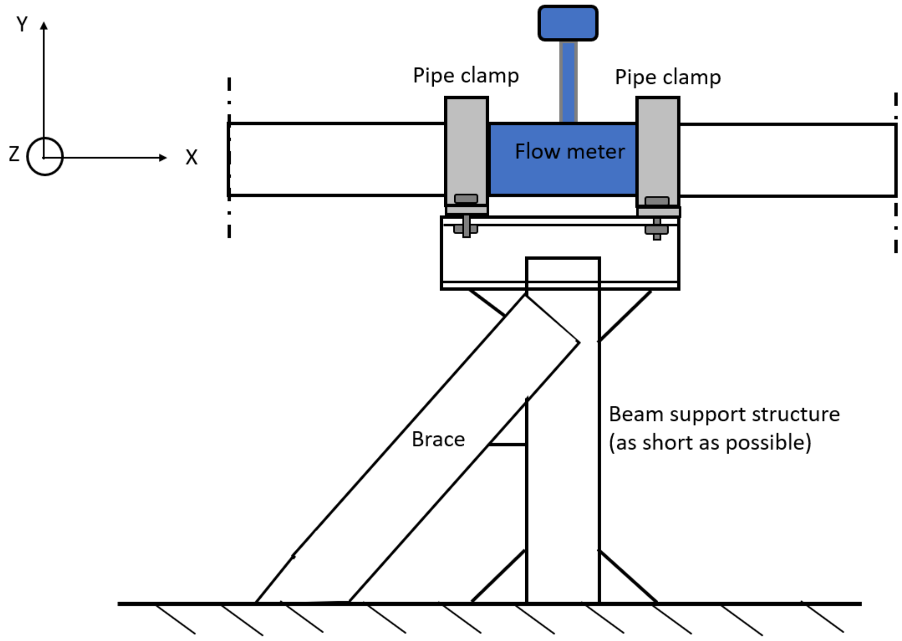


Figure 8.3 Example of a robust and stiff pipe support of a vortex flow meter

9 Coriolis Meter

9.1 Working principle

A Coriolis Mass-Flow Meter (CMFM) is an active device based on the Coriolis force principle for direct mass flow measurements with a high accuracy, range-ability and repeatability. The working principle of a CMFM is shown in Figure 9.1. If a fluid with velocity *v* flows through a tube which is rotating around point O with an angular velocity *ω*, the Coriolis acceleration of the fluid will result in a reactive force *F.*

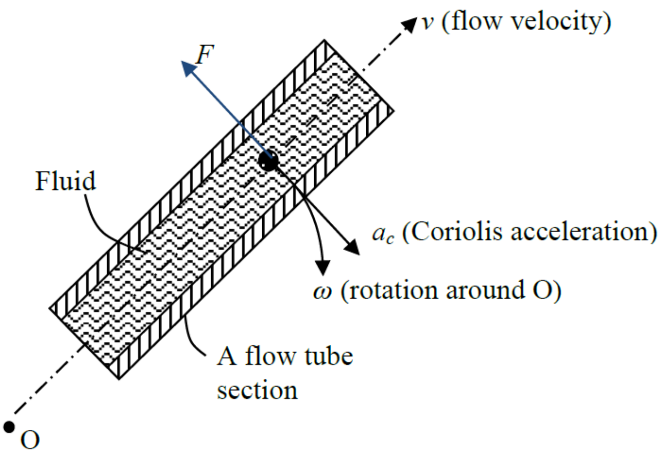


Figure 9.1Coriolis principle (source: Tao Wang, R Baker [XX])

Figure 9.2 shows a simple Coriolis meter where a fluid conveying tube is mechanically actuated with a sinusoidal signal at the location of the red arrow to oscillate with a low amplitude. The Coriolis forces are indicated by the green arrows in the middle picture. These Coriolis forces are working in opposite direction on both sides of the tube and are proportional to the mass-flow. They affect the tube motion and change the mode shape, in general the second mechanical natural frequency (MNF), as shown in lower picture of Figure 9.2.

The frequency of excitation is called the drive frequency f_1 and is in general the frequency of the first mode shape of the measuring tube. The frequency of the vibration mode corresponding to the shape of the distortion introduced by the Coriolis forces is called the Coriolis frequency. For a simple straight tube meter driven in its first mode (f_1), the Coriolis frequency would be the frequency of the second mode (f_2). It is noted that for a meter not driven in its fundamental mode, there will be two Coriolis frequencies, an upper Coriolis frequency, above the drive frequency and a lower Coriolis frequency below the drive frequency.

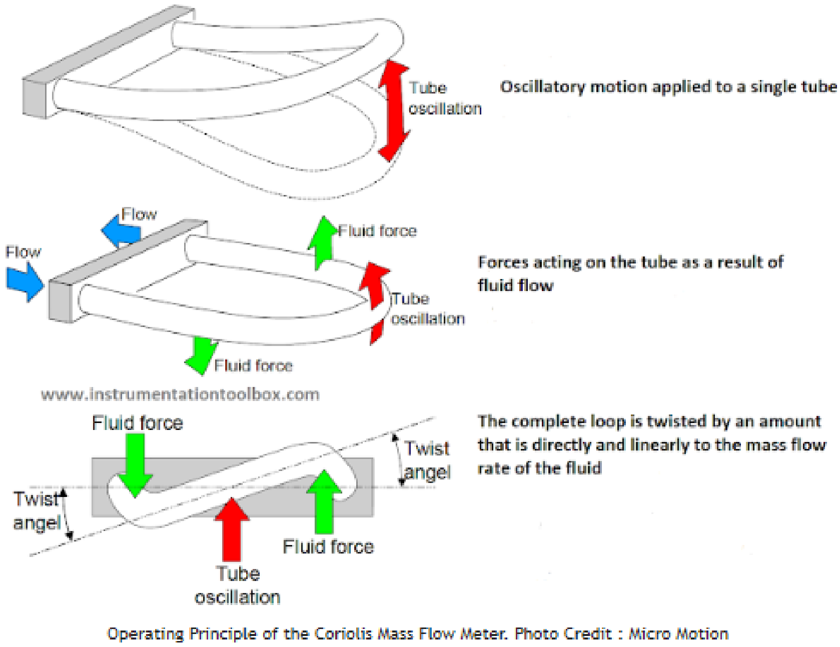


Figure 9.2 Operating principle of a single tube Coriolis mass flow meter (source: Micro Motion)

The sinusoidal motion of the opposite parts of the tube are no longer in phase caused by the Coriolis forces as shown in Figure 9.3. The mass flow rate is in direct proportion to the time delay (t_d) between these two sinusoidal signals.

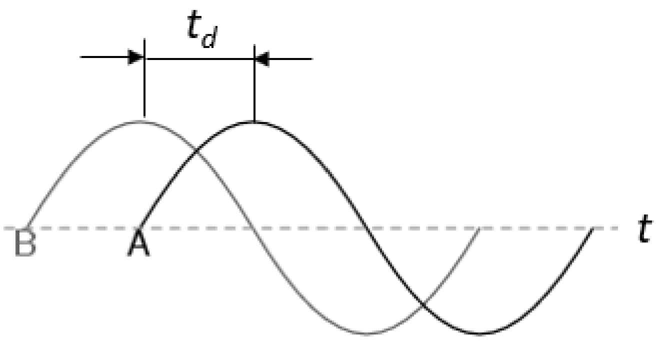


Figure 9.3Time delay caused by Coriolis forces

It is noted that the time delay is shown at the peaks of the sinusoidal signals and this is only for the purpose of illustration. In theory, the time delay can be measured anywhere along two sinusoidal signals. Therefore, different vendors can have different techniques to measure this time delay.

An important part of the flow meter is the signal processing of the measured phase difference. Vendors use different techniques which also determine the accuracy of the flow meter. Measurements by Cheesewright et al. [22] on the effect of flow disturbances at the inlet of the flow meter showed that swirl or asymmetric flow conditions do not have an important effect on the accuracy of the flow meter. This was also shown in a numerical study of Bobovik et al. [30]. A summary of these installation effects are also shown in Table 9.1 of section 9.2.1

More general information on Coriolis meters such as physical installation, effects of process and fluid properties, pressure loss, safety, inspection and compliance, mass flow measurement, density measurement and volume flow measurement can be found in the ISO 10790 [18].

9.2 Flow meter errors

9.2.1 Introduction

External vibrations and *velocity* pulsations may affect the Coriolis meters. However, these effects are difficult to quantify since they depend on the specific conditions, the type and vendor. Despite the fact that the quantification is very difficult, a summary is given of the most important influences to be able to judge if (relative) mitigation measures are required for systems which that are subject to external mechanical vibrations and pulsating flows.
Reciprocating compressors cause pulsations and pipe vibrations which may have a large influence on the Coriolis flow meter. The pipe vibrations can be caused by:

1. Direct excitation by pulsation-induced forces acting on the Coriolis duct pipe. Due to the fact that the tube length is rather short in comparison with the acoustic wave length of the gas, these direct forces are rather small, and this effect might be small therefore.
2. Indirect excitation by external vibrations caused by e.g. rotating equipment in the vicinity of the meter which are transferred via the structure (called structure borne vibrations) to the pipe system in which the flow meter is installed. Depending on the amplitude and frequency, this indirect effect is probably larger than the effect by direct excitation.

There are also several other factors which may influence the flow meter accuracy which are summarised as follows:

- Process conditions such as the effect of temperature and inner pressure.
- Asymmetrical actuator and detector probes.
- Structural non-uniformities.
- Uneven flow rates in e.g. a double flow tube meter as shown in Figure 9.4

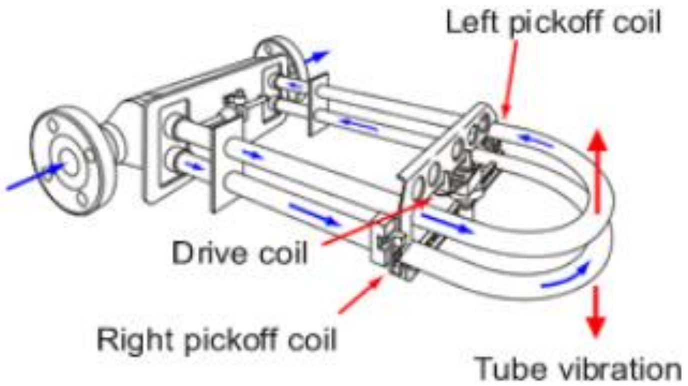


Figure 9.4 Example of a double tube Coriolis flow meter (source OnoSokki)

However, these factors are even more difficult to quantify because most of them are related to the manufacturing process of the meter and cannot be controlled by the user of the system.

It is noted that there are many types of Coriolis meters available, whereby the size depends on the flow range. One category includes meters for low flow rates and the Coriolis force induced motion is relatively small compared to external vibrations. These meters are rather sensitive to external vibrations. This situation can occur for reciprocating compressors, for example for the unloaded conditions.
Table 9.1 shows a summary of the results on the effects of external factors on the accuracy of meter calibration as published by R. Cheesewright et al. [22].

Table 9.1 The effect of external conditions on the calibration of Coriolis flow meters (source [22]) for different meters

	Changes in meter calibration?		
External condition	Meter A	Meter B	Meter C
Flow pulsations at Coriolis frequency	Severe	Severe	Severe
Flow pulsations at drive frequency	No (mean)	No (mean)	Small (mean)
Mechanical vibrations at Coriolis frequency	Severe	Severe	Severe
Mechanical vibrations at drive frequency	No (mean)	No (mean)	Small (mean)
Disturbances at other frequencies which indirectly excite the Coriolis frequency	Yes	Yes	Yes
Swirl in the inlet flow	No	No	No
Asymmetric inlet profile	No	No	Small
Increased turbulence in inlet flow	No	No	No
Two-phase flow (air/water)	Yes	No	Yes
Cavitation	Small	No	No
Installation stresses	No	No	No

9.2.2 *Velocity pulsations*

There are two effects caused by flow pulsations [23, 24, 25, 26] which can cause large flow meter errors as follows:

1. *Direct* excitation of the tube at the drive frequency f_1 caused by pulsation-induced *forces* acting on the tube(s). This is especially the case if there is an acoustic resonance with a pulsation frequency f_p which coincides with the drive frequency f_1.
2. *Indirect* excitation of vibrations if the pulsations beat with the drive frequency f_1 resulting in tube vibrations with frequencies of the sum and difference of the pulsation frequency and drive frequency, respectively $f_p + f_1$ and $|f_p - f_1|$. The *velocity* pulsations have the strongest effect on beating with the drive frequency at $f_p + f_1$ when the dominant frequency of the pulsations coincides with the Coriolis frequency f_2.

It is noted that according to several publications the error can be either positive or negative.

9.2.3 *External mechanical vibration*

Several research projects have been carried out, both numerically and by measurements, to investigate the effect of external vibrations on the flow meter effect [19, 20, 21, 22].
The results of one of the most recent publications to the effect of external vibrations on the flow meter error is from Ridder et al. [21]. This study shows that external vibrations with a frequency of the drive frequency f_1 produce a meter error regardless of the flow meter post processing algorithm. Furthermore, external vibrations with a frequency of the Coriolis frequency f_2 will also produce a flow meter error. The most sensitive directions to external vibrations are translations in the Y-axis and a rotation around the fixation points of the flow tubes which is around the X-axis as shown in Figure 9.5.

Due to the fact that it is not possible to quantify the flow meter error, the most efficient method in reducing the flow meter error caused by external vibrations is to avoid that the frequency of the dominant vibrations do not coincide or are close to the drive frequency *f1* and Coriolis frequency f_2 . In section 9.3 methods are given how to reduce the amplitude of the vibrations thereby reducing the flow meter error.

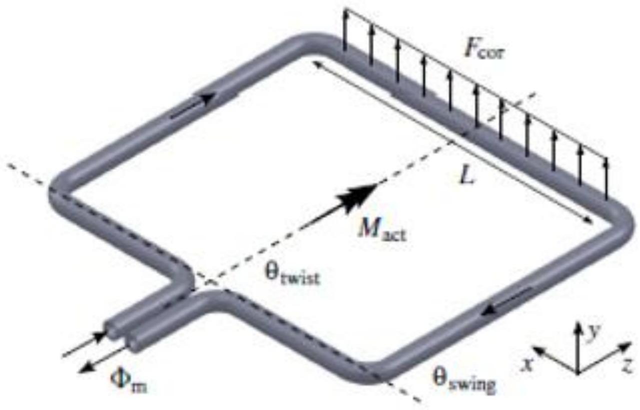


Figure 9.5 Example of a single tube Coriolis meter with a definition of the axes (source [21])

An obvious solution to reduce the influence of external vibrations would be to apply a robust balancing system, e.g. a twin tube configuration. However, applying a twin tube configuration is not an option because some structural non-uniformities can lead to large differences between the two tubes due to their small dimensions. This has a negative impact on the measurement sensitivity of the instrument and reduces the decoupling of external vibrations to the internal measurement system.

A summary of methods for the reduction of the flow meter errors caused by external vibrations is given in section 9.3.1.

9.3 Methods how to reduce flow meter errors

9.3.1 Introduction

In section 9.2.2 and 9.2.3 a summary is given of the causes of the flow meter errors. In general, flow meter errors for a Coriolis meter are the highest when an acoustic, mechanical or a combination of both resonances occur in the pipe section in which the flow meter is installed. The occurrence of these frequencies can be calculated with a detailed pulsation and mechanical response analysis according to e.g. the API 618 [1] if it concerns a system during the design stage. For an existing system the frequencies of the *velocity* pulsations and vibrations shall be measured.

