EFRC Guidelines

EFRC

EUROPEAN FORUM
for RECIPROCATING
COMPRESSORS

Guidelines for Vibrations in Reciprocating Compressor Systems

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Foreword

The EFRC is the European Forum for Reciprocating Compressors and has been founded in 1999 by Neuman & Esser, Leobersdorfer Maschinenfabrik, Hoerbiger Ventilwerke, TNO Science & Industry, TU Dresden, Thomassen Compression Systems, Wärtsilä Compression Systems and Burckhardt Compression. The target of the EFRC is to serve as a platform to facilitate 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), standardization work (e.g. API 618, ISO 13707, ISO 10816) and by joint pre-competitive research projects, aiming at improving the performance and the image of the reciprocating compressor.

In the R&D projects the forces are combined of all interested parties to solve or investigate problems which are beyond the scope of a single player. The basic research and pre-competitive research projects are carried out at research institutes or universities. In this way, the R&D group of the EFRC will serve as the scientific arm of the reciprocating compressor community.
The R&D group is 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.

Vibrations are an important criterion to judge the safety, integrity and efficiency of compressor installations. For that purpose, several international standards have been developed. The existing standards are unspecific and do not make a distinction in vibration levels for different parts of the reciprocating compressor system, e.g. foundation, frame, cylinder, pulsation dampers and piping. For that reason, the "EFRC Guidelines for Vibrations in Reciprocating Compressor Systems" were developed.

The vibration levels from the guidelines are intended to be used during a field survey to judge the long-term safety, reliability and efficiency. The guidelines are not intended for condition monitoring purposes.

Disclaimer

The information in this report is brought to you only as guidance and while every reasonable care has been taken to ensure the accuracy of its contents, the EFRC cannot accept any responsibility for any action taken, or not taken, based on the provided information in this report. The EFRC shall not be responsible to any person for any loss or damage which may arise from the use of any of the information contained in any parts of this report.

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Introduction

This EFRC Guideline is a document which establishes procedures and guidelines for the measurement and classification of mechanical vibration of reciprocating compressor systems. The first version of the EFRC guidelines was released in November 2009.
There is a trend in the international industry to use more ISO standards because it is published documents that establish methods, specifications and procedures designed to ensure the reliability, integrity and safety. Standards consist consistent protocols that can be universally understood and adopted and make it easier to understand and compare competing products.
Standards are globally adopted and applied in many markets and it is only through the use of standards that requirements can be assured. For that reason, the ISO 10816-8 "Mechanical vibration - Evaluation of machine vibration by measurements on non-rotating parts - Part 8: Reciprocating compressor systems" was developed. The first version of the ISO 10816-8 was released in July 2014 and it contains most of the base material of the EFRC guidelines with several improvements and extensions.

Besides having an international standard, it was recognised that there is still a need for having (unconditional) guidelines. For that reason, the EFRC guidelines were updated and include many of the improvements and extensions of the ISO 10816-8. Besides that, the EFRC guidelines have several informative annexes which are not included in the ISO 10816-8 and are intended to serve the user in understanding how to carry out adequate and reliable vibration measurements.

The vibration levels from the EFRC Guidelines are defined primarily to classify the vibration of the compressor system and to avoid fatigue problems with parts in the reciprocating compressor system e.g. foundation, crankcase, cylinder, dampers, piping and auxiliary equipment mounted on the compressor system. The EFRC Guidelines are not intended for condition monitoring purposes.

Typical features of reciprocating compressors are the oscillating masses, the cyclically varying input torques and the pulsation-induced shaking forces in the associated compressor cylinder, pulsation dampers and pipe work.
Without limitation, all these features can cause considerable vibration and cyclic stress levels in different parts of the system. The vibration levels of reciprocating compressors are generally higher than for rotating compressors but, since they are largely determined by the design features of the compressor they tend to remain more constant over the life of the compressors than for rotating compressors.

In the case of reciprocating compressors, the vibrations measured on the main parts of the compressor and quantified according to these guidelines may only give a rough indication of the stress levels and vibratory states of the components within the compressor itself. For example, torsional vibrations of rotating parts cannot generally be determined by measurements on the structural parts of the compressor.
The damage, which can occur when exceeding the guide values for acceptable vibrations, which are based on experience with similar compressor systems, is sustained predominantly by compressor-mounted components (e.g. instrumentation, heat-exchangers, filters, pumps), connecting elements of the compressor with its peripheral parts (e.g. pipelines, pulsation dampers) or monitoring instruments (e.g. pressure gauges, thermometers). The question as from which vibration values damage is to be expected largely depends on the design of these components and their fastenings. In some cases, special measurements on certain compressor system components will be required to ascertain that the vibration values are permissible. It also happens that even if measured values are within acceptable guide values, problems may occur owing to the great variety of components which can be attached.

Such problems can be, and must be, rectified by specific “local measures” (e.g. by elimination of resonances). Experience has shown, however, that it is possible in most cases to state measurable quantities characterizing the vibratory state and to give guide values for these. This shows that the measurable variables and the guide values for acceptable vibrations in most cases permit a reliable evaluation.

If the measured vibration values as given in this document do not exceed the guidance values, abnormal wear of internal compressor components caused by vibration is unlikely to occur.

The vibrations of reciprocating compressor systems are not only affected by the properties of the compressor itself but also to a large degree by the foundation. Since a reciprocating compressor can act as a vibration generator, vibration isolation between the compressor and its foundation may be necessary in special cases.
This, as well as the vibration response of the foundation, can have considerable effect on the vibration of the compressor, pulsation dampers and attached piping itself. These vibration conditions are also dependent on the transmissibility of the environment surrounding the compressor and are therefore not entirely determined by the vibration values of the compressor itself. This guideline can therefore only take an advisory role in relation to the effects of the compressor on the environment.

Even though accelerometers and post processing is not a part of the scope of this document because it is fully covered in existing international standards, reference to the applicable ISO standards is given. Besides that, some general background information is given in respectively Annex B and C for respectively accelerometers and post processing.

High vibrations of a reciprocating compressor system are often caused by improper installation of pipe and vessel supports, loose parts, poor mounting of the compressor to the foundation, poor or no bracing of small lines with heavy masses etc. There are situations in which it is difficult to determine the root cause of the vibrations and for those cases a more extensive performance procedure is necessary. Annex D gives some guidelines how to investigate the source of unusual, abnormal or excessively high vibrations and how to approach such problems.

1 Scope

This EFRC Guideline establishes procedures and guidelines for the measurement and classification of mechanical vibration of reciprocating compressor systems. The vibration values are defined primarily to classify the vibration of the compressor system and to avoid fatigue problems with parts in the reciprocating compressor system, i.e. foundation, compressor, dampers, piping and auxiliary equipment mounted on the compressor system.

These guidelines apply to reciprocating compressor systems mounted rigidly with typical rotational speed ratings greater than 120 r/min and up to and including 1800 r/min. The general evaluation criteria which are presented relate to operational measurements. The criteria are also used to ensure that machine vibration does not adversely affect the equipment directly mounted on the machine, e.g. pulsation dampers and the pipe system.

The machinery driving the reciprocating compressor, however, is evaluated in accordance with other appropriate guidelines or standards, e.g. ISO 10816-3.

It is recognized that the evaluation criteria might only have limited application when considering the effects of internal machine components, e.g. problems associated with valves, pistons, and piston rings might be unlikely to be detected in the measurements. Identification of such problems can require investigative diagnostic techniques which are outside the scope of this document.

Examples of reciprocating compressor systems covered by these guidelines are:

- horizontal, vertical, V-, W- and L-type compressor systems,
- constant and variable speed compressors,
- compressors driven by electric motors, gas and diesel engines, steam turbines, with or without a gearbox, flexible or rigid coupling,
- dry running and lubricated reciprocating compressors.

These guidelines do not apply to hyper compressors.

It should be emphasized that these guidelines are not intended for condition monitoring purposes.

Noise is also outside the scope of the guidelines.

2 Definitions

For the purposes of these guidelines, the terms and definitions given in IS0 2041 and the following apply.

Acceleration
Rate of change of velocity.

In general, velocity is time-dependent.
The reference frame is usually a set of axes at a mean position or a position in rest. In general, a rotation acceleration vector, a translation acceleration vector, or both can represent the acceleration.

Accelerometer
Transducer that converts an input acceleration to an output (usually electrical) that is proportional to the input acceleration.

Amplitude
Magnitude, size or value of a quantity.

Branch ratio
Ratio of small bore connection nominal diameter to mainline piping nominal diameter.

Note 1: The nominal diameters can be given either in mm according to ISO definitions (DN) or in inch according to ASA definitions (NPS), see Table 2.1.

Crest factor (CF)
Ratio between peak and r.m.s. value.

Displacement
Time varying quantity that specifies the change in position of a point on a body with respect to a reference frame.

The reference frame is usually a set of axes at a mean position or a position in rest. In general, a rotation displacement vector, a translation displacement vector, or both can represent the displacement.

Dominant frequency
Frequency at which a maximum value occurs in a spectrum.

Foundation
Structure that supports a mechanical system.

Frequency
Reciprocal value of the period. The unit of frequency is hertz (Hz) which corresponds to one cycle per second.

Mainline piping
Piping of which the small bore connections are branched.

Note 1: Mainline piping can also refer to rotating machinery and pressure containing equipment like vessels or coolers.

Number of lines
Number of spectral lines that are represented.

Root Mean Square (r.m.s.)
From the measured vibration velocity time signal, the r.m.s. value may be calculated as follows:

$$v_{r.m.s.} = \sqrt{\frac{1}{T}\int_0^T v^2(t)dt} \tag{1}$$

In which:

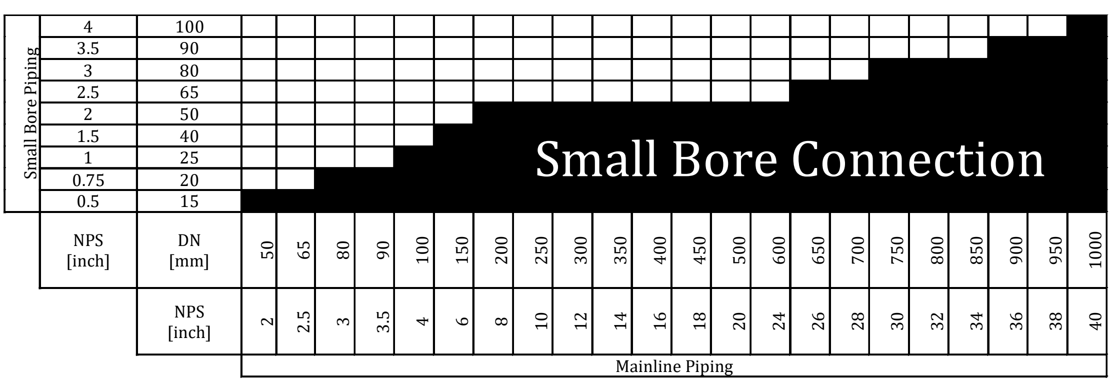
- v(t) time dependent vibration velocity value
- T sampling time

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

Small bore connections (SBC)
Branch connection on mainline piping, vessels, or equipment that is DN 50 (NPS 2") and smaller, including connections that have a branch pipe to mainline pipe ratio ("branch ratio") of less than or equal to 10 %, and excluding connections that have a branch ratio greater than 25%.
Note 1: The small bore connection piping extends until the effect of the mainline piping vibration is negligible, which is typically the first support.