The flow meter error cannot be easily quantified, based on basic parameters such as pulsation and vibration amplitudes. One of the most effective methods of reducing the amplitude is the installation of additional damping with e.g. an orifice plate. However, due the fact that it is not possible to quantify an allowable level, this method can only be used to obtain a relative improvement.

The most efficient method for the reduction of the flow meter error is to ensure that the frequency of the dominant pulsations (f_p) and vibrations shall not coincide with the drive (*f_1*) and Coriolis frequency *(f_2).* To be able to follow this approach, these frequencies shall be provided by the meter vendor to the purchaser.

Recommendations to reduce the external vibrations and flow pulsations with a frequency of the drive and Coriolis frequency are given in section 9.3.2. and in 9.3.3 respectively. The worst-case situation which can occur is a combination of an acoustic and mechanical resonance. In that case a combination of the reduction methods as described in both sections are most effective.

Appendix D.4 gives a flow chart with an overview of flow meter errors, the calculation methods and how to reduce them but a detailed description is given in the following sections.

9.3.2 Shifting pulsation frequency

It is shown that large errors can occur if the frequency of the *velocity* pulsations coincide with the drive and Coriolis frequency. The reduction methods shall focus on avoiding a coincidence of the dominant frequency of the *velocity* pulsations with both the drive (*f_1*) and Coriolis (*f_2*) frequency. It is recommended to use a separation margin of minimum ±20%.It is noted that one meter can have more than one drive and Coriolis frequency and the recommendations apply for all drive and Coriolis frequencies.

A change in length of the pipe system can be very effective in detuning/shifting the frequency of a standing wave or Helmholtz type resonance so that it does coincide anymore with the frequency of the dominant pulsations. Some do's, don'ts and engineering rules on this are given in section A.5 of Appendix A.

It is noted that there is a possibility that the amplitude of pulsations in the piping and Coriolis tube can increase. However, as long as the frequency is far enough from the drive and Coriolis frequency, this shall not be a problem.

It is furthermore noted, that shifting the frequency of the excitation forces is in many cases very challenging for reciprocating compressors due to the fact that the pulsations might can contain more than one frequency component. This is even more difficult for compressors with a variable speed because the frequency of the dominant excitation forces will change for the complete speed range.

9.3.3 Relocation of flow meter

Despite the fact that it is not based on avoiding a coincidence of the drive and Coriolis frequency, an effective method can be to (re)locate the flow meter to a location with a *velocity* node as also described in section 6.3.5.1. Even when there is a coincidence of the pulsation with the drive and Coriolis frequency. This will reduce the flow meter error caused by the *indirect* excitation of vibrations of the Coriolis tube, see also section 9.2.2.
However, one shall be very careful with this method because the flow meter error caused by *direct* (acoustic) excitation of the Coriolis tube may increase caused by an increase of the pulsation forces on the tube.

The only disadvantage of this method is that the pulsation forces with a frequency of the drive or Coriolis frequency, causing the external vibrations of the pipe section in which the flow meter is installed, will not be reduced.

9.3.4 Acoustic filter

An acoustic filter as also discussed in section 6.3.7, might also be a good solution for some cases if the drive and Coriolis frequency can be filtered out.
The disadvantage of this method is that it is more expensive than other reduction methods.

9.3.5 Installation of a meter which is less sensitive for flow meter errors

See section 6.3.9.

9.3.6 External vibrations

Some general methods in achieving the lowest external vibrations of the pipe section in which the flow meter is installed are as follows:

1. Separation of the frequency of vibrations from the drive and Coriolis frequency
Identical to the frequency of the pulsations as discussed in section 9.3.2, a coincidence of the frequency of the dominant external vibrations with both the drive (f_1) and Coriolis ($f2$) frequency shall be avoided. It is recommended to use a separation margin of minimum ±20%.
This can be very challenging due to the fact that the drive and Coriolis frequencies shall be provided by the meter vendor. Besides that, most of the commercial meters have more than one drive and Coriolis frequency. To achieve the separation margin is even more challenging for variable speed compressor systems. Separation can be achieved by the installation of one or more (inlet and outlet of flow meter) stiff pipe supports and is most effective if the vibrations are caused by a mechanical resonance.

In section 9.2.3 it is explained that the flow meter has its highest sensitivity for vibrations in the Y-axis and around the fixation points of the flow tubes around the X-axis a shown in Figure 9.5. To achieve this, it is recommended to mount the flow meter to the supporting structure by using rigid pipe clamps on a minimum one of the flanges as shown in Figure 9.6. Furthermore, the construction on which the pipe support is mounted shall be stiff enough to avoid that the frequency of the external pipe vibrations at the location of the flow meter shall not coincide or is close to the drive (f_1) and Coriolis ($f2$) frequency (separation margin as explained before shall be fulfilled). The beam structure shall be as stiff and as short as possible. The most effective method in achieving this is by the installation of a brace perpendicular to the pipe as shown in the example in Figure 9.6.

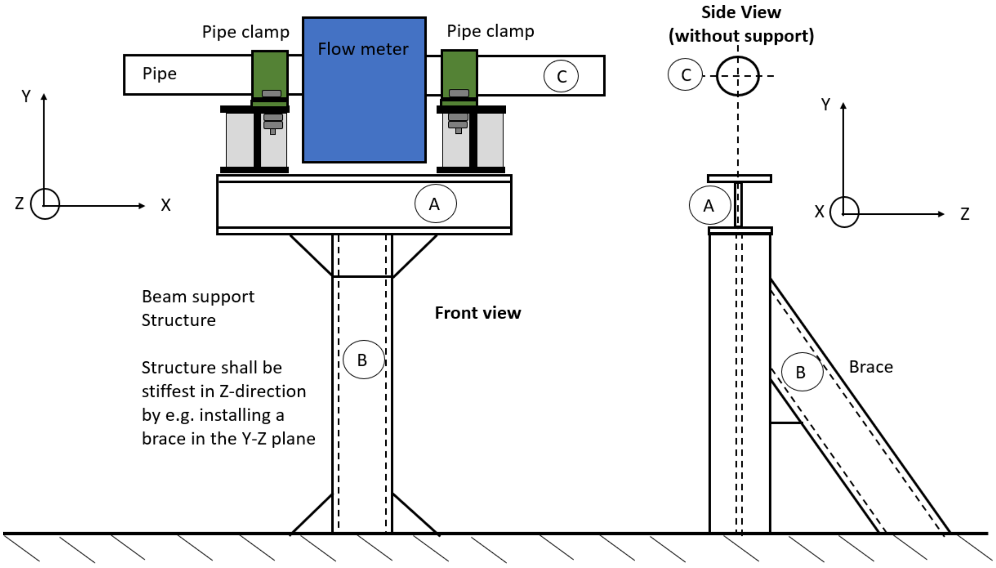


Figure 9.6 Example of a stiff robust support for Coriolis meter

2. Calculation of vibrations

If it is not possible to meet the requirement of the separation margin of minimum ±20% for the drive and Coriolis frequency, the target shall be to keep the vibrations at the location of the flow meter as low as possible in all three perpendicular directions. This can be carried out with a mechanical response analysis according to the API Standards 618.

The allowable vibration level for the flow meter shall be defined at a very early stage of the design of the system and shall be given by the flow meter vendor. However, an allowable vibration level for the Coriolis flow meter is not defined in any standard, guideline or publication.
If an allowable level is not specified, it is recommended to use a level of 25-50% according to evaluation zone A/B (12.7 mm/s RMS) of the ISO 20816-8 of the piping, which gives a value of 3.2-6.4 mm/s RMS. This proposed value shall be applied only at the flow meter location. This acceptable level shall be discussed and agreed upon with the compressor vendor, purchaser, meter vendor and other relevant parties.
An acceptable level can be achieved by e.g. stiffening the structure on which the flow meter is installed and an example is shown in Figure 9.6.

If necessary, the RMS value can be converted to a peak-to-peak value if the vibration consists of one dominant frequency. This can be achieved by multiplying the overall RMS value by $2\sqrt{2}$.

10 Ultrasonic flow meter (USM)

10.1 Working principle

An ultrasonic flow meter (USM) is a type of flow meter that measures the velocity of a fluid with ultrasound to calculate volume flow. Using ultrasonic transducers, the flow meter can measure the average velocity along the path of an emitted beam of ultrasound, by averaging the difference in measured transit time between the pulses of ultrasound propagating into and against the direction of the flow or by measuring the frequency shift from the Doppler effect. USMs are affected by the acoustic properties of the fluid and can be impacted by temperature, density, viscosity and suspended particulates depending on the exact flow meter.

There are important benefits to using ultrasonic meters. The most significant, however, is the ability to diagnose the meter's health. The main question users want to know is whether the meter is still accurate after having been in service for some period of time in the field.
Other primary measurement devices such as orifice and turbine meters offer little insight into whether they are still operating accurately after some period of time. Issues such as contamination from pipeline oil and mill scale can impact the accuracy of any meter. Visual inspection is often required to validate proper operation for traditional primary measurement devices. USMs, on the other hand, offer electronic diagnostics that can help validate proper operation, and thus reduce the internal inspection requirements often required by other devices. These internal diagnostics can also be used to help identify whether the other components at the measurement station, such as temperature measurement and gas composition, are also operating correctly.

USMs measure the difference between the transit time of ultrasonic pulses propagating with and against the flow direction as shown in Figure 10.1 and equation (10-1) and (10-2). This time difference is a measure for the average velocity of the fluid along the path of the ultrasonic beam. The meter may consist of a single or multiple path of which the last one has at least two independent pairs of measuring transducers (acoustic paths).

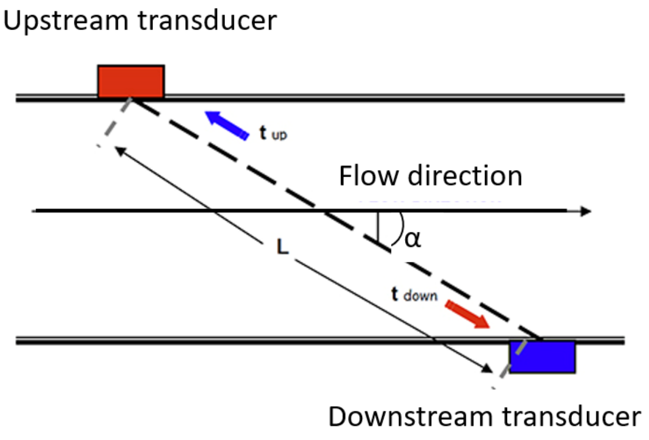


Figure 10.1 Principle of a single path USM

$$t_{BA} = \frac{L}{c - \bar{U}_G \cos(\alpha)} \tag{10-1}$$

$$t_{BA} = \frac{L}{c + \bar{U}_G \cos(\alpha)} \tag{10-2}$$

In which:

t_{AB}	transit time of ultrasonic pulses propagating with the flow direction [s]
t_{BA}	transit time of ultrasonic pulses propagating against the flow direction [s]
$\bar{U}_G$	flow path average flow velocity m/s]
c	speed of sound [m/s]
α	inclination angle between the upstream and downstream transducer [-]
L	distance between receiving and transmitting transducers

By using the absolute transit time t_{AB} and t_{BA}, both the averaged fluid velocity of the flow path average flow velocity $\bar{U}_G$ nd the speed of sound an be calculated as shown in equation (10-3) and (10-4) as follows:

$$\bar{U}_G = \frac{L}{2\cos(\alpha)}\left(\frac{1}{t_{AB}} - \frac{1}{t_{BA}}\right) \tag{10-3}$$

$$c = \frac{L}{2}\left(\frac{1}{t_{AB}} + \frac{1}{t_{BA}}\right) \tag{10-4}$$

From equation (10-3) it is shown that the pressure, temperature, and gas composition have no effect on the flow path velocity calculation from the pulse transit time.

Doppler shift flow meters
Another method in ultrasonic flow metering is the use of the Doppler shift that results from the reflection of an ultrasonic beam off sonically reflective materials, such as solid particles or entrained air bubbles in a flowing fluid, or the turbulence of the fluid itself, if the liquid is clean. Doppler flow meters are used for slurries, liquids with bubbles, gases with sound-reflecting particles.

These equations look relatively simply, and they are. The primary difference between computing gas velocity and speed of sound is that the *difference* in transit times is used for computing velocity, where the *sum* of the transit times is used for computing speed of sound.

Determining the correct flow rate within the meter is more comprehensive than it appears. The path velocity equation from equation (10-3) refers to the velocity of each individual path. The velocity needed for computing volume flow rate, also known as bulk mean velocity, is the average gas velocity across the meter's area.

In the pipeline the gas velocities are not always uniform and often there is some swirl and asymmetrical flow profile within the meter. This makes computing the average velocity a bit more challenging.

Meter vendors have different methodologies for computing this average velocity. Some derive the answer by using proprietary algorithms. Others rely on the design that does not require "hidden" computations.

Regardless of how the meter determines the bulk average velocity, the following equation is used to compute the uncorrected flow rate as follows:

$$\bar{Q} = A\bar{U}_G \tag{10-5}$$

In which:

$\bar{Q}$	average flow rate [m^3/s]
A	flow meter area [m]
$\bar{U}_G$	average flow velocity [m/s]

Summarising: points to keep in mind about ultrasonic meter operation are:

- The measurement of transit time, both upstream and downstream, is the primary function of the electronics.
- All path velocities are averaged to provide a "bulk mean" velocity that is used to compute the flow rate.
- Because the electronics can be determined which transit time is longer (t_{AB} or t_{BA}), the meter can determine the flow direction.
- Speed of sound is computed from the same measurements as gas velocity.

10.2 Flow meter errors

10.2.1 Flow disturbances (upstream piping and flow profiles)

It is well known that flow disturbances such as pulsations, disturbed flow and flow noise, also named installation effects, highly affect the performance of USM meters resulting in a flow meter error. In general, the single path USMs are more sensitive to flow disturbances than the multi path USMs, e.g. a 4-path USM.
This effect is difficult to quantify, and these disturbances shall be avoided as much as possible. This effect can be minimised by installing the flow meter far enough from any disturbances such as bends, tee joints, valves with a large pressure loss, reducers, inlet and outlet of volumes (pulsation dampers, separators, etc.), see also section 10.3.2.

10.2.2 Summary of USM errors according to the PRCI Report "Meter station design procedures to minimize pipe flow-induced pulsation errors", [32] dated 2013

Chapter 6 of the PRCI report gives a very good overview of the mechanisms which can cause flow meter errors in pulsating flow which are briefly summarised below. Some of them are related to fluid phenomena and others are related to a function of the specific meter's data acquisition methods.

10.2.2.1 Pulsations

Pulsations caused by reciprocating compressors can change the shape of the velocity profile as shown in Figure 10.2. when measured with the transit time method. The top of the figure shows the theoretical flow profile at one location in the pipe system for different times. The bottom picture shows the amplitude of the pulsations as a function of time and it shows that pulsations cause a change in velocity profile. The shape of this figure is shown for a laminar flow, but the similar profile can occur for a turbulent flow which was confirmed by measurements of Mizushina [33]. The shape is flattened when the pulsations are accelerated and peaked during deceleration.
The changes in velocity profile may be incorrectly sampled by an USM leading to a flow meter error. It is stated by McKee [34] that USMs can be adjusted for this phenomenon. It is therefore strongly recommended to consider an USM design which is suitable to compensate for pulsation effects. This would be an advantage of the USMs over the other meters as discussed in these guidelines.