Note 2: Nominal diameters can be given either in mm according to ISO definitions (DN, see ISO 6708) or in inch according to ASA definitions (NPS), see table 2.1

Table 2.1 — Definition of small bore connection



Transducer
Device designed to convert energy from one form to another in such a manner that the desired characteristics of the input energy appear at the output

Sampling rate
Number of samples per unit of time, angle, revolutions or other mechanical, independent variable for uniformly sampled data

Velocity
Rate of change of displacement.

In general, velocity is time-dependent.
The reference frame is usually a set of axes at a mean position or a position in rest. In general, a rotation velocity vector, a translation velocity vector, or both can represent the velocity.

Vibration
Mechanical oscillations about an equilibrium point. The oscillations may be periodic or random.

Window function
Pre-defined mathematical function that multiplies a data block and improves some characteristics of the frequency description

Other general terms and definitions as used in this document are:

Compressor system
Machinery system comprised of foundation, compressor (crankcase, crosshead guide, cylinders), pulsation dampers and piping.

Overall vibration value
Single numeric representation of a feature or aggregate of features derived from a raw or processed time waveform or frequency spectrum of a vibration signal and often accompanied by descriptive text or indicators to specify methods used in its derivation.

Note: The overall vibration value as indicated in these guidelines is measured in the frequency range from 2 Hz to 1000 Hz.

Corner frequency
Frequency used to convert the vibration displacement to vibration velocity and vibration velocity to vibration acceleration for a sinusoidal signal. The corner frequencies as used in this document are 10 Hz and 200 Hz, respectively.

Vendor
Vendor or vendor’s agent who supplies the compressor system.

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

3 Measurements

3.1 Measurements procedure

The measurement procedure is as follows:

- The primary measurement quantity shall be overall root mean square (r.m.s.) vibration velocity in mm/s.
- If frequencies below the corner frequencies of 10 Hz are expected or observed, it is recommended to measure also the overall vibration displacement in mm/s r.m.s. (it is common practice to measure displacement in micrometres 1 µm = 10^{-3} mm).
- If frequencies above the corner frequency of 200 Hz are expected or observed, it is recommended to measure also the overall r.m.s. vibration acceleration in m/s^2 (it is common practice to measure acceleration in units of g: 1 g = 9,81 m/s^2).
- All levels shall be within the values for acceptable overall vibration as summarised in chapter 4.3.

Note: The relation between displacement, velocity and accelerations are given in equation 2, 3, and 4 in section 4.3.

Vibration acceleration levels are normally applied for condition monitoring of internal parts of the compressor. However, the EFRC Guideline is not intended to be applied for condition monitoring purposes. If the condition of e.g. the compressor valves is to be monitored, other procedures and standards with different values can apply. The vibration accelerations levels from this guideline should therefore only serve as a criterion to judge the overall integrity of the compressor system and equipment which is attached to the compressor; e.g. pressure/temperature transmitters, valve lifting devices etc. When the vibration acceleration levels in this guideline are exceeded this does not imply that corrective actions are required. The susceptibility of components to large accelerations values (instruments, heavy components on small equipment nozzles, etc.), the presence of audible noise or knocking sounds or unusual or sudden changes of vibration levels should then become a point of attention and further analysis.
Further on it should be kept in mind that the measured acceleration values on locations as shown in figure 3.1 to 3.5 are not the values of the attached equipment but the values of the compressor system parts (foundation, crankcase, cylinder, dampers and piping) to which they are mounted.

3.2 Measuring instrumentation and measured quantities

Criteria for classifying vibration levels for reciprocating compressor systems are specified in chapter 4. It is recognized that the main excitation frequencies for reciprocating compressor systems are generally found in the range 2 Hz to 300 Hz. However, when considering the complete compressor system, including auxiliary equipment that is a functional part of the compressor, a typical range 2 Hz to 1000 Hz is applied to characterize the overall vibration. For the purpose of these guidelines, the overall r.m.s. vibration value shall represent vibration across the frequency range from 2 Hz to 1000 Hz.

Since the overall vibration signal usually contains many frequency components, there is no simple mathematical relationship between the r.m.s. and peak or peak-to-peak overall vibration measurements. Therefore, the preferred measuring system should provide the overall r.m.s. values of displacement, velocity and acceleration. However, crest factors can be a useful way to convert an r.m.s. vibration value into a peak value as explained in Annex F.

The measuring system shall provide the r.m.s. values with an accuracy of ±10 % over the range 10 Hz to 1000 Hz and with an accuracy of +10% and -20% over the range 2 Hz to 10 Hz. These values may be obtained from a single transducer whose signal is processed to derive the quantities not directly measured; preferably an accelerometer whose output is integrated once for velocity and twice for displacement. ISO 2954 gives requirements for instruments for measurements for measuring vibration severity. Guidelines on applying methods of signal processing and display, e.g. time and frequency domain, windowing and averaging, are covered in e.g. ISO 13373-2, ISO 18431-1 and common examples given in e.g. ISO 18431-2.

For small bore connections, the difference between the highest and lowest vibration velocity level between two locations shall be measured because this determines the maximum cyclic stress levels (see Annex E). The guidance values for acceptable overall vibrations are for that reason based on the difference in vibration levels measured on the two locations as defined in E.2.1. The correct phase between these two locations shall be considered.

Care should be taken to ensure that any processing does not adversely affect the required accuracy of the measuring system. Both the frequency response and measured vibration levels are affected by the method of attachment of the transducers, see Appendix B.
It is especially important to maintain a good attachment between the transducer and the compressor when the vibration velocities and frequencies are high. For example, ISO 5348 gives guidance on the mounting of accelerometers.

Note: The guidance vibration values are not applicable for ovalling shell modes of pulsation dampers and large diameter pipe systems.

3.3 Locations and direction of measurements

3.3.1 Locations

As a minimum, the vibration measurements shall be carried out on the locations as shown in figure 3.1 to figure 3.5. as follows:

- foundation: at all compressor frame bolt locations;
- compressor frame: on each corner point and between all cylinders for a compressor with more than 2 cylinders, all at the top of the frame;
- compressor cylinders (lateral and rod): at the rigid part of each cylinder cover flange;
- suction and discharge pulsation dampers: at the inlet/outlet flange and at the heads (for vertical dampers only the head on the top);
- piping: at all critical parts of the system to be determined by inspection and in agreement with the purchaser.
- small bore connections: see Figure E.1. in Annex E.

NOTE Accelerometers are often mounted on the crosshead guide for condition monitoring purposes of internal parts of the compressor. The vibrations are measured in the direction of the force exerted by the crosshead on this guide, which is in vertical direction of a horizontal compressor. Experience on horizontal compressors has shown that the vibration values measured on the crossheads guide can be used in addition to the vibration values of other locations to judge the integrity of the compressor. The procedures for measuring the vibration values on the crosshead guide are summarized in Annex G.

3.3.2 Direction of measurements

Horizontal compressor:

- foundation, frame, cylinder, pulsation dampers and piping: three mutually perpendicular X, Y and Z directions as indicated in Figure 3.1.

Vertical compressor:

- foundation, frame, cylinder, pulsation dampers and piping: three mutually perpendicular X, Y and Z directions as indicated in Figure 3.2

V-type compressor:

- foundation, frame, pulsation dampers and piping: three mutually perpendicular X, Y and Z directions as indicated in Figure 3.3.
- cylinder: three mutually perpendicular X1 (perpendicular to cylinder), Y1 (perpendicular to cylinder) and Z1 (rod direction) directions as indicated in Figure 3.3.

W-type compressor:

- foundation, frame, pulsation dampers and piping: three mutually perpendicular X, Y and Z directions as indicated in Figure 3.4.
- cylinder: three mutually perpendicular X1 (perpendicular to cylinder), Y1 (perpendicular to cylinder) and Z1 (rod direction) directions as indicated in Figure 3.4.

L-type compressor:

- foundation, frame, cylinder, pulsation dampers and piping: three mutually perpendicular X, Y and Z directions as indicated in Figure 3.5.

Location numbers for all type of compressors:

1. All compressor frame bolt locations.
2. Each frame corner point.
3. Each frame location between the cylinders (required for a compressor with more than two cylinders).
4. Each cylinder (cover flange at rigid location).
5. Pulsation vessels (only shown for one vessel in the figure).

NOTE The numbers apply to all types of compressors (for clarity only one point is shown in the figure for most of the locations). Piping is not shown in the figure and the locations shall be agreed upon with the vendor.