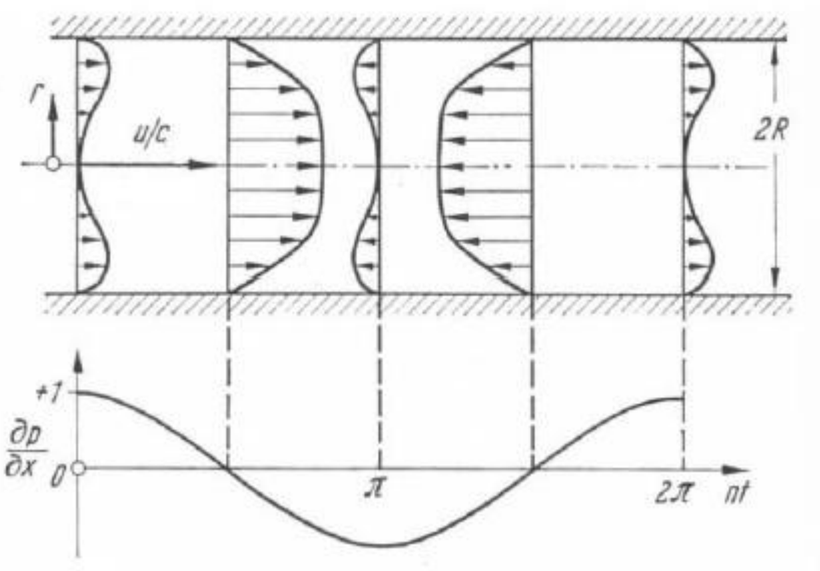


Figure 10.2 Velocity distribution at one location in oscillating laminar pipe flow at different times in the pulsation cycle (source: Boundary Layer Theory, Schlichting 1968)

10.2.2.2 Distortion of meter generated pulses

Pulsations can distort the ultrasonic pulses. Especially high frequency broadband pulsations, as explained in chapter 3 of this document, can distort the velocity profile leading to flow meter errors. In general, the situation of a flow meter, on the downstream side of a pressure let-down device (HF broadband source) is not often found, certainly not in reciprocating compressor systems.

10.2.2.3 Aliasing

Aliasing is an effect that causes different signals to become indistinguishable when sampled. It also often refers to the distortion or artefact that results when a signal reconstructed from samples is different from the original continuous signal. This happens when the signal is not sampled with a high enough sampling rate resulting in a waveform with a different frequency and amplitude than the original signal. This is the case when the sampling frequency is lower than the frequency of the velocity pulsations (not the frequency of the ultrasonic pulses) and it will lead to a flow meter error. In general, the sampling frequency of USMs is between 5-100 Hz (depending on the brand and type). These frequencies coincide reasonably well with typical pulsation frequencies of reciprocating compressors.

Several references (TNO, Peters and Bokhorst [15], AGA Report 9 [31]) cite this aliasing effect as the primary cause of flow meter error caused by pulsations. TNO has measured aliasing errors over 5% at velocity pulsations of 10% RMS, even for pulsations with a frequency lower than the sampling frequency.
If the maximum frequencies of the sampled *velocity* pulsations are known, which can be derived from the results of an API 618 pulsation study, the aliasing error can be avoided in theory by sampling this maximum pulsation frequency with at least twice that of the original signal which is known as the Nyquist criterion. This is feasible for low frequency pulsations as generated by reciprocating compressors but is not feasible for high frequency noise above 1000 Hz.

10.2.2.4 *Data averaging*

The meter electronics process the measured data before reporting an average flow velocity. Time averaged flow rates and diagnostic data can be reported anywhere from several times per second to one over tens of seconds. Because USMs sample the flow periodically, they may not be able to follow the flow changes in a pulsating flow. This averaging can result in a loss of information and can hamper or eliminate the ability to compensate for pulsating flow. This shall not be confused with aliasing in that it is the processing software and not the waveform detection that leads to the flow meter error. Depending on the brand and type, USMs may be capable of measuring pulsations but in general due to the low reporting rate, the information is averaged out. In general, the reporting rates for different brands and type of USMs differ and due to this the effect is not well known.

10.2.3 *Mechanical vibrations*

The measured signals of the USM are not sensitive to external vibrations. However, according to the APG Report 9 [31], too high vibrations of the pipe section in which the USM is located may lead to damage of the electronic boards and components, card cages, wiring and connectors which will negatively affect the USM performance. That means that USMs shall not be mounted in a pipe section with too high vibrations which is general the case when a mechanical natural frequency is excited by e.g. a pulsation induced shaking force which is typical for reciprocating compressor systems.

In principle, the meter vendor shall provide the maximum allowable vibration level (amplitude and frequency) which can be compared with the maximum vibration level calculated with a mechanical response analysis according to the API 618.

10.3 Methods for reducing flow meter errors

10.3.1 *Introduction*

It is known that different types and brands of USMs use different methods to generate and detect acoustic pulses to measure the velocity pulsations which also means that any errors caused by velocity pulsations are dependent on the brand and type of USM.

All errors as explained in this section of the USM, including the installation effects as discussed in the former section, cannot (yet) be predicted and they also vary by the type and brand of the USM caused by different sampling and post-processing techniques. That means that it is more difficult to apply the most efficient methods in reducing the USM errors caused by pulsations.

All methods as discussed in this section to reduce the flow meter error will therefore give a *relative* reduction based on the state of art technology at the moment of publication of these EFRC guidelines. However, it is known that USM manufactures are continuously improving their technology to reduce the errors as summarised in the previous section.

The following general statements are given by the AGA Report number 9 for situations where pulsations are expected:

- *The designer should consider the possible existence of pulsations in the vicinity of the USM caused by but not limited to flow, control valves, check valves, mechanical installation and/or induced by compression*.

- *The designer should provide an appropriate piping design or dampening solution to mitigate the potential increase in measurement uncertainty caused by pulsations.*
  A pulsation study may be required to arrive at the correct pulsation dampening equipment configuration and location.

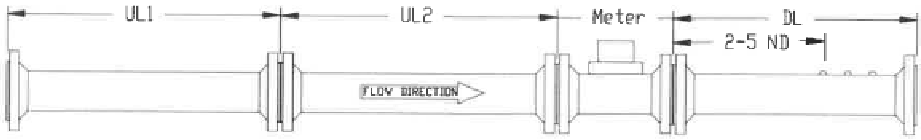
These AGA statements are rather general but for a better design of the system and choice of the USM, the next sections give more specific methods to reduce the USM meter error caused by pulsations and vibrations.

Appendix D.5 gives a flow chart with an overview of flow meter errors, the calculation methods and how to reduce them but a detailed description is given in the following sections.

10.3.2 Flow conditioners and flow straighteners

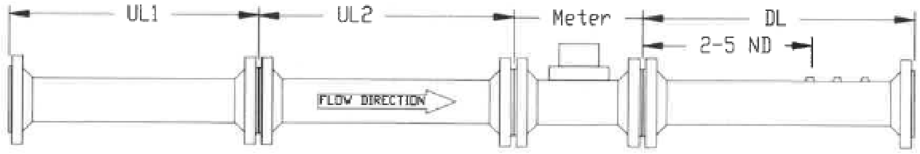
To reduce the errors caused by the installation effects as discussed in section 10.2.1, the flow meter shall be installed in the pipeline at a position such that the flow conditions immediately upstream of the USM approximate to those of swirl-free, fully developed pipe flow. In section 5.2 of the AGA Report number 9 [31] recommendations are given for the location of the flow straightener upstream and downstream of the USM to minimize the installation effects, see also Figure 10.3. From this figure it is clear that the location of the flow straightener upstream the USM shall be specified by the USM vendor.
Furthermore, it is noted that multiple path USMs tolerate more flow disturbances than a single path USM.

Option 1: A conservative configuration with a flow conditioner (between spools UL1 and UL2) as shown below. The manufacturer shall specify the flow conditioner(s) approved for use in this configuration based on independently certified test data.



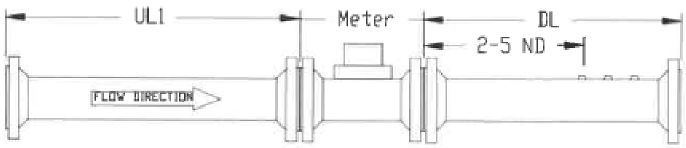
Where: UL1 = min. 10 ND length
UL2 = min. 10 ND length
DL = Variable

Option 2: Manufacturer-recommended configuration with use of a flow conditioner between spools UL1 and UL2 as shown below. The manufacturer shall specify the lengths of UL1 and UL2, as well as the flow conditioner(s) approved for use in this configuration, based on independently certified test data.



Where: UL1 = Manufacturer-specified
UL2 = Manufacturer-specified
DL = Variable

Option 3: Manufacturer-recommended configuration with one upstream spool and no flow conditioner as shown below. The manufacturer shall specify the length of UL1 based on independently certified test data.



Where: UL1 = Manufacturer-specified
DL = Variable

<u>For bi-directional flow</u>; upstream piping spool(s) and flow conditioner as applicable from Options 1, 2 or 3 can be used on both ends of the metering package.

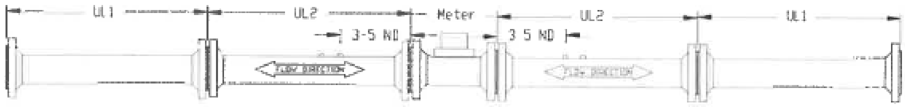


Figure 10.3 Picture from AGA Report 9 with recommendations for the location of flow conditioners

10.3.3 *Application of an USM which adjusts the meter for expected pulsations, aliasing and data averaging errors*

It is known that several USM manufactures have improved recently or are improving their technology to reduce the flow meter errors caused by flow disturbance. Reduction of the disturbing effects due to pulsations may be partly achieved as well but is generally not considered as the first priority. In general, the optimization focusses on the optimal number of paths and optimal (variable) sampling rate. This results in a robust integration algorithm to determine the flow profile.

Furthermore, a sampling frequency is required of a minimum of two times the expected/calculated frequency of the *velocity* pulsations with the highest amplitudes. In general, the maximum expected frequency of the dominant pulsations is 10 times the maximum speed of the reciprocating compressor. That means that the sampling frequency shall be minimum 20 times the maximum compressor speed. It is noted that even higher harmonic components than 10 times the compressor speed may occur which is e.g. the case for systems with a stepless flow control system. In that case the sample frequency shall be higher.

10.3.4 Reduction of pulsation amplitudes and frequencies

It is obvious that a reduction of the amplitude and frequency of the *velocity* pulsations will lead to a reduction of the flow meter error. If the frequency of the dominant *velocity* pulsations coincides or is close to the sampling frequency, the error can be reduced by filtering the sampling frequency of the pulsations. The reduction methods as described in section 6.3.3 through 6.3.7 can also be applied for the USM.

10.3.5 External vibrations

As explained in section 10.3.5, mechanical vibrations will not have an effect on the flow meter error. However, the electronics of an USM are very sensitive to mechanical vibrations and too high vibration levels may lead to damage of the electronic components. Preferably, the delicate electronic components shall be mounted off-line, if possible.

If the electronic components are mounted directly on the vibrating pipe structure, the meter vendor shall provide the maximum allowable vibration level (amplitude and frequency) which can be compared e.g. with the maximum vibration level as calculated with a mechanical response analysis according to the API 618.
Some USM vendors use the following international standards in the meter design with respect to vibrations:

- Shock Resistance: IEC 60068-2-31 [39]
- Vibration Resistance: IEC 60068-2-47 [40] and IEC 60068-2-64 [41]

However, if this information is not available, the best solution in mitigating the vibrations is to mount the USM on a very stiff pipe support structure to avoid the excitation of mechanical resonances. A sketch of a typical stiff structure is shown in Figure 9.6.

11 References

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A Acoustic wave theory

A.1 Introduction

Several methods are given in these guidelines to reduce the flow meter errors. The error of several flow meters strongly depend on the *velocity* or *pressure* pulsations which are caused by a positive displacement machine such as a reciprocating compressor. These pulsations can be amplified throughout the system by acoustic resonances such as a standing wave type or Helmholtz type resonance or by a combination of these resonances.
One of the reduction methods as given in these guidelines is based on a (re)location of the flow meter to a location with the lowest *velocity* pulsations. Several other reduction methods are based on changing the acoustic properties of the system such that acoustic resonances can be avoided e.g. by shifting the resonances to avoid a coincidence with a frequency for which a meter is sensitive. To achieve an acceptable flow meter error, all parties involved in the preliminary or final design and operation of the system such as pipe design engineer, compressor vendor, flow meter vendor and operator, shall have some background information on acoustic wave theory as given in this Appendix.
As explained in several sections of these guidelines an optimum system in achieving an acceptable flow meter error can only be determined with a detailed pulsation analysis according to the API Standard 618.

A.2 General wave theory

A.2.1 Basics in duct acoustics

A reciprocating compressor generates *velocity* (flow) pulses caused by the reciprocating action of the piston and the opening and closing of the compressor valves. The waves are travelling upstream and downstream of the excitation source with the speed of sound into the pipe system as a function of time as shown in Figure A.1. The upstream and downstream propagating waves are represented respectively by $p(x,t) = F(x + ct)$ and $p(x,t) = G(x - ct)$.

The *x* is the location along the pipe, *t* is the time and *c* is the speed of sound which can be calculated with the following equation:

$$c = \sqrt{\frac{\gamma Z R_o T}{M}} \tag{A-1}$$

In which:

- c speed of sound [m/s]
- Z compressibility factor of the gas [-]
- γ isentropic gas constant [-], being the ratio of the specific heat constant at constant pressure c_p and constants volume c_v
- R_o gas constant 8314 [kJ/Mol]

T Absolute temperature [K]
M molecular weight of the gas or gas mixture [kg/kMol]

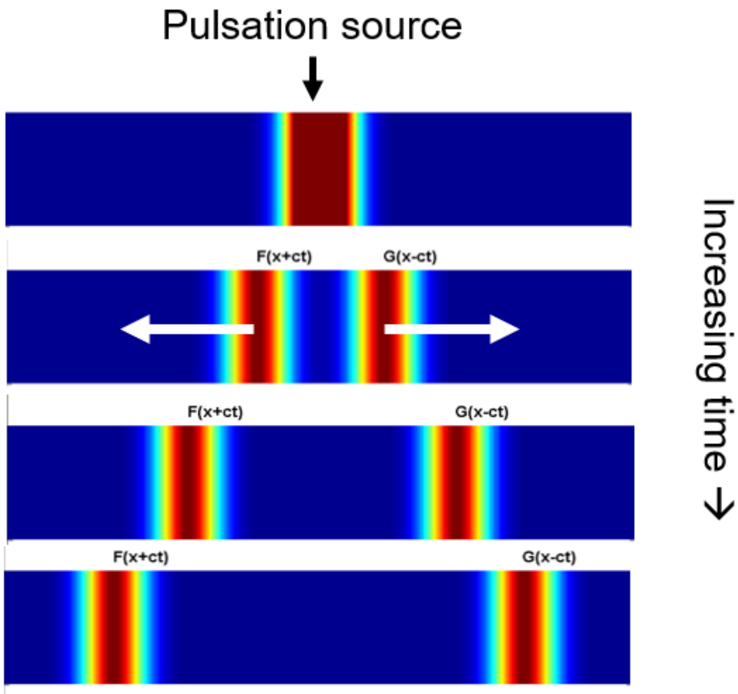


Figure A.1 Pressure pulsations as a function of time and distance

A.2.2 *Reflection and transmission at discontinuities*

Incident pressure waves are (partly) reflected and partly transmitted as shown in Figure A.1 and Figure A.2 when there is:

- a change in pipe cross section;
- a branch connection;
- a change in acoustic impedance (*ρc).*

The pulsations which are not reflected are transmitted and reflected and transmitted pulsations may also be damped by flow separation or by increasing the damping locally at a certain location e.g. by the installation of a restriction (orifice plate).

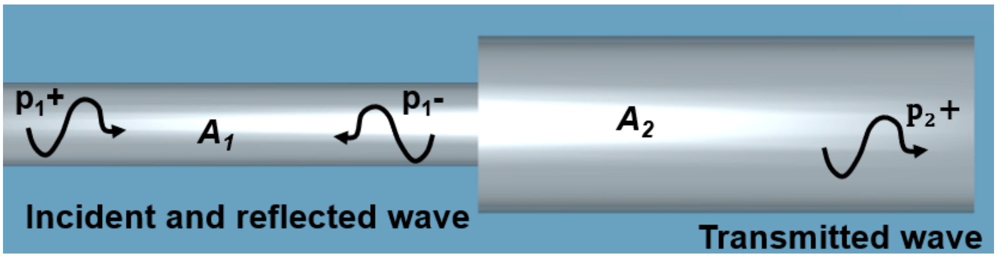


Figure A.2 Example of an incident, reflected and transmitted pressure pulsations as a function of time and distance for a change in pipe cross section

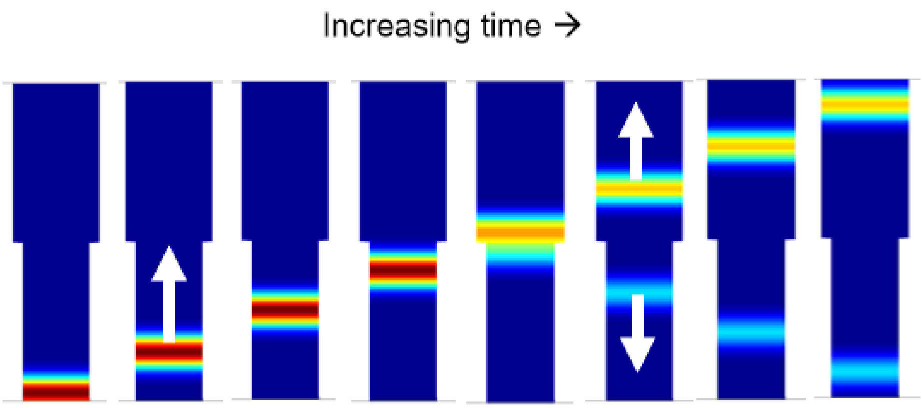


Figure A.3 Example of reflected and transmitted waves as a function of time and distance

Figure A.4 shows two examples of different reflections and transmissions. The left-hand picture is an example of a hard reflection, e.g. a large diameter change or a closed valve. It can be seen from the picture that the incident pulse is fully reflected and inverted at the reflection location. At the right-hand side picture an example is shown of an open-end reflection e.g. a large diameter change (e.g. a large header diameter) or a large volume (pulsation damper of separator). It can be seen here that the incident pulse is fully reflected without inverting at the reflection location. An open pipe termination has also a full reflection.

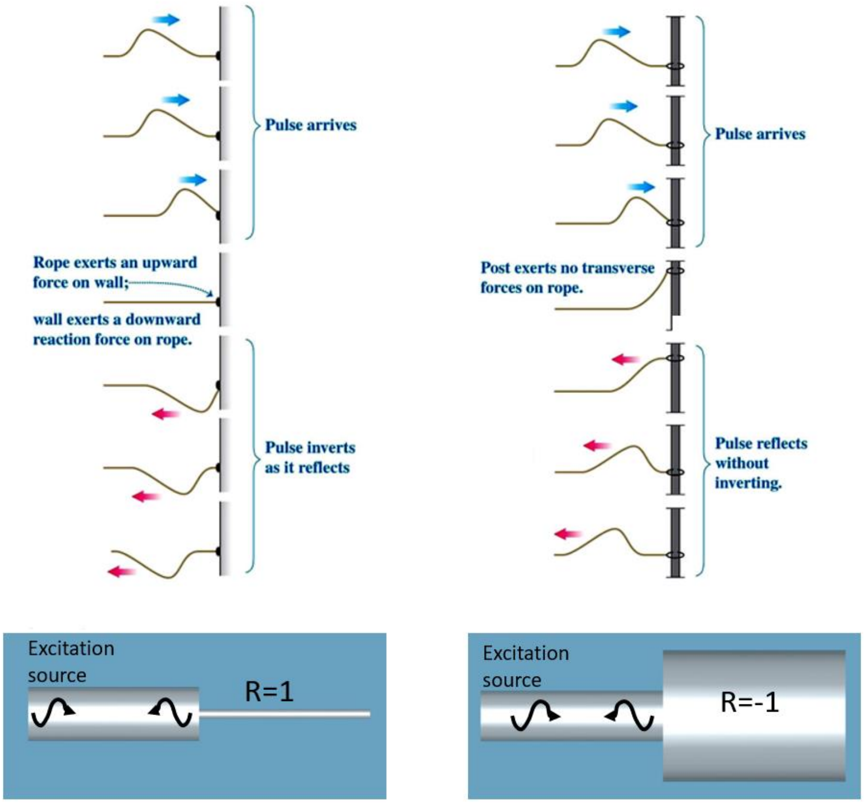


Figure A.4 Example of reflected and transmitted waves as a function of time (t) and distance (x) for a large diameter change which behaves like a closed end reflection at the left side of the picture and which behaves like an open end reflection at the right side

A.2.3 *Interference of acoustic waves*

When two or more acoustic waves (e.g. incident and reflected waves) occur at the same time in a (pipe) system, they will affect each other. The waves do not bounce off of each, but they move through each other. The resulting wave depends on how the waves add up. Two identical waves can add constructively (amplitude will increase) or destructively (amplitude will decrease) to give different results.

Example A in Figure A.5 shows a constructive interference with two waves with the same frequency and amplitude in phase. The maximum of the peaks of the two waves will add up and the result is a wave that has twice the amplitude of the original waves. This is called an acoustic resonance.

Example B in Figure A.5 shows a destructive interference when similar waves add up which are out of phase. The result is a cancellation of the waves.

Example C in Figure A.5 shows the result of any combination of waves of which the sum is simply the addition of the various complex wave forms.

Example D in Figure A.5 shows the addition of two different waves with slightly different frequencies causing "beats". The resulting wave has points of constructive interference and destructive interference. A wave with the beat pattern will have an amplitude that varies at a regular rate. An example of this wave pattern occurs when two identical compressors are running with a slight difference in speed.

Depending on the phase of the interfering waves, minima (nodes) and maxima (anti-nodes) in pulsation amplitude will occur. For flow meters of which the meter error is depending on the *velocity* pulsation, the minimum flow meter error will be achieved if the meter is installed on the location of the *velocity* node.
In the next section more detailed information will be given of the locations for different type of acoustic resonances.

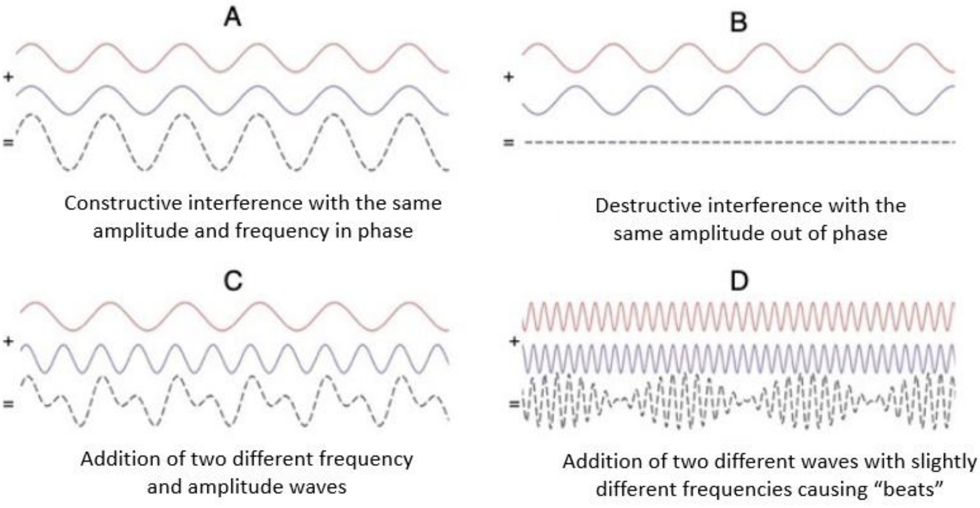


Figure A.5 Examples of constructive and destructive interference waves (source: University of Waikato Te Whare Wananga o Waikato)

A.3 Standing wave type resonances

Standing waves are formed from two or more traveling waves (incident and reflecting) that collide and are “in phase” with each other in such a way that their amplitudes add (called constructive interference) or subtract (called destructive interference) in repetitive ways which is called “resonance”. The locations at which the absolute value of the amplitude is minimum are called nodes, and the locations where the absolute value of the amplitude is maximum are called antinodes. These standing wave patterns are only possible at specific wavelengths and are determined with the following equation:

$$\lambda = \frac{c}{f} \tag{A-2}$$

In which:

- λ pulsation wave length [m]
- c speed of sound [m/s]
- f frequency [Hz]

For a pipe segment which has a strong reflection at both ends, different standing wave type resonances can occur for a length (L) between both ends which are summarised in Table A.1 where n is an integer value of 1,2, 3, … n.

Table A.1 Summary of length (L) for which a standing wave type can occur for different boundary conditions at both ends

Type of standing wave	**Length L**	**Figure number**
Open-open with a pressure node at both pipe ends	$L = \frac{n\lambda}{2}$	Figure A.6
Closed-closed with a pressure anti-node at both pipe ends	$L = \frac{n\lambda}{2}$	Figure A.6
Open-closed with a pressure node at the open end and a pressure anti-node at closed end	$L = \frac{(2n+1)\lambda}{4}$	Figure A.7

The pressure and velocity nodes and anti-nodes are clear to be seen Figure A.6 and Figure A.7. For flow meters of which the flow meter errors are a function of the *velocity* pulsations, the meter shall be located on the *velocity* nodes.

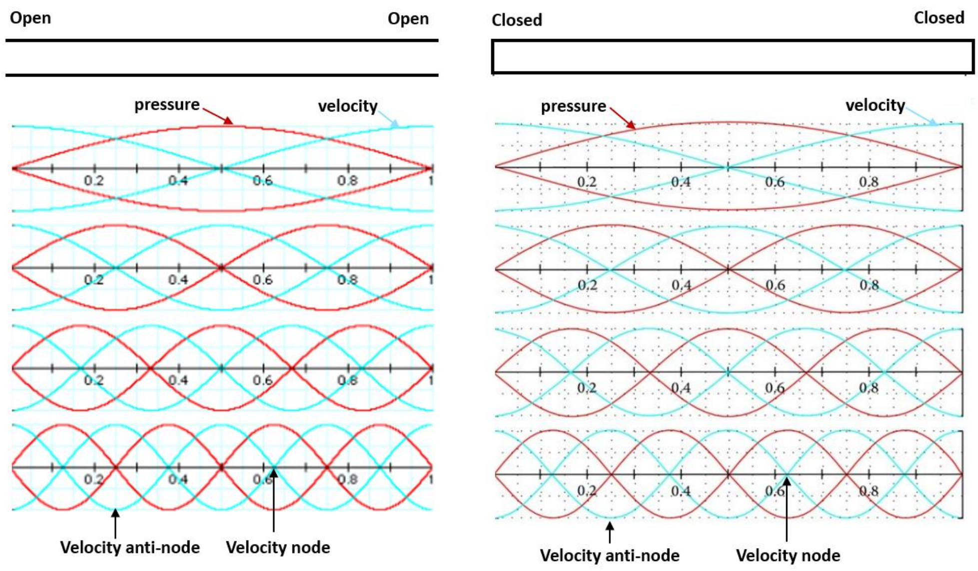


Figure A.6 Open-open pipe (left) and closed-closed pipe (right) with different standing wave type patterns for the pressure and flow (source: UNSW Sydney)

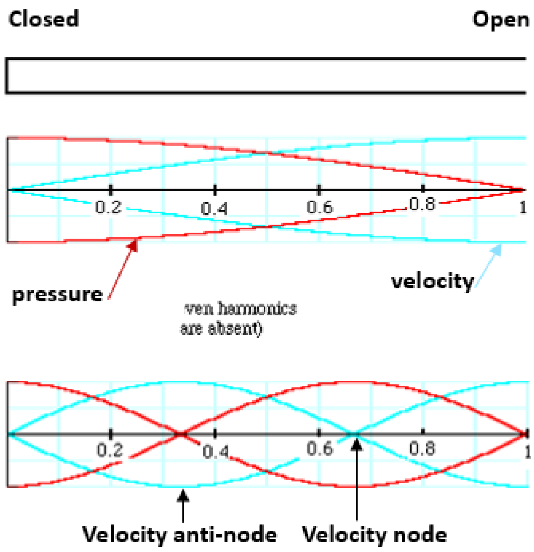


Figure A.7 Closed-open pipe with different standing wave type patterns for the pressure and flow (source: UNSW Sydney)

A.4 Helmholtz type resonances

A Helmholtz resonance is the phenomenon of gas resonance in a cavity, such as when one blows across the top of an empty bottle. The name comes from a device created in the 1850s by Hermann von Helmholtz, the *Helmholtz resonator*, which he used to identify the various frequencies or musical pitches present in music and other complex sounds.

When gas from the pipe system is forced into the cavity, the pressure inside increases. When the external force pushing the gas into the cavity is removed, the higher-pressure gas from the cavity inside will flow out. Due to the inertia of the moving gas the cavity will be left at a pressure slightly lower than the outside, causing gas to be drawn back in. This process repeats, with the magnitude of the pressure oscillations. The Helmholtz resonance behaves like a mass spring system of which the mass is the mass of gas inside the pipe system and the spring is the pressure of the gas inside the cavity. A picture of a one-and two chamber Helmholtz resonance is shown in Figure A.8. If one of the volumes of a two-chamber Helmholtz resonance is much larger than the other volume, the system can be considered as a one-chamber Helmholtz resonator.

A typical two-chamber Helmholtz resonance is e.g. a system which consists of a pipe section with two volumes, e.g. separator and pulsation damper or an interstage pipe system consisting of the two pulsation dampers and the piping.
A Helmholtz type resonator can also be designed to act as an acoustic filter. The effect of an acoustic filter is that pulsations with a frequency higher than the Helmholtz frequency will be attenuated.