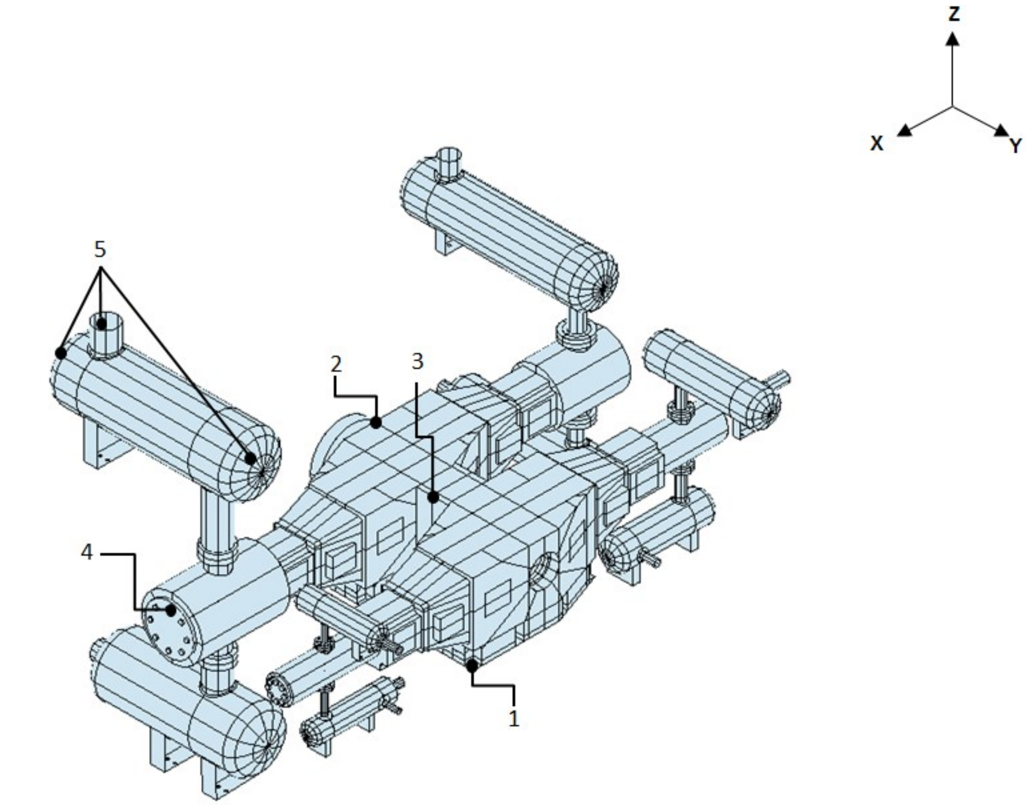


Figure 3.1 Measuring locations for a horizontal compressor

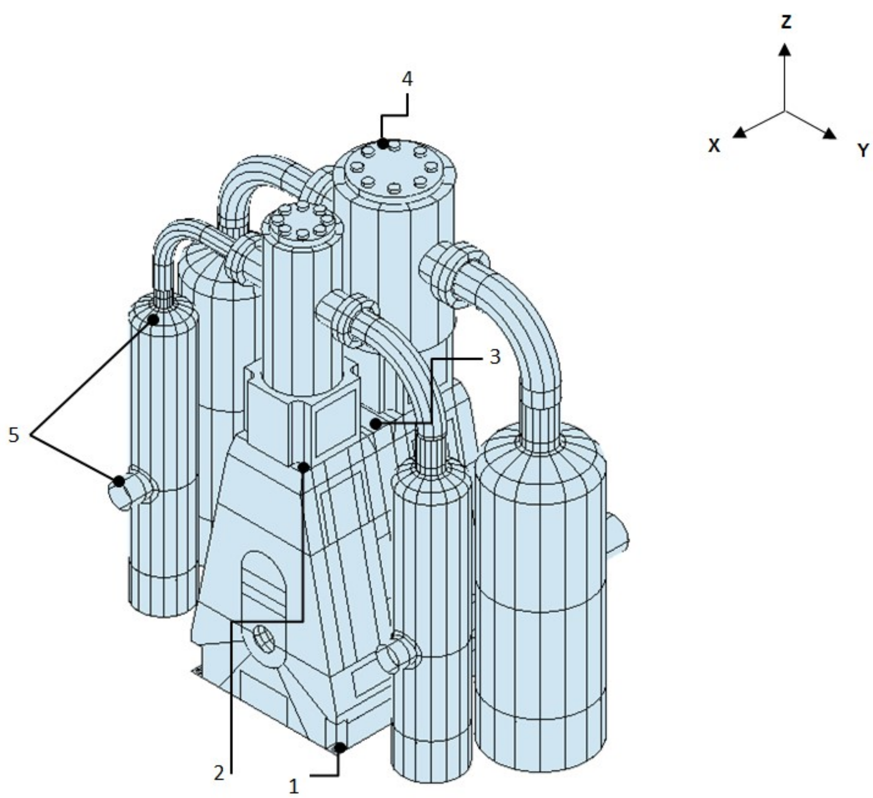


Figure 3.2 Measuring locations for a vertical compressor

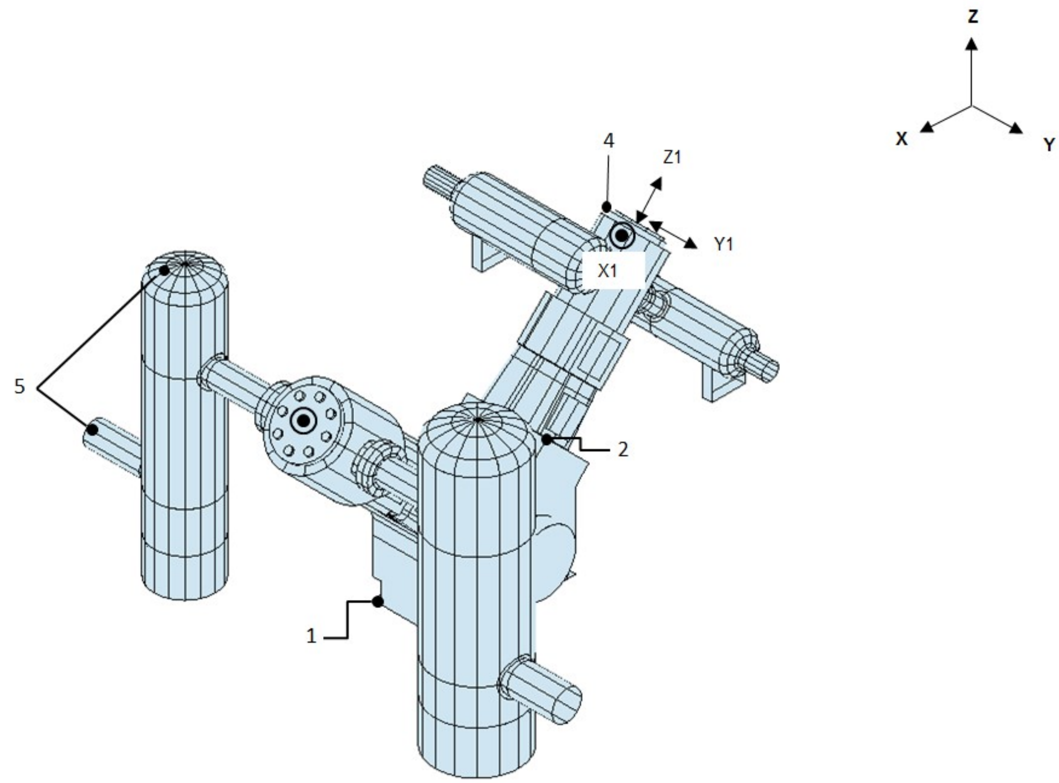


Figure 3.3 Measuring locations for a V-type compressor

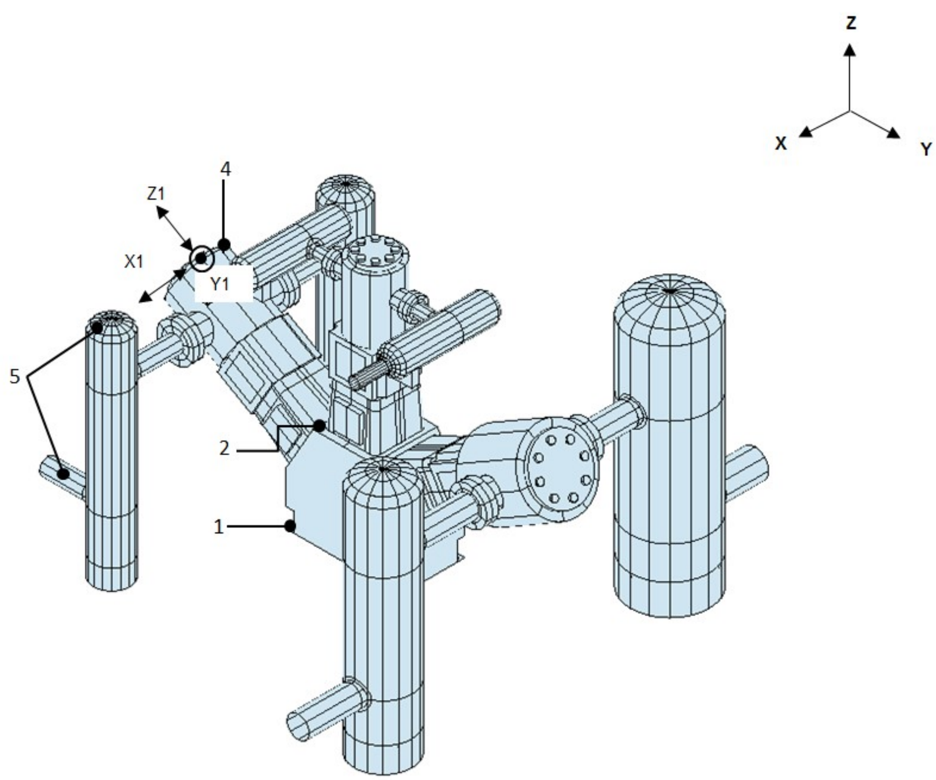


Figure 3.4 Measuring locations for a W-type compressor

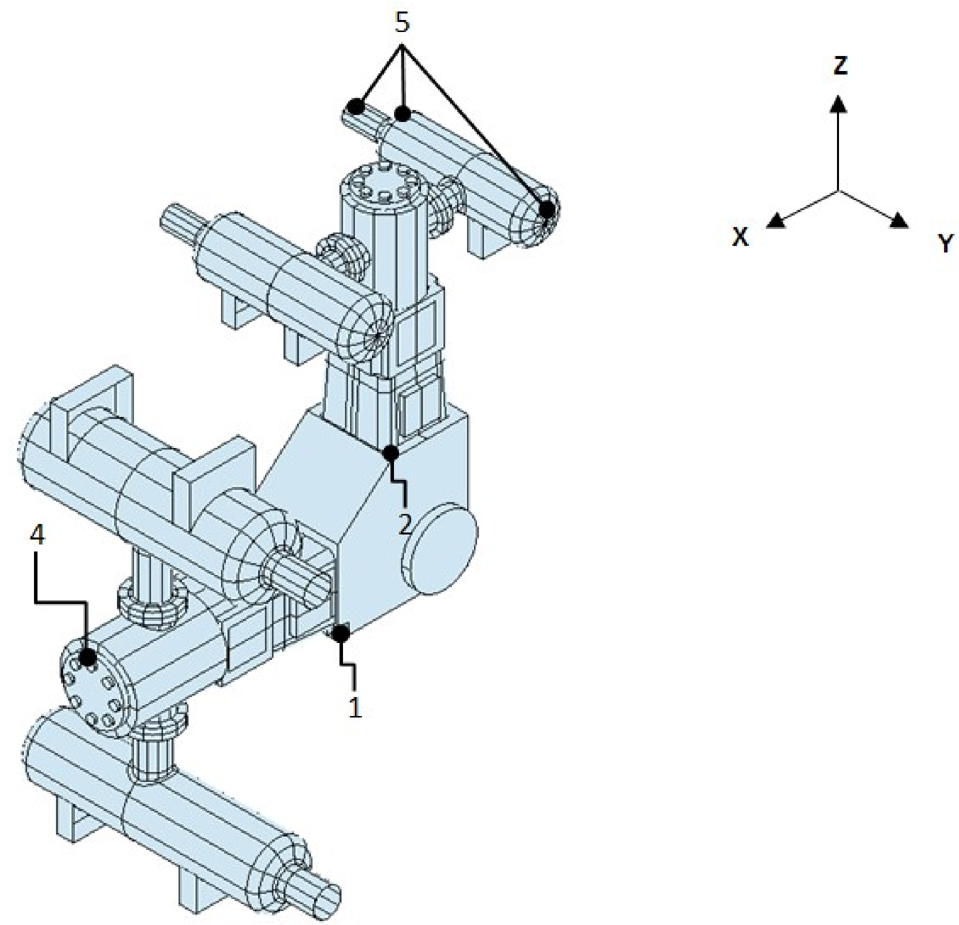


Figure 3.5 Measuring locations for an L-type compressor

3.4 Operating conditions

Measurements shall be taken when the compressor has reached its steady-state operating conditions (e.g. normal operating temperature). The determination of the vibration values shall be based on the maximum vibration values occurring over the entire speed range (if applicable), for all operating process conditions (e.g. different pressures, temperatures), specified alternative gases (e.g. N_2 for start-up), unloading conditions, single and multiple compressors in service, etc.