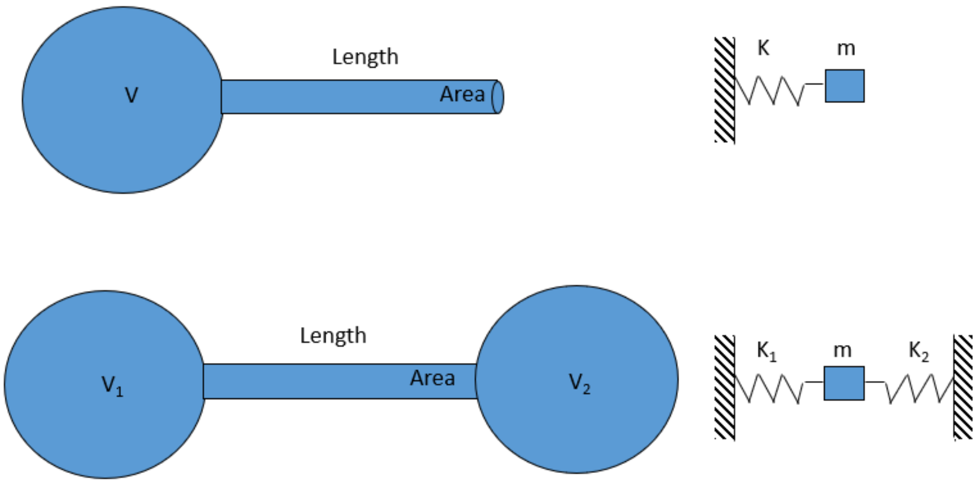


Figure A.8 One chamber (above) and two chamber (below) Helmholtz resonance system

The Helmholtz frequencies for a one-chamber and two-chamber Helmholtz resonance frequency are given by respectively equation (A-3) and equation (A-4).

$$f_H = c/2\pi\sqrt{\frac{A}{LV}} \qquad \text{(A-3)}$$

In which:

A	pipe area [m^2]
L	length of pipe [m]
V	cavity volume [m^3]
c	speed of sound [m/s]
f_H	Helmholtz resonance frequency [Hz]

$$f_H = c/2\pi\sqrt{\frac{A}{LV}\left(\frac{1}{V_1}+\frac{1}{V_2}\right)} \qquad \text{(A-4)}$$

In which:

V_1	cavity of volume 1 [m^3]
V_2	cavity of volume 1 [m^3]

The pressure and velocity nodes and anti-nodes of a Helmholtz resonance are shown in Figure A.9. The *velocity* pulsation is constant over the length between the two volumes. That means that if the flow meter error is a function of the *velocity* pulsations, an optimum location to reduce the error is not possible. In that case other methods shall be applied to reduce the flow meter error, e.g. by damping the resonance or by shifting the resonance frequency, preferably below the compressor speed.

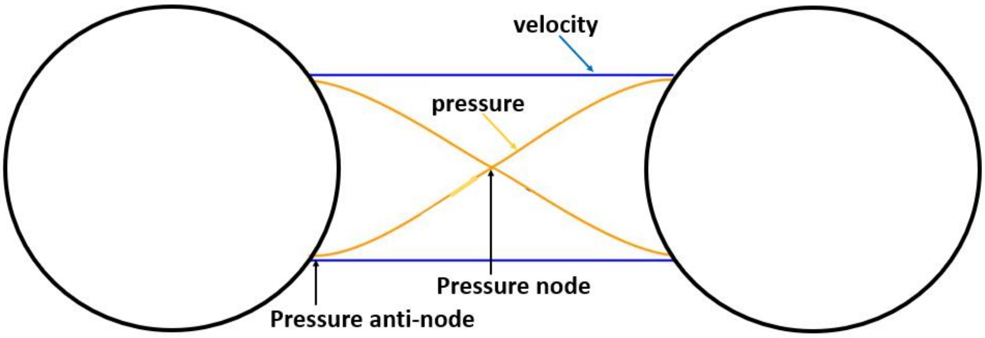
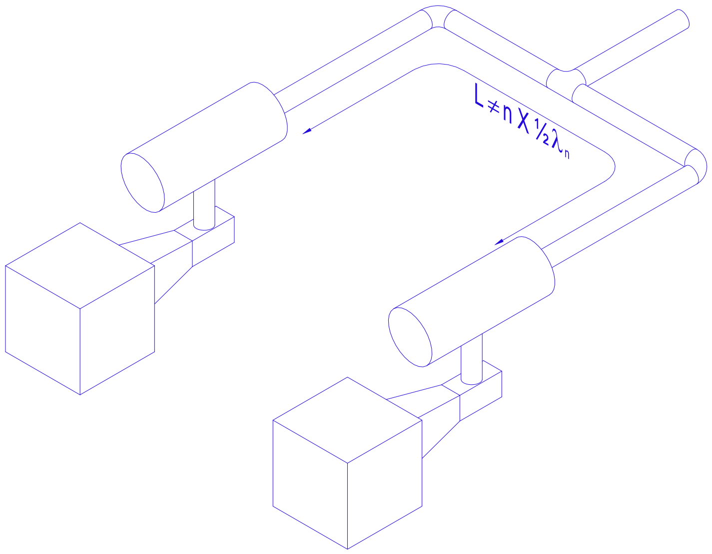


Figure A.9 Pressure and velocity nodes and anti-nodes of a Helmholtz resonance

A.5 Do’s don’ts and engineering rules

A.5.1 Avoid standing waves and Helmholtz resonance between dampers



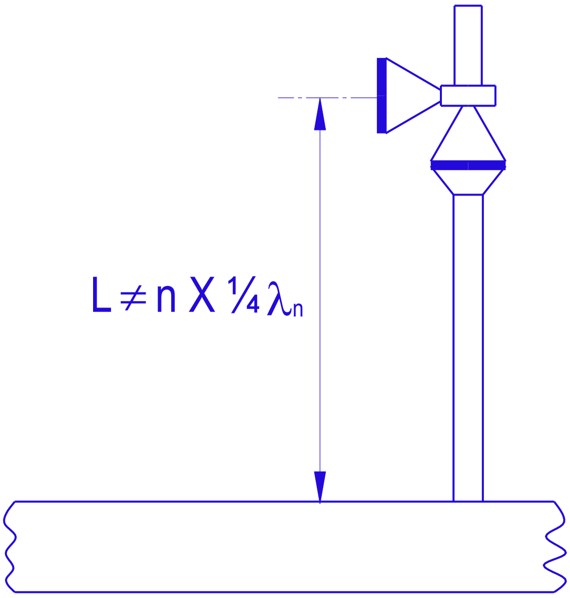
Standing waves

$$L \neq \frac{n\lambda}{2}$$ (see also equation (A-2))

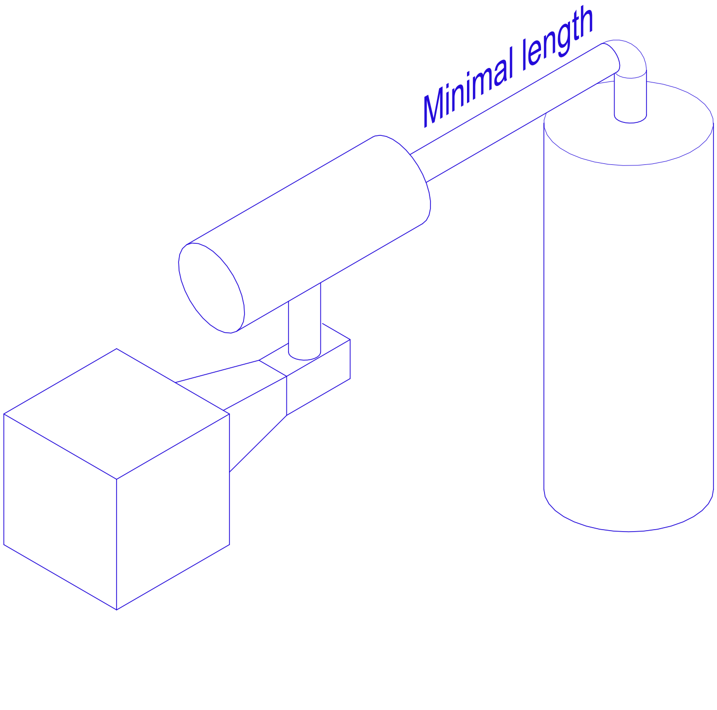
Helmholtz resonances:

$$f_H = c/2\pi\sqrt{\frac{A}{LV}\left(\frac{1}{V_1}+\frac{1}{V_2}\right)}$$ (see also equation (A-4))

A.5.2 Avoid standing wave resonance in closed side branches:



A.5.3 Install large volumes as close as possible to the damper:



B Illustrations for some practical examples

B.1 Introduction

As an illustration, some practical example cases are given, that show the main features of the acoustic theory of Appendix A described. The cases highlight specific aspects, relevant for the errors of flow meters that may be installed in the system.

B.2 Reciprocating compressor system

A typical reciprocating compressor system is shown below in Figure B.1. A single double-acting compressor cylinder is included (running at 100% and 50% load), with a suction and discharge pulsation damper. The compressor speed is 480 rpm. At the suction side, a KO drum is included; flow is taken from a large suction header. At the downstream side, the compressor discharges into a large outlet header. The medium is natural gas (MW=17), pressures are 10 bar (suction) and 25 bar (discharge).

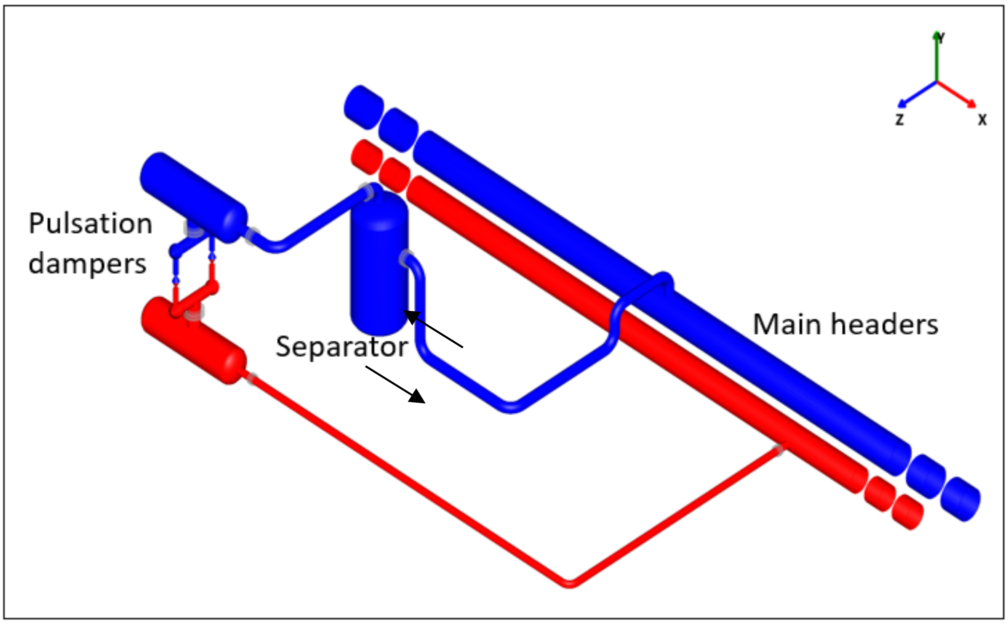


Figure B.1 Typical reciprocating compressor system (1 cylinder). Suction system (blue) and discharge system (red).

B.2.1 Pulsation levels in the system and overall effect of restriction orifice plates

The compressor cylinder delivers an unsteady (pulsating) flow. The pulsation dampers supress the pulsation amplitude to the attached pipe system, in order to prevent various negative consequences, such as vibrations, fatigue failure, erroneous disturbance of check valves/pressure safety valves and flow metering errors.

The maximum *pressure* pulsation amplitudes (compared with the API 618 limit) are shown in Figure B.2. Levels exceeding limits are found on both sides of the compressor, caused by various acoustic resonances. The occurrence and amplitude of the acoustic resonances depend on the load conditions of the compressor, as will be illustrated further below. A common strategy to reduce acoustic resonance effects are restriction orifice plates. The beneficial effect on the *pressure* pulsation amplitudes is shown in Figure B.3. It is obvious that an overall reduction in *pressure* pulsation amplitudes will also be beneficial for the flow meter accuracy. The system with orifice plates is compliant with the pulsation limits in API 618, but the key question is: does the optimized layout also result in small velocity pulsation levels and acceptable flow metering errors?

As a general recommendation, in principle the system part with the lowest *velocity* pulsation amplitude is most suited for placing the flow meter which is explained in detail in the next section.

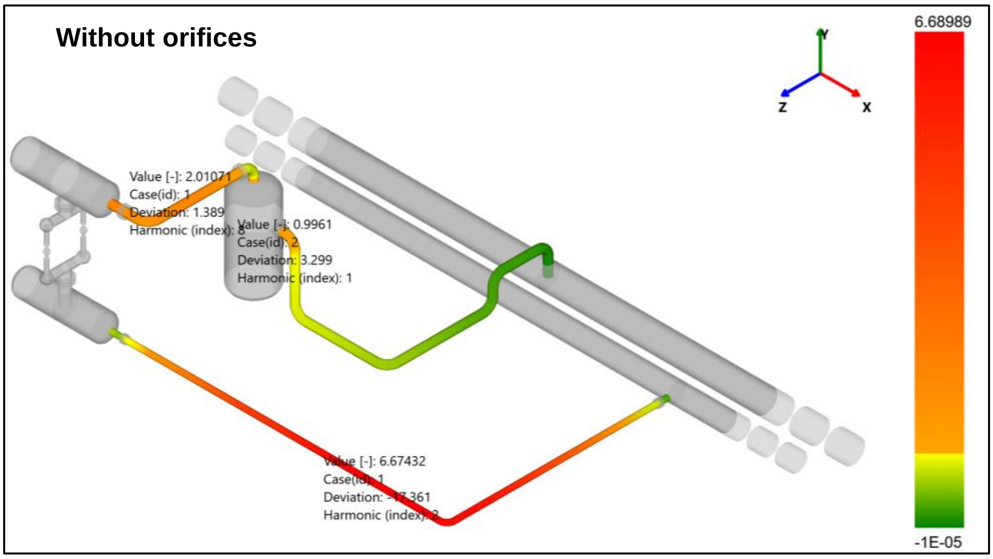


Figure B.2 Maximum *pressure* pulsation levels compared to API 618 limits. Layout without orifice plates.

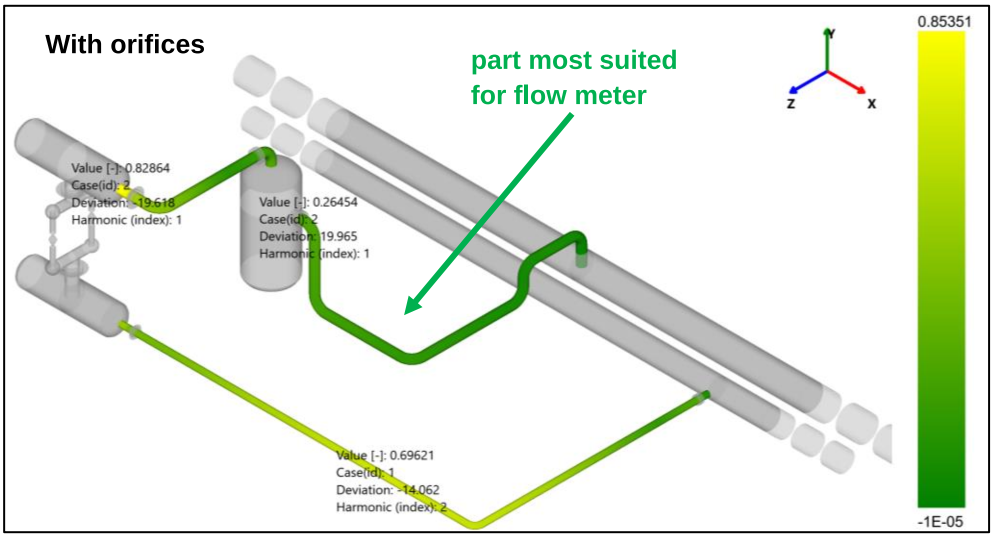


Figure B.3 Maximum *pressure* pulsation levels compared to API 618 limits. Layout with orifice plates.

B.3 Acoustic resonances

The high-pressure pulsation levels (in the original layout without orifices) are caused by acoustic resonances. In case of an acoustic resonance, the amplitude of *pressure* and *velocity* pulsations may be strongly amplified, depending on the acoustic damping in the system. The actual pulsation amplitudes will be strongly dependent on the location in the pipe system. On some locations, the *pressure* amplitude is minimal (node) and at other locations the amplitude is maximal (anti-node). The maxima and minima of *pressure* and *velocity* pulsations do generally not coincide.

For the example case, at the discharge side, the acoustic resonance is clearly a standing wave between the discharge damper and the discharge header. The strongest effect is found at 100% load, dominated by the 2nd order of the compressor speed. The pulsation amplitude for *pressure* and *velocity* is shown in Figure B.4 and Figure B.5 respectively. Near the elbow, the *pressure* pulsation amplitude is maximal (anti-node), while the *velocity* pulsation amplitude is minimal (node). Near the discharge damper and the large outlet header, the opposite is true: the *pressure* pulsation amplitude is minimal, while the *velocity* pulsation amplitude is maximal. For this specific, dominant resonance, the optimal location for a flow meter would be near the elbow. Note that other acoustic resonances may be present as well; for those the location near the elbow may be less favourable and the results for all cases will finally determine the optimal location

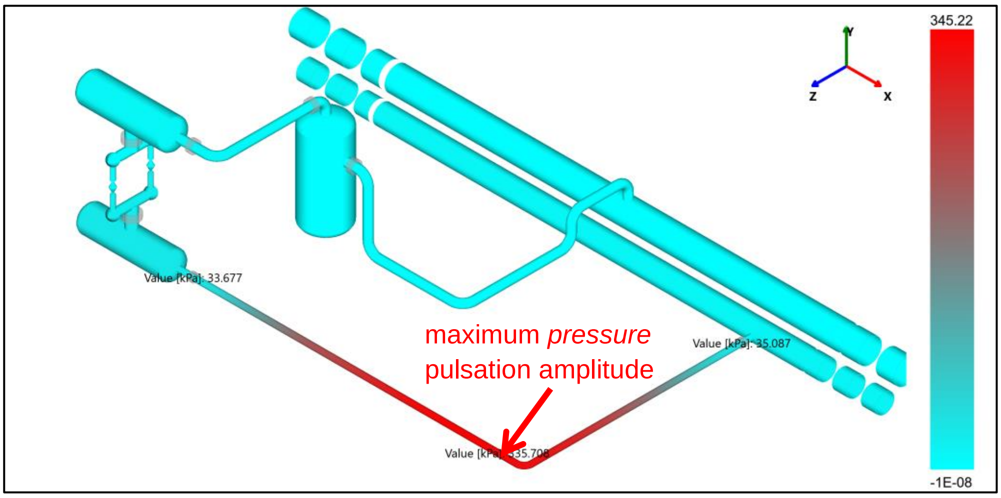


Figure B.4 *Pressure* pulsation amplitude, in acoustic resonance condition (standing wave).

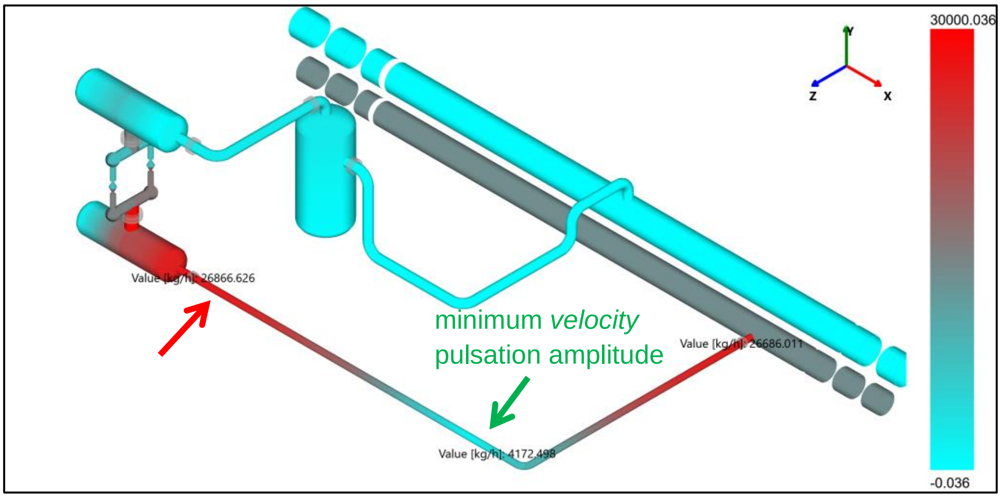


Figure B.5 *Flow* pulsation amplitude, in acoustic resonance condition (standing wave).

At the suction side, at 100% load, a standing wave resonance is found between the KO drum and the suction damper. In this case, the resonance occurs at a higher frequency (8^{th} order) while the *pressure* pulsation amplitude is lower. The pulsation amplitude for *pressure* and *velocity* is shown in Figure B.6 and Figure B.7 respectively. Also, in this case, the maxima in *pressure* and *velocity* pulsation amplitude do not match. The *pressure* amplitude is maximal between the two vessels, while the *velocity* pulsations are minimal. For this acoustic resonance, the optimal location of a flow meter would be in the middle of the section.

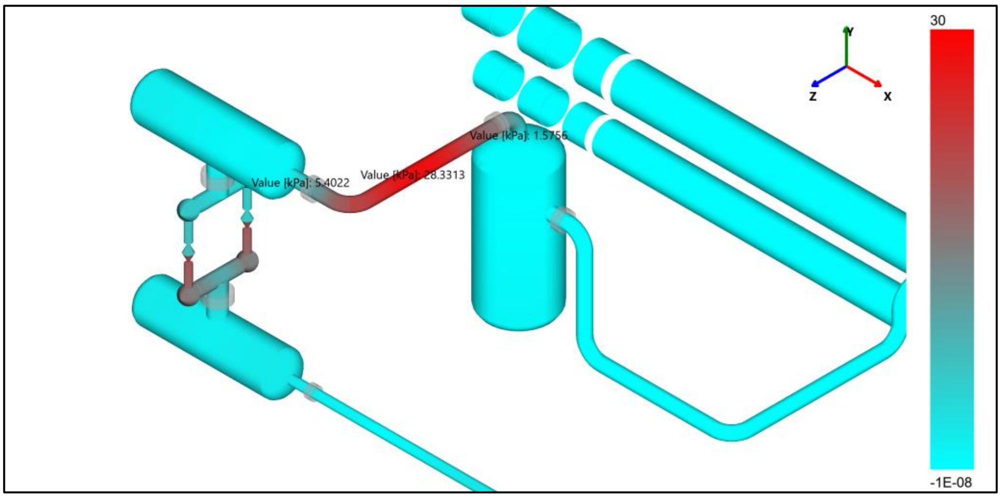


Figure B.6 *Pressure* pulsation amplitude, in acoustic resonance condition (standing wave).

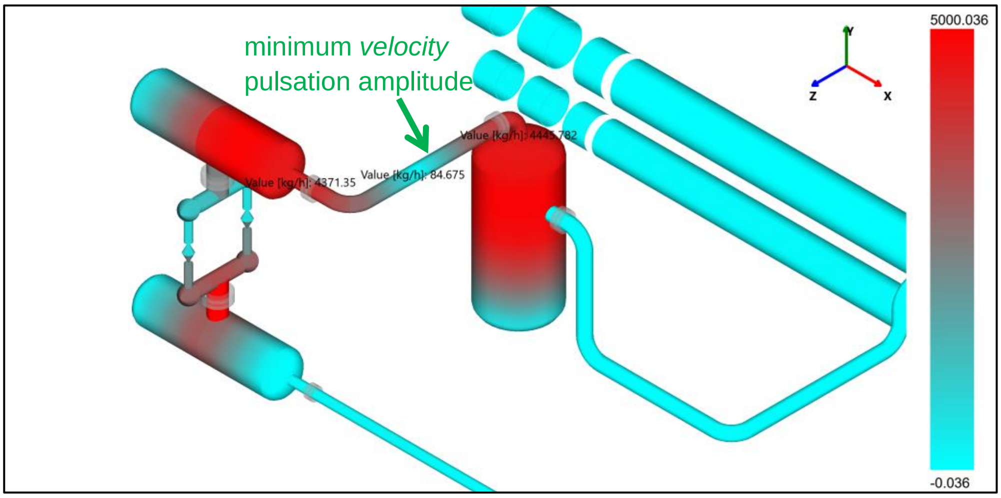


Figure B.7 *Velocity* pulsation amplitude, in acoustic resonance condition (standing wave).

Finally, another type of acoustic resonance occurs at 50% load. This is a Helmholtz resonance that occurs between the volumes of the KO drum and the pulsation damper. The pulsation amplitude for *pressure* and *velocity* is shown in Figure B.8 and Figure B.9 respectively. In this case, the *velocity* pulsation amplitude (the main cause for flow meter errors) is constant over the connecting pipe. The *pressure* pulsation amplitude is maximal near the volumes and minimal half-way the interconnecting pipe. Because the *velocity* pulsation amplitude is constant, the <u>location</u> for the flow meter is not critical (similar metering error expected for all locations on the interconnection pipe). However, in absence of restriction orifice plates, the *velocity* pulsation *amplitude* is very severe, leading to flow reversal and a gross error of the flow meter.

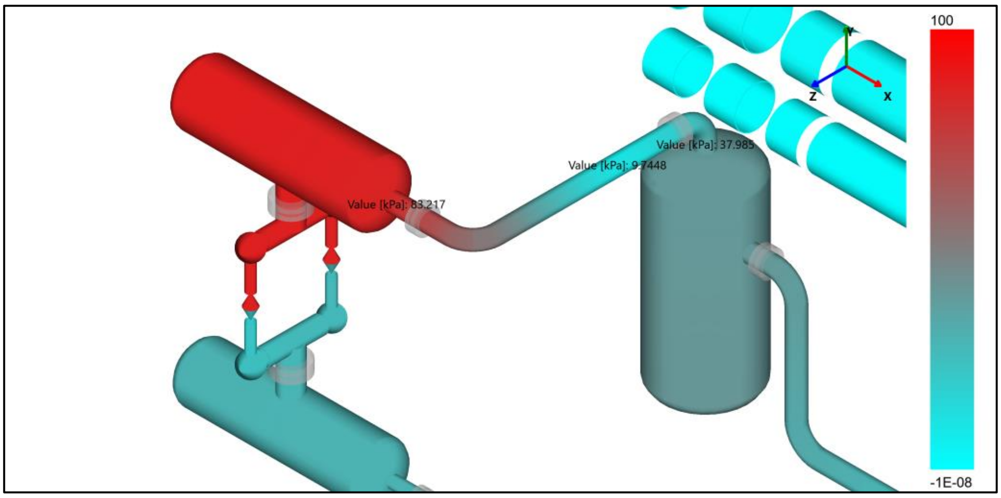


Figure B.8 *Pressure* pulsation amplitude, in acoustic resonance condition (Helmholtz resonance).

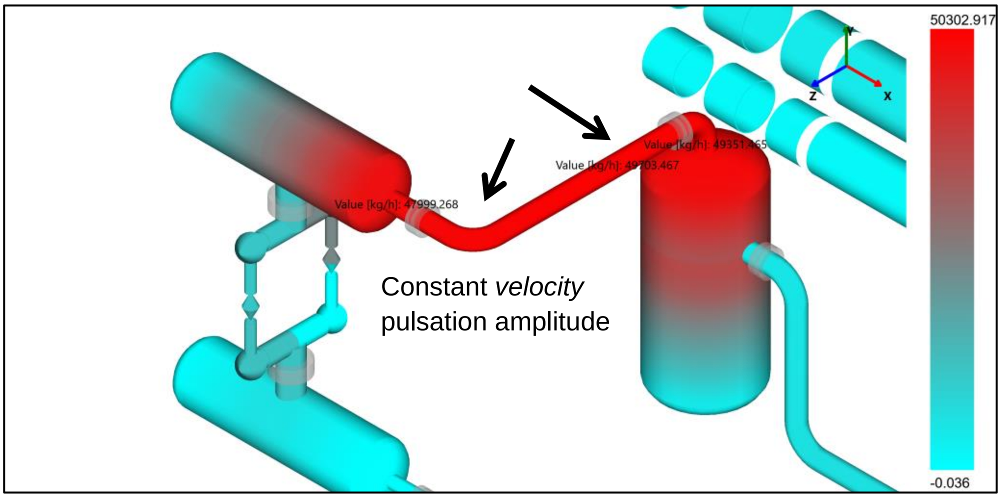


Figure B.9 *Velocity* pulsation amplitude, in acoustic resonance condition (Helmholtz resonance).

B.4 Velocity pulsation levels in optimized layout

It was illustrated before that restriction orifices have a beneficial effect in suppressing pulsation amplitudes in acoustic resonance conditions. In the example above, with restriction orifices, the *pressure* pulsation limits in API 618 are met. Nevertheless, the residual *velocity* pulsation levels may be still very substantial, as will be illustrated below.

As the first example, consider the acoustic resonance at the discharge side (100% load, 2^{nd} order). *Velocity* pulsation amplitudes, with orifice plates, are shown in Figure B.10. The relative *velocity* pulsation levels are derived from the time signals (peak-peak values). Two signals are shown, one at the immediate outlet of the damper (high *velocity* pulsations, Figure B.10) and one near the elbow (low *velocity* pulsations, see Figure B.10). As a simplified approach, the RMS values are derived from the peak-peak values, assuming a single dominant frequency component. At the worst location (damper outlet) $U'_{RMS}/\bar{U}$ = 13%.

At the best location (elbow), $U'_{RMS}/\bar{U}$ = 5%. Note that these values are well above the limits proposed in ISO TR 3133 and API 618, see also section 4.3 and 4.4.

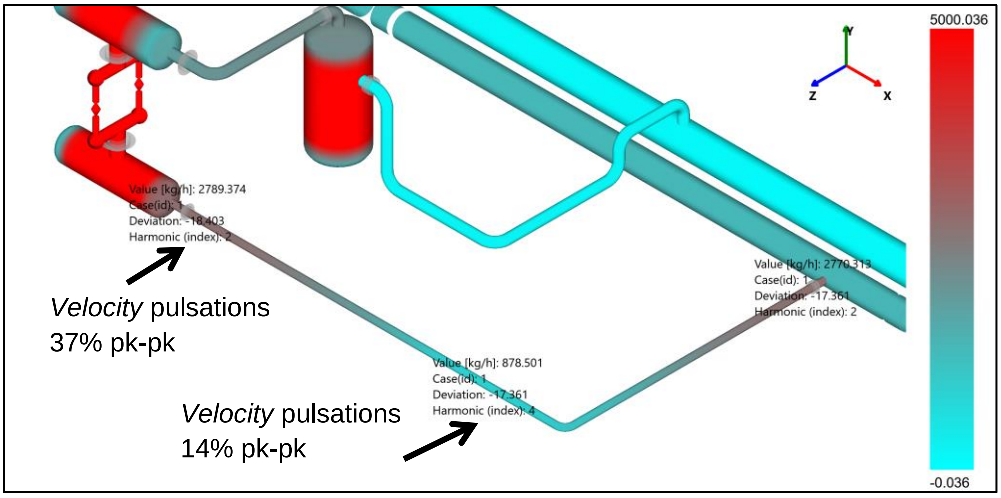


Figure B.10 *Velocity* pulsation amplitudes (peak-peak), optimized layout with orifice plates.