3.5 Record of measured results

Records of all measured results should include essential data of the compressor system and of the measuring system used.

4 Vibration Criteria

4.1 Measuring quantities

The limits for overall vibration displacements, vibration velocities and vibration accelerations are in r.m.s.

4.2 Evaluation zones

4.2.1 General

The following evaluation zones are defined to permit a qualitative assessment of the vibration on a given compressor system and to provide guidelines on possible actions. Numerical values assigned to the zone boundaries are primarily intended to serve as guidance values and are not intended to serve as a final acceptance criterion. The guidance values for acceptable vibrations are intended to ensure that gross deficiencies or unrealistic requirements are avoided.
In certain cases, there can be specific features associated with a particular compressor system which would require different boundary values (smaller or larger) to be used, which shall be agreed between the vendor and purchaser. In such cases, it is normally necessary to clarify the reasons for this and, in particular, to confirm that the compressor system will not be endangered by operating with larger vibration values.

Zone A, zone B: Compressor systems with vibration within these zones are normally considered acceptable for long-term operation.

Zone C: Compressor systems with vibration within this zone are normally considered unsatisfactory for long-term continuous operation. Generally, the compressor may be operated for a limited period in this condition until a suitable opportunity arises for remedial action such as analysis and possible correction.

Clarify between vendor and purchaser that the compressor is suitable for long-term safe operation.

Zone D: Vibration values within this zone are normally considered to be of sufficient severity to cause damage to the compressor and attached equipment.

Transitions for the zone boundaries for reciprocating compressor systems are summarized in Table 4.1.

Table 4.1 Evaluation zone descriptions:

Zone	**Range**	**Criterion**	**Description** (See Notes)
A	≤ A/B	Acceptable	Compressor systems with vibration within these zones are normally considered acceptable for long-term operation.
B	> A/B and ≤ B/C		
C	> B/C and ≤ C/D	Marginal	Analysis and possible correction to be considered. Clarify between vendor and purchaser that the compressor is suitable for long-term safe operation.
D	> C/D	Unacceptable	Urgent correction or shutdown to be performed (see Note 3 and Note 4).

NOTE 1 These guidance values are not applicable to test bed conditions. Test bed conditions are unlikely to represent in-situ conditions due to variations in foundation flexibility, fixings and supports, loading, flow, gas conditions, resonances, piping, valves, vessels, etc. For test bed conditions, other values might need to be applied based on the compressor OEM's experience and in agreement with the purchaser.

NOTE 2 Zone B is included to define the range A/B to B/C. It may be used as an engineering reference. Field measurements taken on mainly low-speed, constant process-condition machines are centred around the zone boundary A/B.

NOTE 3 If the vibration velocity of the main pipe system exceeds the appropriate C/D vibration value (zone D), this does not, by definition, mean that a fatigue failure in the main piping will occur. Fatigue failures often occur in small bore piping and attached equipment to the main piping, e.g. pressure and temperature transmitters or drains. For that reason, consideration to shut down the system is not necessary if all of the following is fulfilled:

- The maximum vibration velocity in of the main pipe system does not exceed an r.m.s. value of 45 mm/s;
- The vibration levels of the Small Bore Connections (SBC's) which are attached to the relevant main pipe system shall be measured and shall not exceed the guidance levels as defined in Annex E. If the vibration levels exceed the guidance levels, Note 4 is applicable.
- Vibration displacement values of the main piping are smaller than the defined values of zone boundary C/D;
- Analysis of the relevant main pipe section shows that fatigue failure is not likely to occur, e.g. by analytical methods, finite element analysis, modelling, or strain gauge measurement;
- Acceptance for long-term operation shall be agreed upon between the vendor and purchaser.
- Vibrations in zone D are commonly caused by the excitation of mechanical natural frequencies. For that reason, the design shall be such that the excitation frequencies do not coincide with the mechanical natural frequencies.

NOTE 4 If the relative vibration velocity of the small bore connection exceeds the appropriate C/D vibration value (zone D), this does not, by definition, mean that a fatigue failure of the small bore connection will occur. The stress in a small bore connection is influenced by the geometry, connection type, weld details and quality, etc. For that reason, consideration to shut down the system is not necessary if the cyclic stress does not exceed the endurance limit which can be proved as follows:

- Measure the actual cyclic stress at the critical points, in general the welds close to the mainline, with a strain gauge measurement and compare the cyclic stress with the fatigue limit of the weld.
  or:
- Measure the difference of the vibration displacement time domain waveform [peak-to-peak displacement in mm] between the small bore connection and the mainline piping (relative vibration).
- Take geometry data on the small bore connection and mainline piping, including diameters, lengths and wall thickness.
- Conduct a fatigue analysis using the relative vibration displacement using proved analytical methods or finite element analysis and check if the maximum calculated cyclic stress does not exceed the endurance limit.

4.2.2 *Acceptance criteria*

Acceptance criteria shall be subject to agreement between the vendor and purchaser prior to purchase the installation. Table 4.1. provides a basis for defining acceptance criteria for new or refurbished machines

4.3 Guidance values for acceptable overall vibration values

The guidance values for acceptable overall vibration displacement, vibration velocities and vibration accelerations values for a horizontal and vertical compressor system are summarised in table 4.2 to 4.4 and are graphically shown in figure 4.1 to figure 4.11.
Unless otherwise specified, the guidance values for V- and W-type compressors are the same as for vertical compressors. For L-type compressors, the values for the horizontal and vertical throw are the same as those for horizontal and vertical compressors, respectively.

The values were derived from constant vibration displacement in the frequency range 2 Hz to 10 Hz, constant vibration velocity from 10 Hz to 200 Hz and constant vibration acceleration from 200 Hz to 1000 Hz. The frequencies of 10 Hz and 200 Hz are the corner frequencies. For sinusoidal signals, the relation between vibration displacement, vibration velocity and vibration acceleration is as follows and can be used to convert vibration displacement and vibration acceleration to vibration velocity at the 10 Hz and 200 Hz corner frequencies for a sinusoidal signal.

displacement: $x = \int v dt = \iint (a dt) dt = -\frac{1}{\omega^2 a}$

velocity: $v = \frac{dx}{dt} = \int a dt = \omega x$

acceleration: $a = \frac{dv}{dt} = \frac{d^2 x}{dt^2} = \omega v = \omega^2 x$

where ω is the angular frequency of the vibration with $\omega = 2\pi f$

Table 4.2- Summary of overall constant vibration displacement values for different compressor system parts

Compressor system part	r.m.s. vibration displacement values for horizontal compressors [mm]			r.m.s. vibration displacement values for vertical compressors [mm]		
	Evaluation zone boundary			Evaluation zone boundary		
	A/B	B/C	C/D	A/B	B/C	C/D
Foundation	0,032	0,048	0,072	0,032	0,048	0,072
Frame (top)	0,084	0,127	0,191	0,084	0,127	0,191
Cylinder (lateral)	0,139	0,207	0,310	0,170	0,255	0,382
Cylinder (rod)	0,170	0,255	0,382	0,139	0,207	0,310
Dampers	0,202	0,302	0,454	0,202	0,302	0,454
Piping	0,202	0,302	0,454	0,202	0,302	0,454
Small bore connection	See Table E.1					

Table 4.3- Summary of overall constant vibration velocity values for different compressor system parts

Compressor system part	r.m.s. vibration velocity values for horizontal compressors [mm/s]			r.m.s. vibration velocity values for vertical compressors [mm/s]		
	Evaluation zone boundary			Evaluation zone boundary		
	A/B	B/C	C/D	A/B	B/C	C/D
Foundation	2,0	3,0	4,5	2,0	3,0	4,5
Frame (top)	5,3	8,0	12,0	5,3	8,0	12,0
Cylinder (lateral)	8,7	13,0	19,5	10,7	16,0	24,0
Cylinder (rod)	10,7	16,0	24,0	8,7	13,0	19,5
Dampers	12,7	19,0	28,5	12,7	19,0	28,5
Piping	12,7	19,0	28,5	12,7	19,0	28,5
Small bore connection	See Table E.2					

Table 4.4- Summary of overall constant accelerations values for different compressor system parts

Compressor system part	r.m.s. vibration acceleration values for horizontal compressors [m/s^2]			r.m.s. vibration acceleration values for vertical compressors [m/s^2]		
	Evaluation zone boundary			Evaluation zone boundary		
	A/B	B/C	C/D	A/B	B/C	C/D
Foundation	2,5	3,8	5,7	2,5	3,8	5,7
Frame (top)	6,7	10,1	15,1	6,7	10,1	15,1
Cylinder (lateral)	10,9	16,3	24,5	13,5	20,1	30,2
Cylinder (rod)	13,5	20,1	30,2	10,9	16,3	24,5
Dampers	16,0	23,9	35,8	16,0	23,9	35,8
Piping	16,0	23,9	35,8	16,0	23,9	35,8
Small bore connection	See Table E.3					

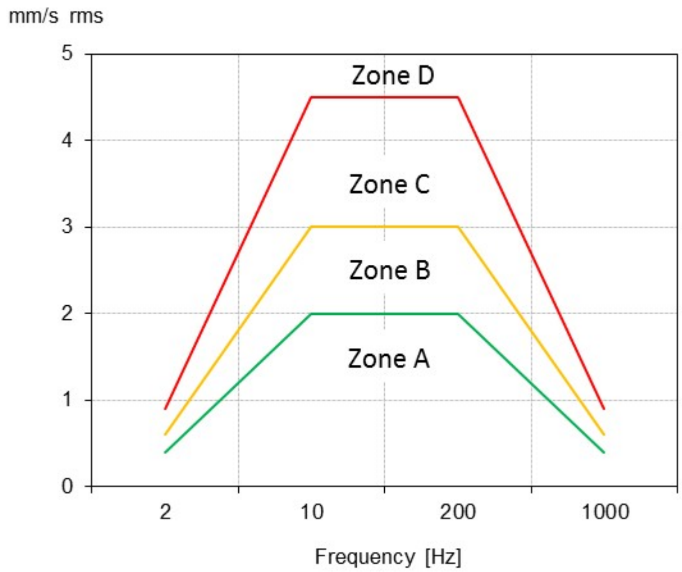


Figure 4.1. Overall vibration velocity limiting curve for the foundation of a horizontal compressor.

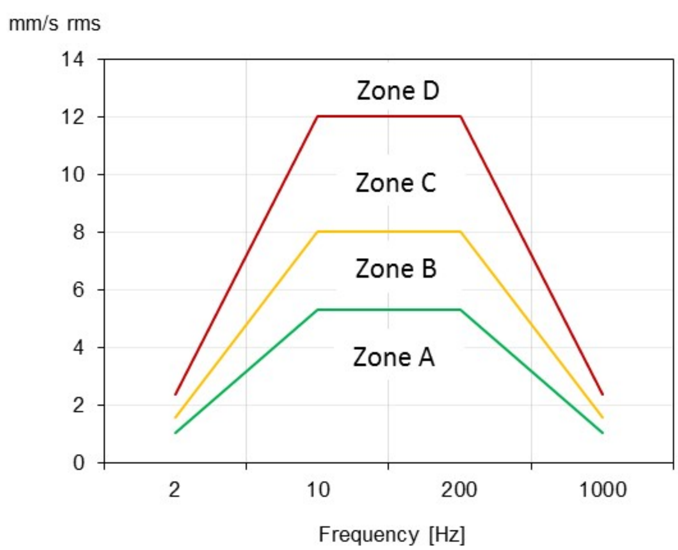


Figure 4.2- Overall vibration velocity limiting curve for the frame of a horizontal compressor.

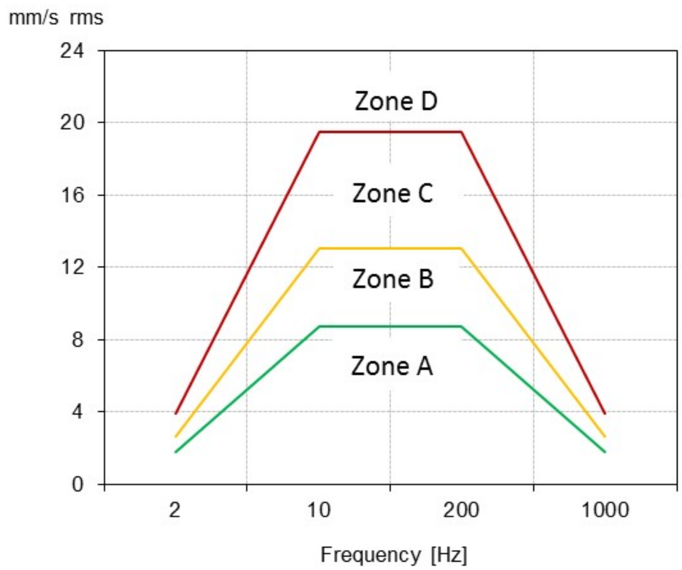


Figure 4.3-. Overall vibration velocity limiting curve for the cylinder in lateral direction of a horizontal compressor

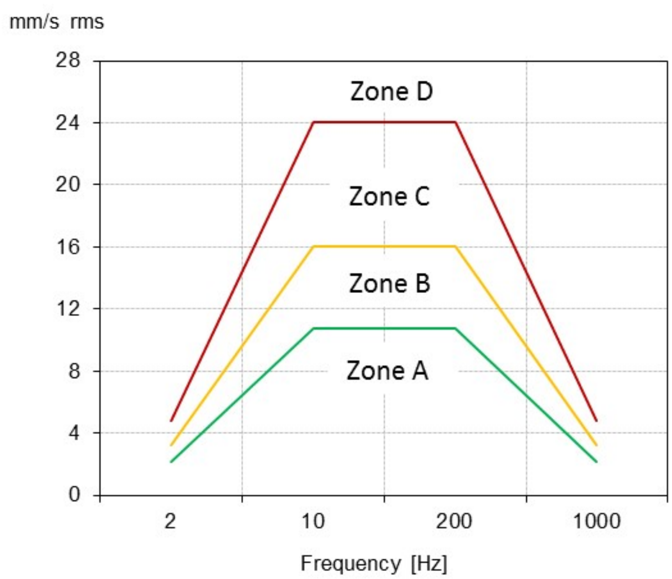


Figure 4.4.- Overall vibration velocity limiting curve for the cylinder in rod direction of a horizontal compressor

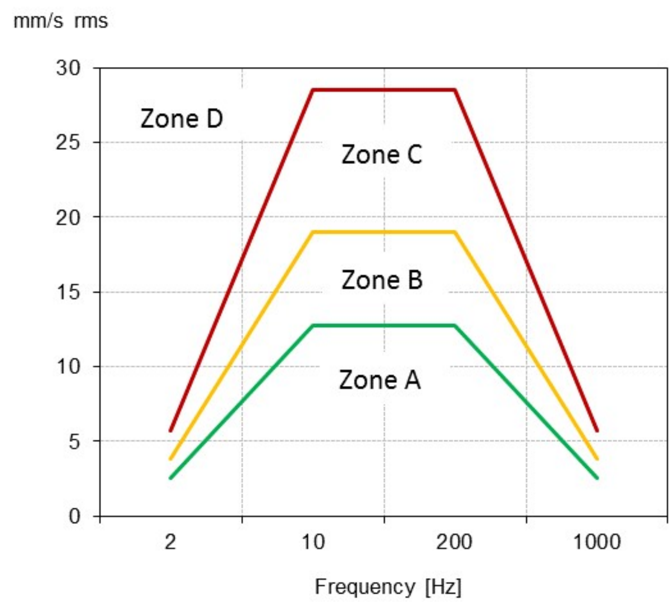


Figure 4.5.- Overall vibration velocity limiting curve for the piping and dampers of a horizontal compressor

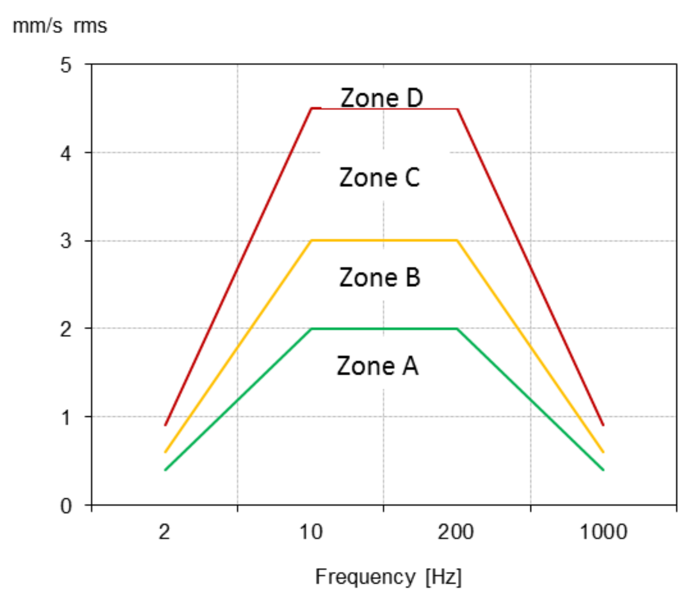


Figure 4.6. Overall vibration velocity limiting curve for the foundation of a vertical compressor

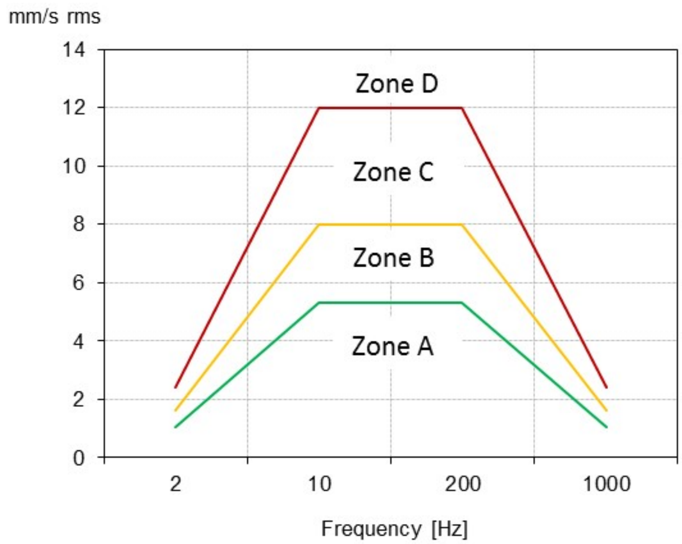


Figure 4.7- Overall vibration velocity limiting curve for the frame of a vertical compressor

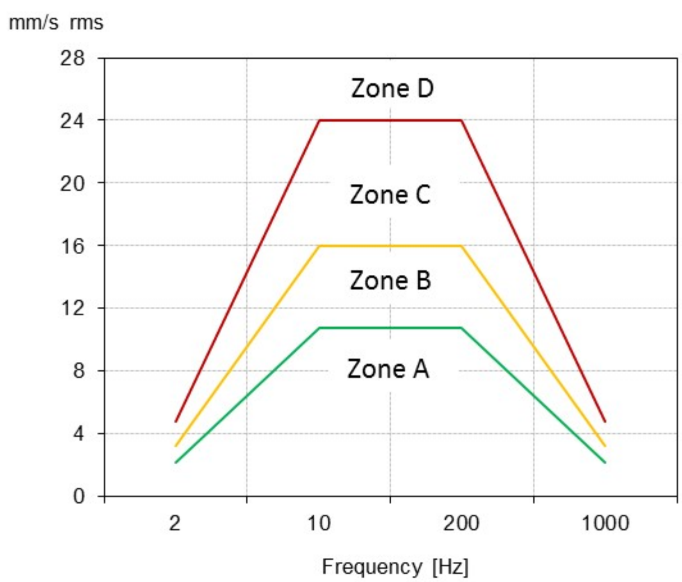


Figure 4.8- Overall vibration velocity limiting curve for the cylinder in lateral direction of a vertical compressor