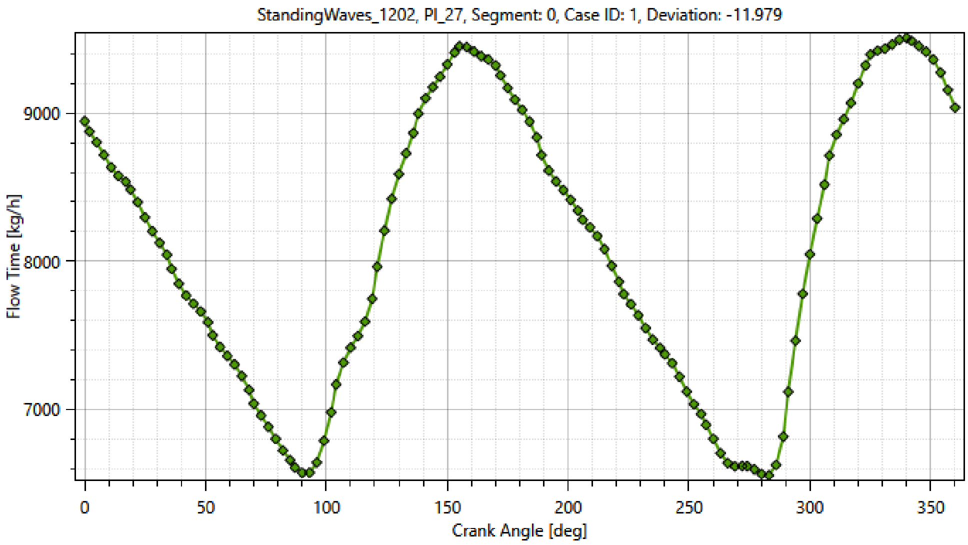


Figure B.11 Flow time signal at outlet of pulsation damper: high flow pulsation level, at 2nd order.

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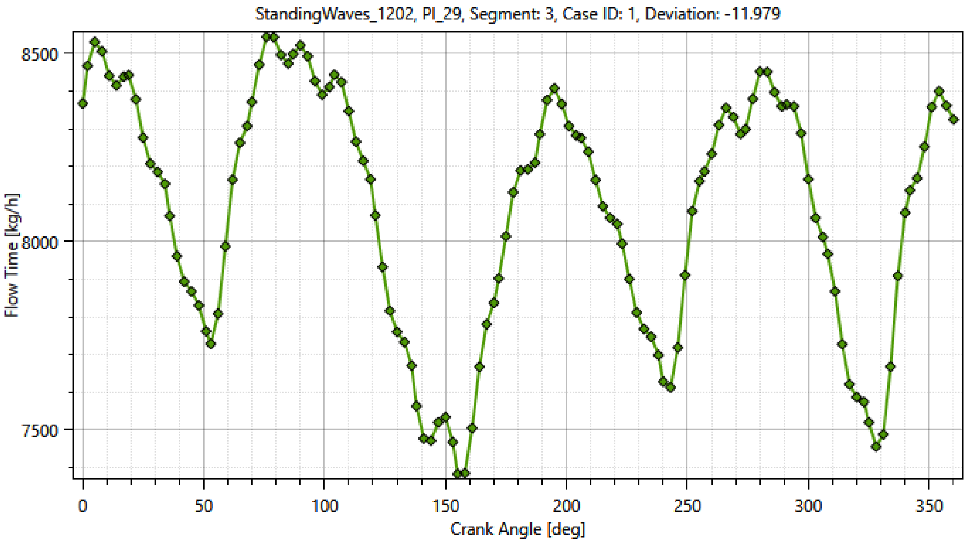

Figure B.12 *Velocity* time signal near elbow: lower pulsation level, dominated by 4th order

As a second example, consider the Helmholtz resonance between the KO drum and the suction damper (50% load, 1st order). *Velocity* pulsation amplitudes with orifices are shown in Figure B.13. The relative *velocity* pulsation levels are derived from the time signals (peak-peak values). Again, the RMS value is derived from the peak-peak value, assuming a single dominant frequency component. The relative *velocity* pulsation amplitude is very high: $U'_{RMS}/\bar{U}$ = 300%, even resulting in flow reversal as shown in Figure B.14. These values are far above the limits proposed in ISO TR 3133 and API 618 and will lead to gross flow meter errors.
Note that these high levels are a consequence of the strong resonance, the low mean flow (part load condition of the cylinder) and the limited effect of the orifice plates at low flow. Even though the effects of the unsteady flow are very significant, the relative *pressure* fluctuation is within limits of the API 618 standard. The shaking forces on the pipe sections are relatively small and, possibly, no vibration issues are experienced at all. The fact that this part of the system is totally unsuited for flow measurement, may easily be overlooked in practice!

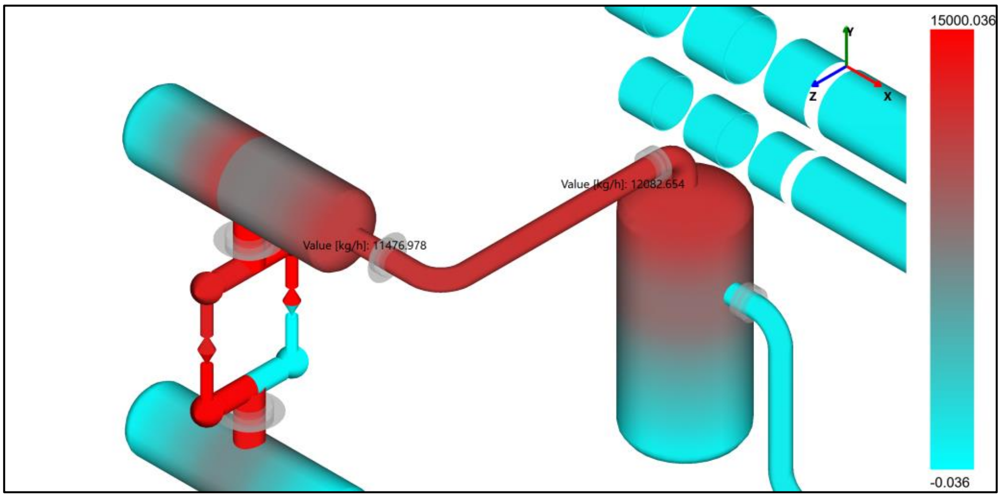

Figure B.13 *Velocity* pulsation amplitudes, optimized layout with orifice plates

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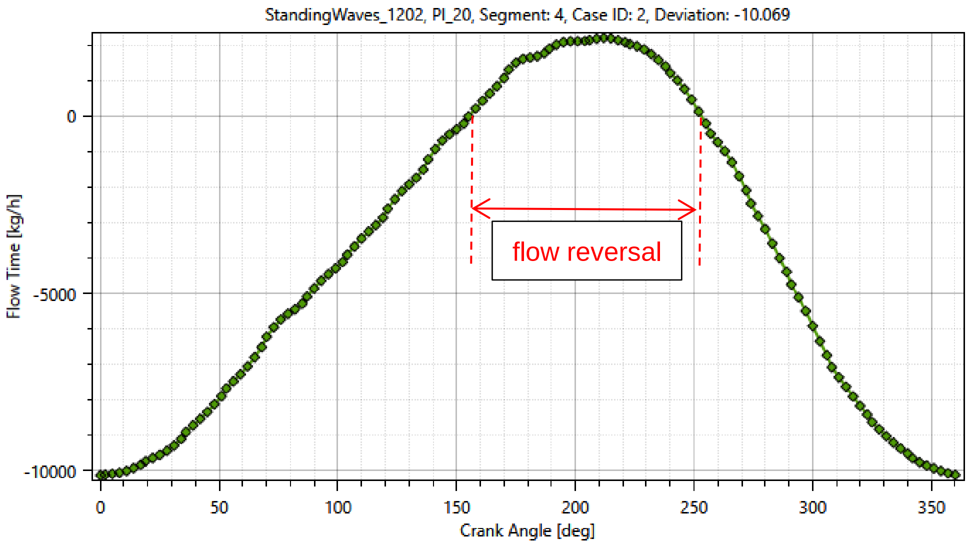


Figure B.14 *Velocity* pulsation time signal, dominated by 1st order (note that for this location, the mean flow is defined as negative).

B.5 More complex system layout

Even though the quantification of the *velocity* pulsation levels requires a simulation tool, the overall conclusions and observations for *velocity* pulsations may look straightforward for the example above. Note however, that for increasing complexity, the intuition for *velocity* pulsation amplitudes quickly reduces. This is illustrated in the model below (see Figure B.15), that has an extra relief line on the discharge side. The *pressure* pulsation result is rather similar to the previous example without the relief line (see Figure B.3), displaying the standing wave at the 2nd order of the compressor speed. However, the *velocity* pulsation amplitude has a striking feature, up- and downstream of the relief line connection.
Upstream of the relief line, the *velocity* pulsation amplitudes are much larger ($U'_{RMS}/\bar{U}$ =15%) than downstream of the connection ($U'_{RMS}/\bar{U}$ = 5%). Obviously, for a flow meter, the downstream location would be more favourable. For most of the realistic, complex system layouts, a numerical simulation model is a mandatory tool for an adequate analysis of the optimal flow meter location.

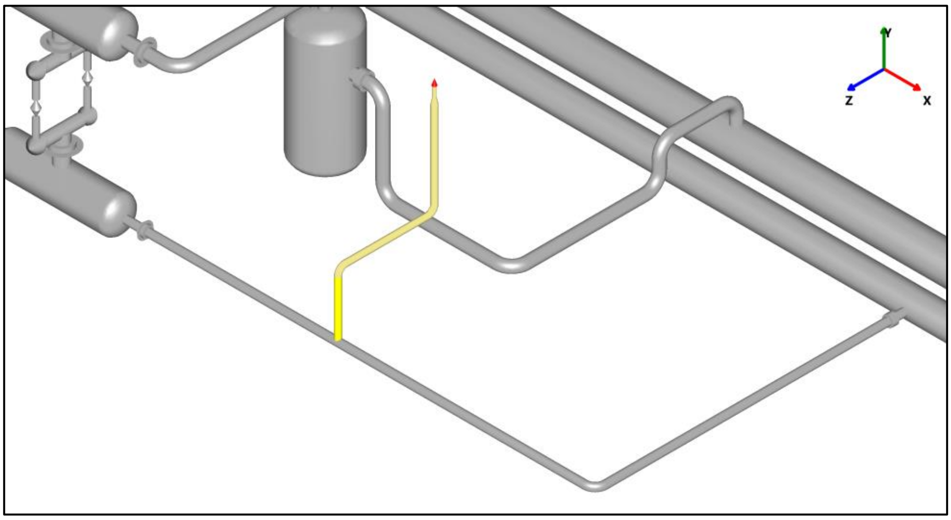


Figure B.15 Update layout, with pressure relief line connected to discharge side.

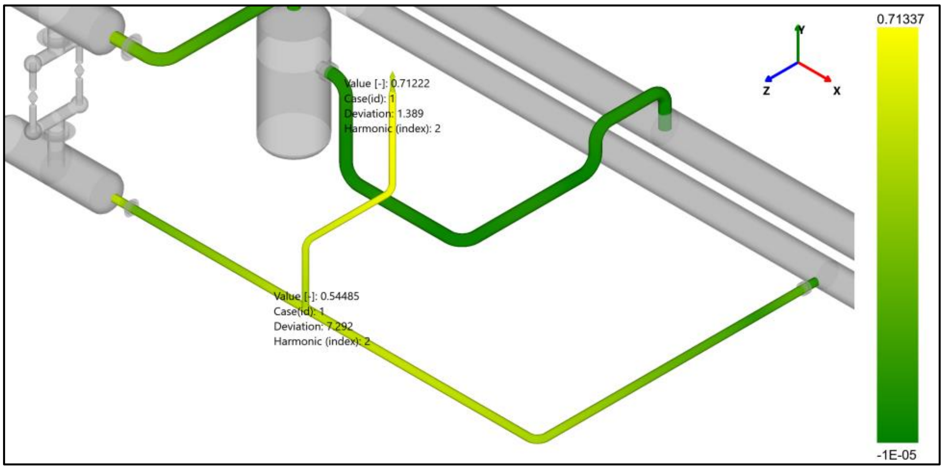


Figure B.16 *Pressure* pulsation levels compared to API 618.

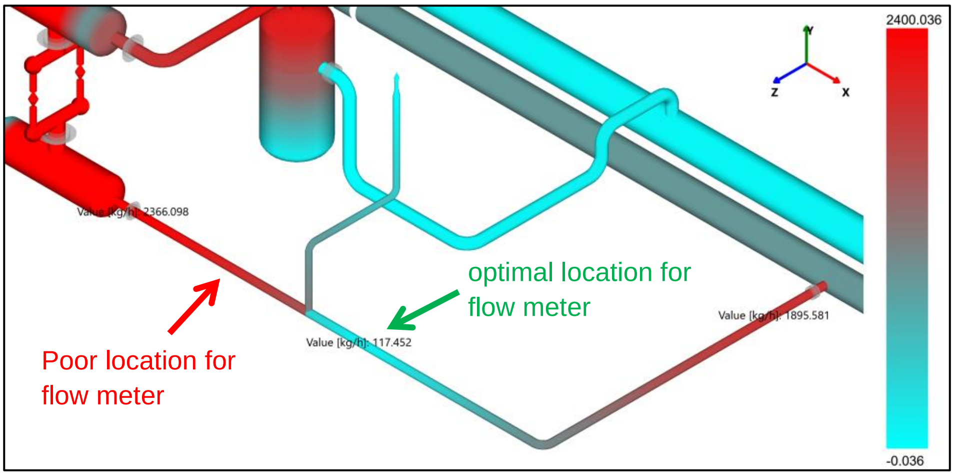


Figure B.17 *Velocity* pulsation amplitude, in acoustic resonance condition (standing wave).

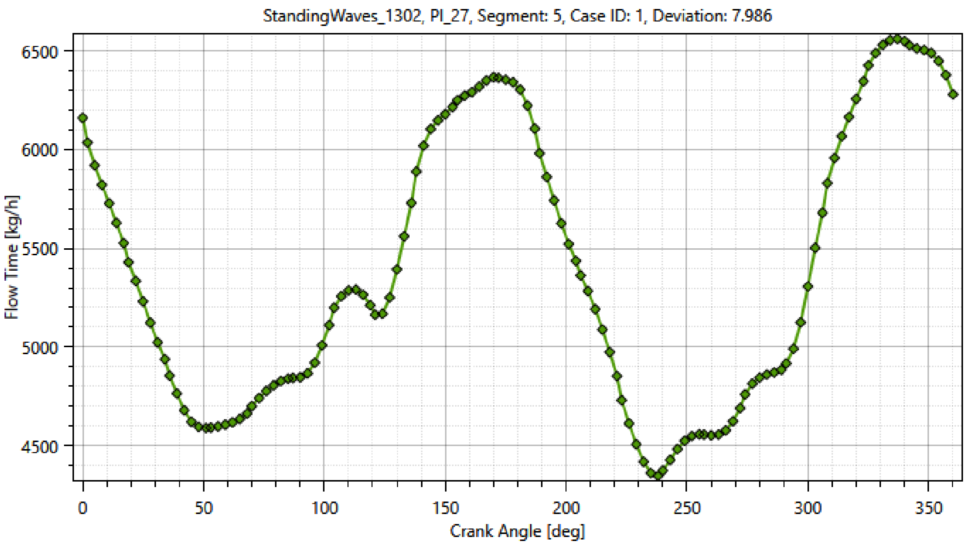


Figure B.18 Velocity pulsation time signal upstream of relief line: high *velocity* pulsation level, at 2nd order.