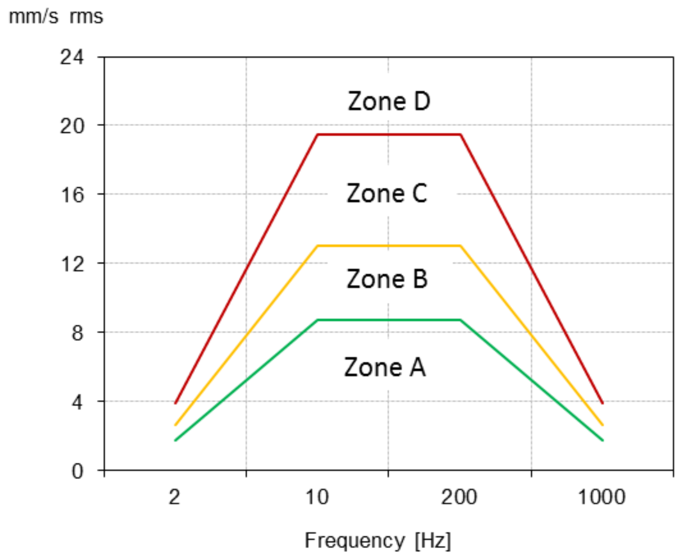


Figure 4.10.- Overall vibration velocity limiting curve for the cylinder in rod direction of a vertical compressor

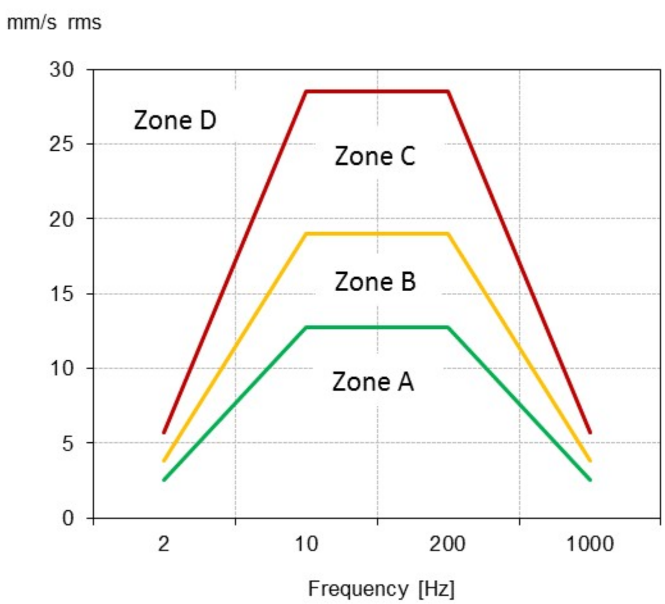


Figure 4.11.- Overall vibration velocity limiting curve for the piping and dampers of a vertical compressor

4.3.1 Vibration values and the effect of mountings and foundations.

The vibration values as given in table 4.2 to 4.4 are valid for rigidly mounted compressor systems. This means that the compressor and driver must be mounted directly to the concrete foundation. If the compressor and driver are mounted on a skid, the skid must be stiff enough and directly mounted to the concrete foundation. The structure on which the compressor is mounted, either concrete or skid, shall not be in resonance. Operation at or near resonance shall be avoided.

Isolated mounted foundations e.g. concrete block on springs and skids on anti-vibration mounts (AVM), are an exception and the vibration values for such systems should be agreed upon between with the purchaser and vendor.

4.3.2 Vibration values for horizontal compressors

The gas (stretching) force in the cylinder causes vibration in the rod direction. In general, the vibration in the rod direction is larger than in the lateral direction. The vibration in the rod direction of the cylinder causes tensile and compression stresses and is generally considered less harmful than the lateral vibrations which causes bending stresses. For that reason, larger vibration values in the rod direction of the cylinder are allowed than in the lateral direction.

4.3.3 Vibration values for vertical and V-type compressors

For vertical and V-type compressors, larger vibration values in lateral direction than in the rod direction of the cylinder are allowed since the vertical and V-type compressor is in general more flexible in the lateral direction than a horizontal compressor.

4.3.4 Material

Several materials are more susceptible to crack initiation, e.g. cast iron and several high alloy materials are more brittle than carbon steel. When more information is available, guidance may be provided on this subject to adjust the guidance values for acceptable overall vibration values.

Annex A: Measuring procedure and data processing

A 1 Compressor details:

As a minimum for each compressor system being measured, the following information shall be recorded:

ITEM	EXAMPLE
Unique compressor identifier	equipment code or tag number
Compressor type:	gas compressor/other
Number of cylinders:	
Rated rotational speed:	r/min or Hz
Constant or variable speed:	constant/ variable
Speed variation (if applicable):	minimum speed, maximum speed: r/minor Hz
Rated power:	kW
Configuration:	horizontal / vertical / V-type / L-type / W-type
Compressor support:	rigid or resiliently mounted
Shaft coupling:	rigid or flexible
Type of flow control:	valve unloaders, bypass, clearance pocket, step less flow reversal control, speed

The following information may also be useful to record:

ITEM	EXAMPLE
Driver type:	electric motor, internal combustion engine

A 2 Measurements

For each measuring system, the following information should be recorded:

ITEM	EXAMPLE
Date, and time (including time zone)	
Instrument type:	make and model of instrument
Measurement location, orientation:	drawing (preferred), description or code
Measurement units:	mm, µm, mm/s, m/s^2 Measurement qualifier r.m.s.
Measurement type:	overall/amplitude/spectrum/time history
Transducer type:	accelerometer, eddy current, velocity
Transducer attachment method:	probe/magnet/stud/adhesive
FFT or other Processing:	filter (i.e. low and high cut-off frequencies), number of lines, number of averages, number of samples, window type
Calibration requirement:	type and date of last required calibration

The following process and operating parameters shall also be recorded:

ITEM	EXAMPLE
Speed during measurement:	r/min or Hz
Power during measurement:	kW
Operation of multiple compressors:	single, parallel
Load condition:	load%, load steps
Operating parameters:	temperatures, pressures, gas composition

A 3 Other information

Extra information on the compressor system and the measurements may be recorded in addition to the above, e.g. historical maintenance data. An example of a form to record asset and measurement data for the compressor types is shown in Table A.1.

Table A.1 — Form for recording typical compressor details

General	
Record No.:	Installation site:
Date:	Measured by:
Details of compressor system	
Unique compressor ID No.:	Type/Serial No.:
Type: gas compressor / other [a]	Driver type: [a]
Number of cylinders: 1/2/3/4/5/6/8/12/other [a]	Coupling: rigid/flexible[a]
Configuration: horizontal/vertical/other [a]	Load condition during measurement:
Rated speed: r/min	Rated power: kW
Actual speed: r/min	Power during measurement: kW
Mounting: rigid/resilient[a]; directly/on base plate[a]	Running hours:
Vendor:	
Details of each measuring system	
Instrument make:	Instrument model:
Measurement units:	FFT or other processing details:
Measurement unit qualifier:	
Transducer type and make:	Attachment:
Diagram	Sketch compressor below:
Measurement records, readings, diagrams, etc. should be attached giving locations of measurement, and the conditions at the time of measurement, if applicable.	
[a] Delete/supplement as appropriate	

Annex B: Accelerometers

B 1 General

The accelerometer is a contact sensor (as opposed to a non-contacting proximity probe) that measures the motion of the surface to which it is attached. Its many benefits include linearity over a wide frequency and dynamic range. Accelerometers are available with various mounted resonant frequencies, typically from 1 kHz upwards.
In general, it is recommended that the linear range of the mounted accelerometer covers the frequencies of interest. It is common to integrate the output of an accelerometer to provide a velocity signal. However, caution shall be exercised when double-integrating to provide displacement, especially at low frequencies. If accelerometers are used with slow running compressors, suitable types will have a high transmission factor to ensure an adequate signal-to-noise ratio despite the double integration which is required for vibration displacement measurements. If the used accelerometer is not suitable to measure low frequencies, electro dynamic vibration velocity transducers should be applied in that case.
To apply the accelerometer and get reliable measurements, proper attention must be paid to the following areas:

- Sensor mounting configurations
- Frequency range of interest
- Amplitude of interest
- Characteristic of the accelerometer under considerations
- Cabling and signal condition
- Environmental considerations

B 2 Mounting

Since the accelerometer is a contact device, care in mounting is of importance because improper installation can affect the performance of the device and give unreliable and unexpected output signals. Proper measurement of compressor vibration is critically dependent on accurately transmitting the motion to the transducer. The broadest range of fidelity is obtained with fixed transducer attachments. However, in many cases, hand-held or magnet-attached probes are sufficient.
For a complete description of transducer attachment methods of accelerometers and their effect on performance, refer to ISO 5348. General guidelines are included below.
The preferred method for attaching fixed transducers is a rigid mechanical fastening which is commonly achieved by drilled and tapped holes in the transducer and the compressor, and joining the two by a threaded stud. Stud mounting can transfer high-frequency signals with little or no signal loss.
The contact surface should be smooth, flat and clean. However, this method of mounting is rather time consuming and not always allowed or practically feasible. In most cases, it is not required because the frequency range of interest is limited up to typically 1000 Hz for vibration measurements for reciprocating compressor systems. For that reason, a common applied technique for non-intrusive transducer fastening is with a permanent magnet.
However, it should be noted that the flatness of the mounting surfaces is critical in this technique. For pipe systems, it is advised to mount the transducer on the flat areas of flanges, or to utilise magnets with two parallel feet which allow attachment to surfaces with single-axis of curvature.
In case of non-magnetic materials (often applied for piping and pulsation dampers), cements or glues can be used to fasten the transducer to the surface. The cement or glue used shall be of the type that

has high stiffness characteristics when cured. Resilient adhesives should be avoided as they reduce the fidelity of the transmission of the signal.
Another method is the application of hand-held probes. Hand-held probes are frequency-limited and are normally not recommended for use above approximately 1 kHz. Both accuracy and repeatability are likely to be compromised using hand-held probes. Moreover, some structural motions at the higher frequencies can invalidate hand-held probe measurements, even though such motions may not be detectable with the probe. The mounting of an accelerometer has a large influence on the transducer response as shown in figure B.1. for a frequency range from zero up to the resonance frequency of different mountings.
To demonstrate more into detail the effect on transducer performance of the various transducer attachment methods described above, the mounted resonant frequency of an accelerometer, with an internal 30 kHz resonant frequency, is typically reduced as shown in Table B.1. The response close to the resonance frequency of different mountings is shown in figure B.2. The numbers in this figure refer to the numbers of the first column of Table B.1.

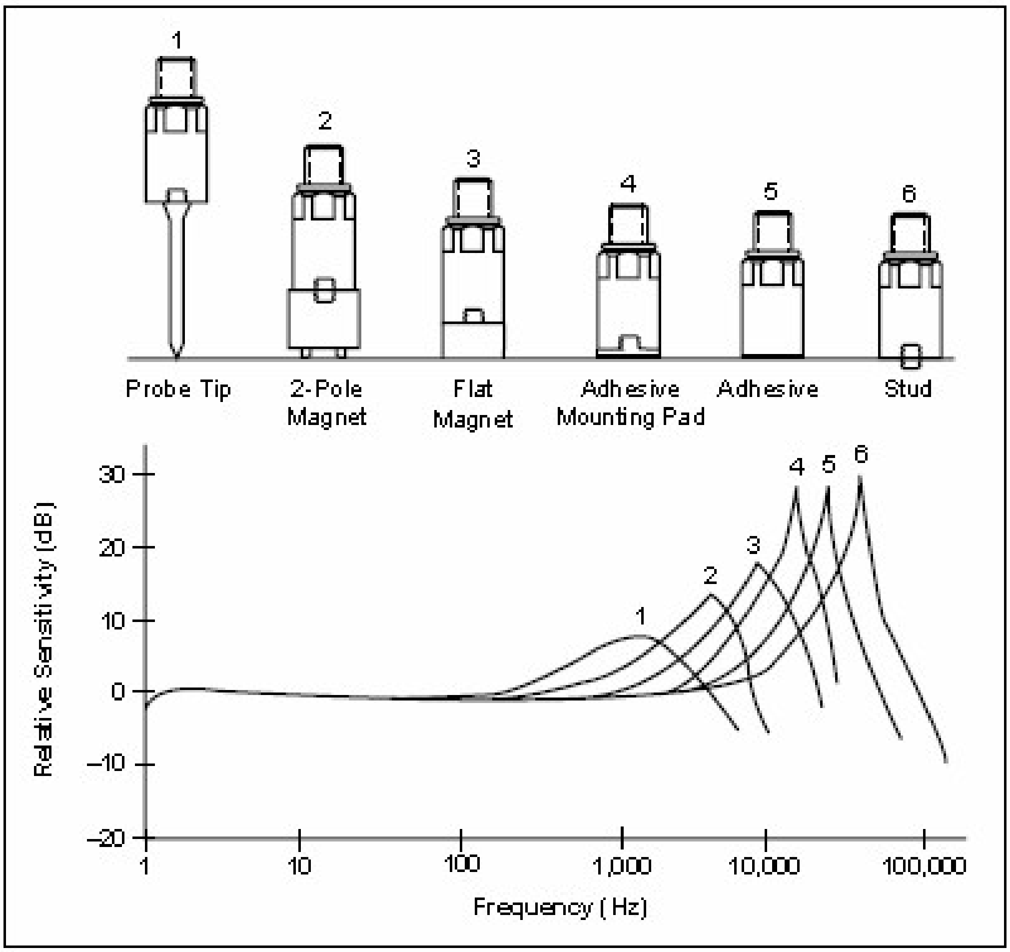


Figure B.1 Typical transducer response for different mountings

Table B.1 Effect of accelerometer mounting method on resonant frequency

Effect See Fig B.2	Mounting method	Effect on system resonant frequency (f_r)	Comment Assuming an accelerometer with 30 kHz resonant frequency
1	Rigid stud mount	No effect	e.g. M6 stud with flat machined location point
2	Isolating stud mount	Reduced to about 28 kHz	e.g. mica washer
3	Stiff cement mount	Reduced to about 28 kHz	e.g. cyanate adhesive (superglue™ or crazy glue)
4	Quick-fit stud mount	Reduced to 10 – 20 kHz	Will depend on cleanliness
5	Soft epoxy mount	Reduced to about 8 kHz	e.g. two-part epoxy filled resin
6	Permanent magnet mount	Reduced to 2 – 7 kHz	Will depend on magnet and attachment surface flatness/curvature
7	Hand-held probe (stinger)	Reduced to 0,5 – 2 kHz	Will depend on material and length of probe e.g. For probe length 228 mm (9") $f_r \approx 0{,}75$ kHz

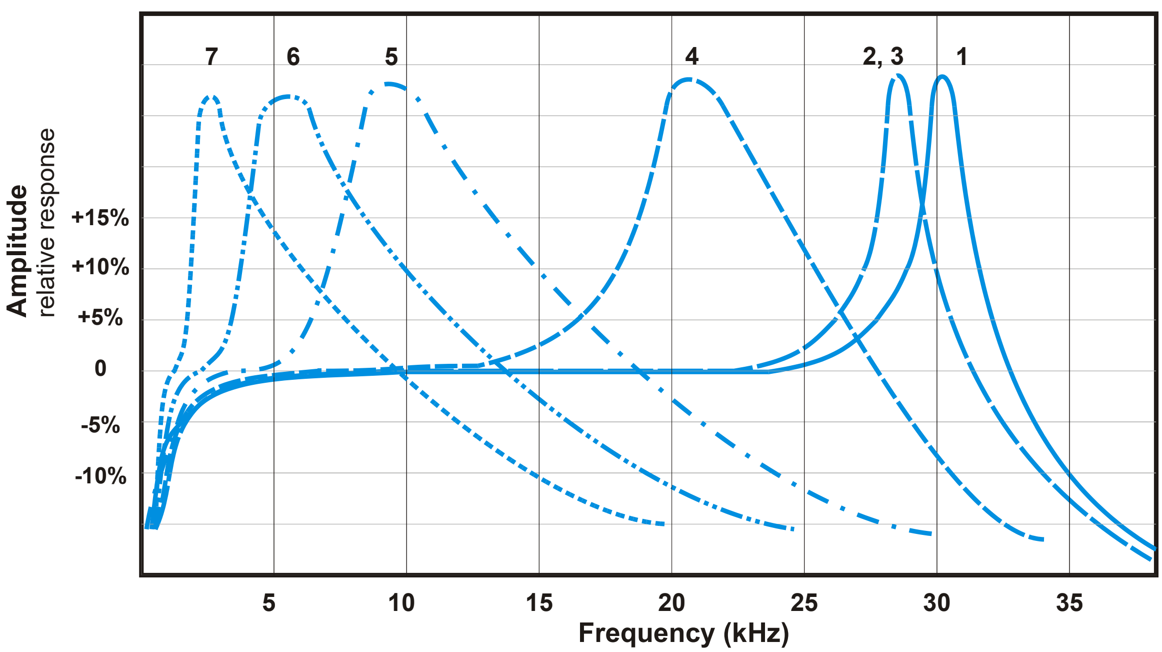


Figure B.2 Response close to the resonance frequency of different mountings according table B.1

B 3 Cables

Stiff cables can cause case strain when used with accelerometers with axial connectors. Careful clamping of the cables is required to avoid such problems (see figure B.3.). Loose cables may introduce tribo-electric effects for piezo-electric type transducers with separate remote charge-coupled amplifiers. This is generally not a problem with IEPE (Integrated electronics piezo-electric) accelerometers.

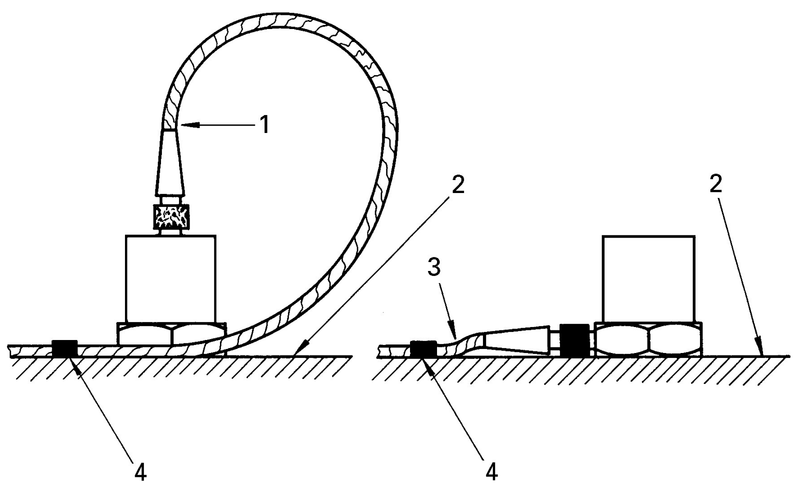


Figure B.3. Correct mounting of cables

a) left picture: accelerometer with axial connector
b) right picture: accelerometer with radial connector

Annex C: Post processing

C 1 General

Unless prior knowledge dictates otherwise, the baseline broadband vibration should be acquired covering a sufficient dynamic range and frequency response to include all forcing excitations of interest within the compressor. Procedures for measuring broadband vibration can be found in the ISO 10816 series, respectively.
Broadband vibration does not always provide sufficient information to identify the specific cause of a problem. This is especially true for complex equipment where several excitation frequencies appear in the frequency spectrum. In such cases, it is advisable to split the broadband vibration signal into discrete frequency components (amplitude and/or phase).
In most cases, individual frequencies can be matched with corresponding compressor speeds or harmonics of the compressor speed and several types of problems (pulsation-induced of mechanical induced fatigue) can be detected by discrete frequency analysis. A sample frequency spectrum (FFT) plot is given in Figure C1. It clearly shows vibration amplitudes at specific frequencies.
It is important to evaluate the source of significant frequency peaks as their magnitudes can be abnormal and when immediately investigated can lead to early detection of a problem. Unexpected frequency components in the frequency analysis can also reveal some design configuration.
More detailed procedures for converting a broadband time trace to a frequency spectrum is given in ISO 13373-2.

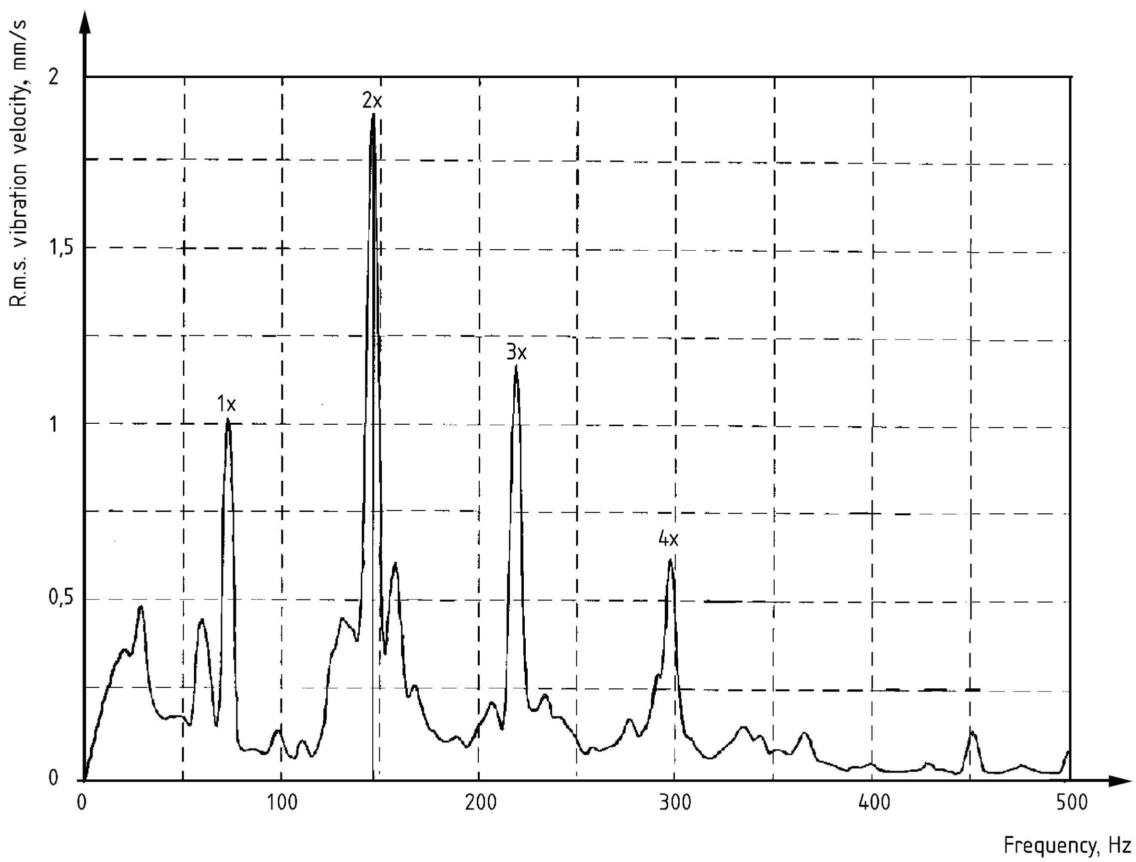


Figure C1. Typical frequency spectrum of a reciprocating compressor system

The following table shows some of the measurement parameter settings required to be selected before taking a vibration measurement.