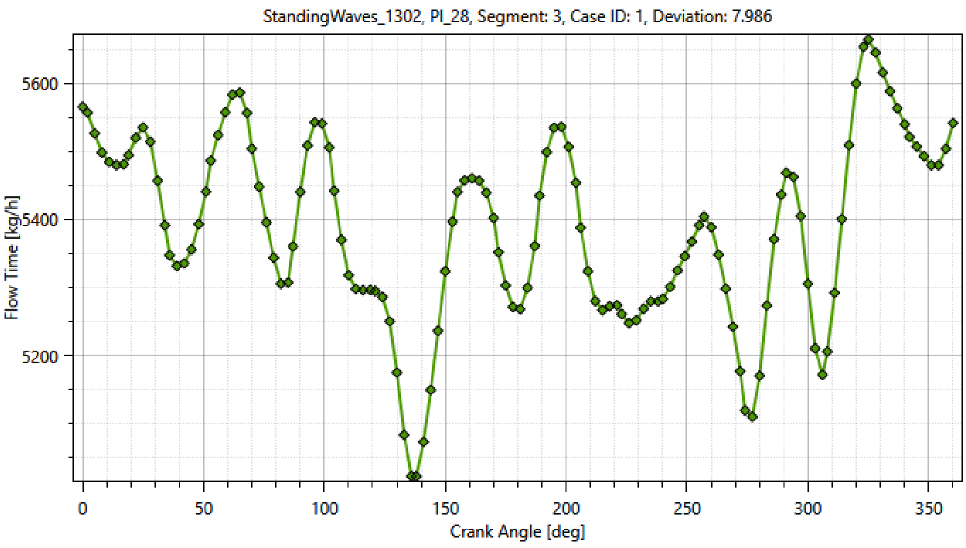


Figure B.19 Velocity pulsation time signal downstream of relief line: lower *velocity* pulsation level, at 2nd order.

B.6 Conclusions

Based on the examples shown above, the following conclusions are drawn:

- In general, parts of a system displaying consistently low *velocity* pulsation amplitudes, are more suited for installation of a flow meter, than parts with high *velocity* pulsation amplitudes. This may apply, for example, upstream of a KO drum.
- In case of acoustic resonance conditions, the pulsation (both velocity and pressure) amplitudes may be significantly increased. The exact locations of *velocity* pulsation maxima and minima depends on the nature of the acoustic resonance.
- In general, locations with maximum pulsation amplitudes for *pressure* and *flow* do not coincide. For the accuracy of flow meters, the relative *velocity* pulsation amplitude is the crucial parameter.
- To suppress the pulsation (both velocity and pressure) amplitudes in acoustic resonance conditions, restriction orifice plates may be very effective.
- Even if *pressure* pulsations are relatively low (consistent with limits in API 618), *velocity* pulsation amplitudes may still be very substantial.
- In particular, this situation occurs at low capacity, part load, conditions and/or minimum compressor speed. Then, the mean flow is small, and the effect of restriction orifices plates is limited.
- Analysis of *velocity* pulsation amplitudes for realistic compressor systems requires a detailed simulation model. A detailed simulation model enables the user to determine the optimal locations for a flow meter. For differential pressure and turbine flow meters, the simulation results also enable a quantification of the flow metering error (based on guidelines in ISO TR 3313 and/or vendor specifications as shown in section 6.2.1 and 7.2.

C Calculation of response parameter of a turbine meter

The response parameter of a turbine meter can be calculated in the following ways:

1. For short *rectangular* blades according Grey [citation]:

$$b = \frac{2I(1 + 2\eta/A_R)}{N\rho\eta LR} \tag{C.1}$$

In which:

b	rotor response parameter [m^3]
I	moment of inertia of complete rotor [kgm^2]
A_R	blade aspect ratio (ratio between blade length and rotor radius [-]
N	number of blades [-]
ρ	density of the rotor material [kg/m^3]
η	blade "air foil efficiency", Grey suggest using a value of 0.9 [-]
L	length (axial) of the blades [m]
R	mean radius of the blades [m]

2. For short *helical* blades according Jepson [11]:

$$b = \frac{2I(1 + 2\eta/A_R)R(r_{t-}\, r_h)}{N\rho\eta \int_{r_h}^{r_t} Lr^2 dr} \tag{C.2}$$

In which:

r	blade radius [m]
r_t	blade tip radius [m], see Fig C.1
r_h	blade hub radius [m], see Fig C.1

Moments of inertia of blades only

If the inertia of the rotor hub is negligible compared with that of the blades it can be calculated as follows, considering that the thickness and chord of the blade are independent of the radius:

$$I = N\rho_b t_b C({r_t}^3 - {r_h}^3)/3 \tag{C.3}$$

In which:

I	moment of inertia of blades [kgm^2]
N	number of blades [-]
ρ_b	density of the blade material [kg/m^3]
t_b	blade thickness [m]
r_t	blade tip radius [m]
r_h	blade hub radius [m]
C	blade chord [m]

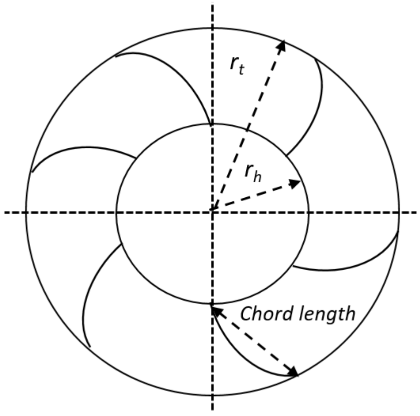
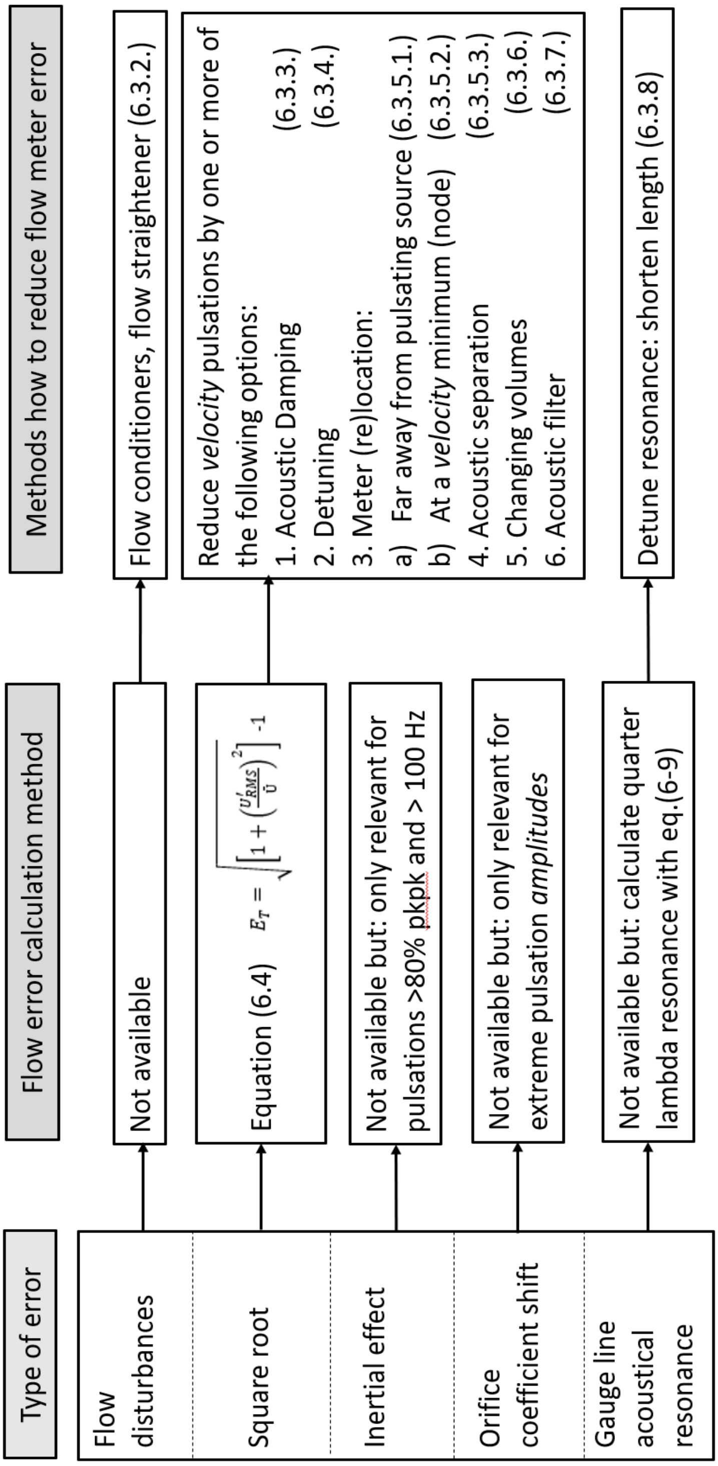


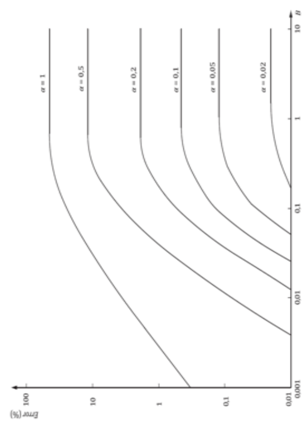
Figure C.1 Definitions of blade hub (r_h), and blade tip (r_t) radius and chord length

D Summary of flow meter errors, calculation methods and how to reduce the errors for different meters

D.1 Differential pressure meter, see chapter 6



D.2 Turbine flow meter, see chapter 7

Type of flow error	Flow error calculation method	Methods how to reduce flow meter error
Flow disturbances	Not available	Flow conditioners, flow straighteners (7.3.4.)
Rotor and fluid Inertia (7.2.)	Equation (7.1- 7.5) to calculate α and B and then read error from figure 7.2 	Change rotor response parameters (7.3.3.)
		Reduce *velocity* pulsations by one or more of the following options 1. Acoustic Damping (6.3.3.) 2. Detuning (6.3.4.) 3. Meter (re)location: a) Far away from pulsating source (6.3.5.1.) b) At a *velocity* minimum (node) (6.3.5.2.) 4. Acoustic separation (6.3.5.3.) 5. Changing volumes (6.3.6.) 6. Acoustic filter (6.3.7.)

D.3 Vortex, see chapter 8

Type of flow error	Flow error calculation method	Methods how to reduce flow meter error
Flow disturbances	Not available (8.3.1.)	Flow conditioners, flow straighteners (7.3.4.)
Lock-in (8.2.1.)	a) Calculate RMS ratio of *velocity* pulsation and mean flow ($\frac{U'_{RMS}}{\bar{U}}$). If this ratio is < 0.03, lock-in will not occur and an *amplitude* reduction of *velocity* pulsations is not required. For ratios >0.03, an amplitude reduction is required (see right column) (8.3.2.) b) Calculate ratio of pulsation frequency and vortex frequency (f_p/f_v). If this ratio is <0.25 or > 2, lock-in will occur and a shift in f_p is required such that: (f_p/f_v) < 0.2, or (f_p/f_v) >3.6 (8.3.2.)	Reduce *velocity* pulsations by one or more of the following options: 1. Acoustic Damping (6.3.3.) (will not work for option b) from left column) 2. Detuning (6.3.4.) 3. Meter (re)location: a) Far away from pulsating source (6.3.5.1.) b) At a *velocity* minimum (node) (6.3.5.2.) 4. Acoustic separation (6.3.5.3.) 5. Changing volumes (6.3.6.) 6. Acoustic filter (6.3.7.)
External vibrations (8.2.2)	a) Calculate ratio of pipe vibration frequency and vortex frequency (f_{vpipe}/f_v). This value shall be *< 0.2 or >3.6* (8.3.4.) *OR:* b) Calculate vibration levels which shall be: <3.2-6.4 mm/s RMS (8.3.4.)	Case a) Detuning the system by changing the stiffness of the supporting structure or pipe support layout (8.3.4.) Case b) Stiffening the supporting structure of the meter is very efficient (8.3.4.)

D.4 Coriolis meter, see chapter 9

Type of error	Flow error calculation method	Methods how to reduce flow meter error
Flow disturbances	Not available	Flow conditioners, flow straighteners (7.3.4.)
Lock-in of the frequency of *velocity* pulsations (f_p) with drive (f_1) and Coriolis (f_2) frequency (9.2.2.)	Not available but lock-in can be avoided if the frequency of the *velocity* pulsations (f_p) do not coincide with the drive (f_1) and Coriolis (f_2) frequency (8.3.2.) as follows: • (f_p/f_1) *and* (f_p/f_2) <0.8 *OR* • (f_p/f_1) *and* (f_p/f_2) >1.2	Detuning can be achieved by one or more of the following options : 1. Detuning pipe length (6.3.4.) 2. Meter (re)location: a) Far away from pulsating source (6.3.5.1.) b) At a *velocity* minimum (node) (6.3.5.2.) 3. Acoustic separation (6.3.5.3.) 4. Changing volumes (6.3.6.) 5. Acoustic filter (6.3.7.)
External vibrations (9.2.3)	a) Calculate ratio of pipe vibration frequency and drive frequency (f_{vpipe}/f_1) *and* ratio of pipe vibration frequency and drive frequency (f_{vpipe}/f_2) which meet the following: • (f_{vpipe}/f_1) *and* (f_{vpipe}/f_2) <0.8 *OR* (9.3.6.) • (f_{vpipe}/f_1) *and* (f_{vpipe}/f_2) >1.2 *OR* b) Calculate vibration levels which shall be: <3.2-6.4 mm/s RMS (9.3.6.)	Case a) Detuning the system by changing the stiffness of the supporting structure or pipe support layout (9.3.6.) Case b) The most efficient way is to stiffen the supporting structure of the meter (9.3.6.)

D.5 Ultrasonic meter, see chapter 10

Type of error	Flow error calculation method
Flow disturbances (10.2.1)	Not available
Flow shape changes (10.2.2.1)	Not available
Distortion of meter Pulses (10.2.2.2)	Not available
Aliasing* (10.2.2.3)	Not available
Data averaging (10.2.2.3)	Not available

Methods how to reduce flow meter error

Flow conditioners, flow straightener (6.3.2.)

Reduce *velocity* pulsations by one or more of the following options:

1. Acoustic Damping (6.3.3.)
2. Detuning (6.3.4.)
3. Meter (re)location:
   a) Far away from pulsating source (6.3.5.1.)
   b) At a *velocity* minimum (node) (6.3.5.2.)
4. Acoustic separation (6.3.5.3.)
5. Changing volumes (6.3.6.)
6. Acoustic filter (6.3.7.)

*Some meter vendors are able to adjust the sampling frequency and for those meters the sampling frequency shall be minimum 2 times the highest expected pulsation frequency