Table C.1 – Typical vibration parameter acquisition options

Ref.	Description	Typical Choice	Examples
1	Transducer	Accelerometer	Accelerometer, velocity, displacement
2	Measurement Type		Acceleration, velocity, displacement
3	Data Format	Spectra	Time sample, spectra,
4	Trend Parameters		Overall RMS acceleration, Overall RMS velocity, Overall RMS displacement
5	Input Filter	2,5 Hz – 20 kHz	High pass (HPF), band pass (BPF), ISO 10816: 10-1000Hz
6	Other Trend	Crest Factor	Crest Factor, HFD
7	Other Processing	Envelope	
8	Sampling Rate	9600 Hz	4800 Hz, 9600 Hz, 19200 Hz (may be pre-set via option 10 & 11)
9	Sample Length	4096	2048, 4096, 8192 (may be pre-set via option 10 & 11)
10	FFT Lines	1600	100, 200, 400, 800, 1600, 3200, 6400, 12800 etc.
11	Max. Frequency	2 kHz	500 Hz, 1 kHz, 2 kHz, 5 kHz, 10 kHz etc.
12	FFT averages	4	0, 2, 4, 8, 16 etc. (averaging reduces noise in FFT)
13	FFT Window	Hanning	None, Uniform, Hanning, Hamming, Exponential
14	FFT Anti-Aliasing	Yes	Yes, No

C 2 Digital signal processing (DSP)

C.2.1 General

Modern digital Dynamic Signal Analysers (DSA) carry out sampling and processing of data quickly and cheaply. However, they can apparently modify the input signal and introduce spurious components unless careful selection of features such as linear averaging, time synchronous averaging, exponential averaging, overlapping data, acquisition time, dynamic range influence, truncation, anti-aliasing and noise elimination. See ISO 13372-2 for details on these functions.

To illustrate some of these requirements, a schematic of some of the functions of a modern DSP are shown in Figure C.2

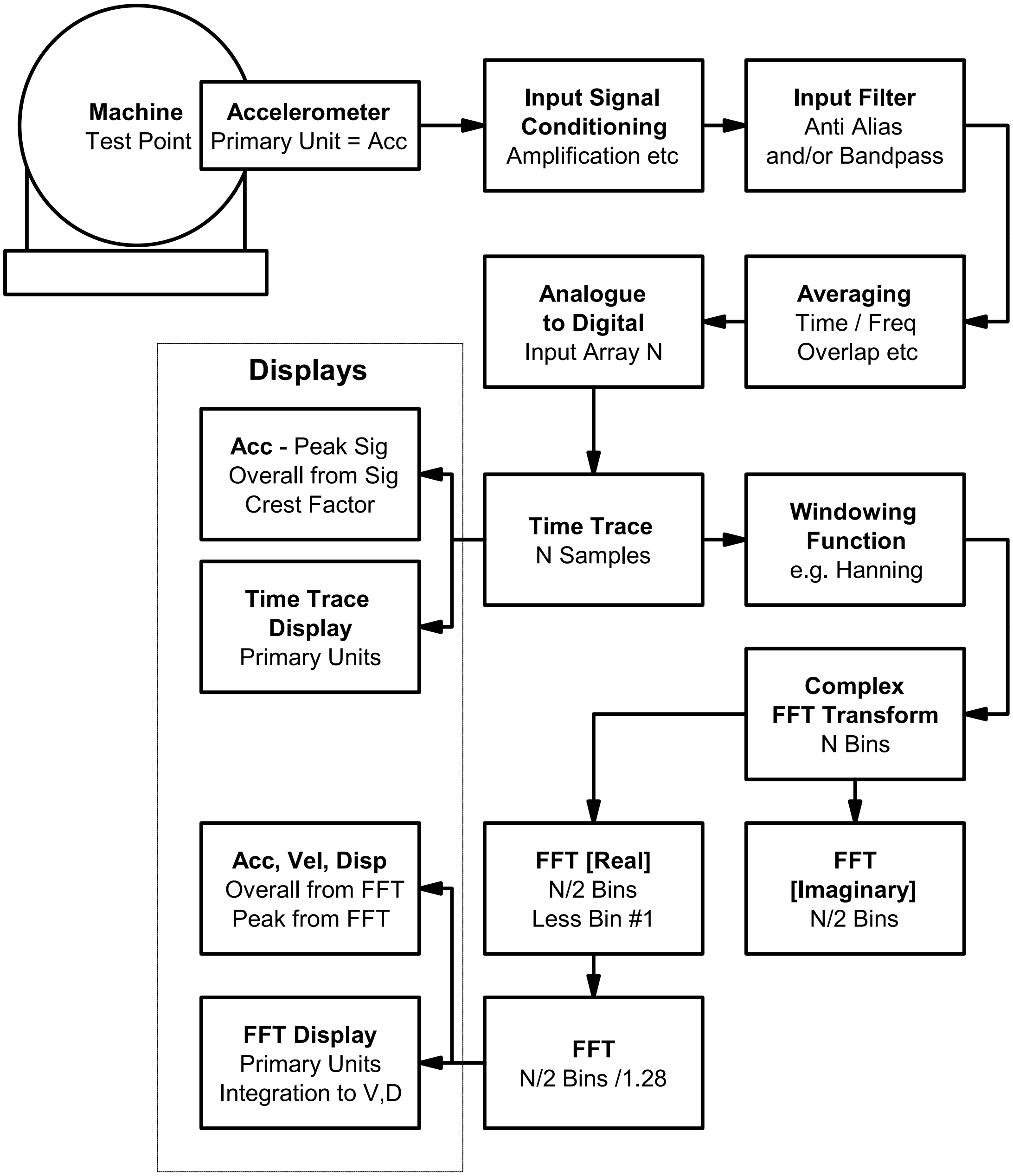


Figure C.2 – Schematic of Typical Digital Signal Processing

C 2.2 Averaging

Averaging is a process to improve clarity of harmonic signals, and may be carried out in the time or frequency domain. ISO 13372-2 contains useful information regarding averaging. Averaging reduces the effect of non-synchronous components and noise.

Digitized signals can be averaged over configurable periods of time by means of various methods. Averaging in the time range is a much faster way of removing noise and noise-like signal components than averaging spectra. However, time signals must be in-phase to be averaged.

A reference mark transmitter or a very distinct signal flange are suitable ways of ensuring this. The averaging process can be followed either live by displaying the instantaneous values (dynamic averaging) or the result is not displayed until all averaged values have been collected (static averaging). In the case of linear averaging the individual measurements are given equal weight; in other words, they exert the same influence on the mean value.
With exponential averaging this influence varies. The greatest weighting is given to the latest measurements. For this reason, it is particularly suitable for following an unfolding process and observing the influence of different operational quantities.
Peak value averaging, on the other hand, saves the signal value which is numerically the greatest. It is useful for analysing individual results such as shocks or passages of resonance.

C 2.3 Filtering

A distinction is drawn in digital filters between recursive and non-recursive filters. Recursive digital filters are an imitation of the corresponding analogue filters. All the same, in this case slope is not a question, but instead the cut-offs for the pass and stop ranges are defined with reference to the sampling frequency.
Low-pass filters are used, for example, for excluding interference arising from the excitation of the weakly attenuated natural frequency of accelerometers (see figure C.3).
Furthermore, the ripple in the pass range and the attenuation in the stop range must be defined. Non-recursive digital filters cannot be compared with analogue filters. Any frequency responses can be handled with this type of filter.

To avoid large errors when integrating measured accelerations into vibration and displacement, the low frequency cut-off should be set at a value about half the lowest frequency of interest, keeping in mind the frequency response of the applied accelerometer. The basic frequency of the compressor should always be included and for measurements on reciprocating compressor installations the filter should be adjusted to a typical value of 2 to 3 Hz.

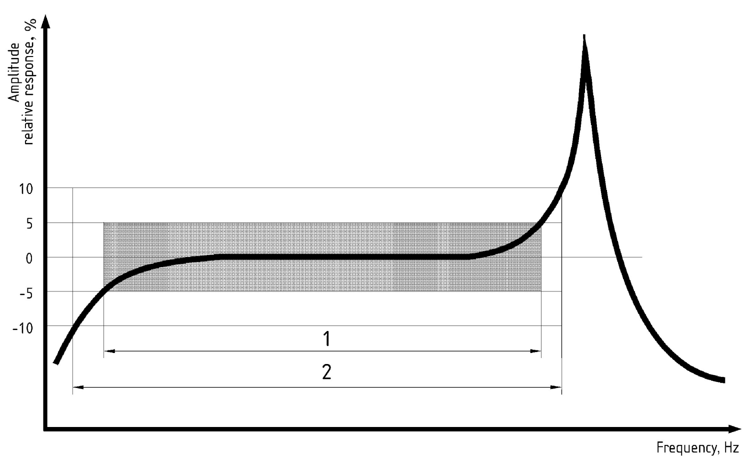


Figure C.3 Acceleration frequency response (type 1 linear range, type 2 non-linear range)

C 2.4 Formation of the characteristic quantities

r.m.s.-values (or peak) values can be obtained directly from the digitized signal mathematically. To prevent fluctuations in digital displays an averaging process can also be connected, as described in Section C 2.1.

C 2.5 Signal display

How the measured and processed signals are displayed is extremely important in diagnostics. There is not just one way of displaying signals but several different ways which have varying degrees of suitability for the various vibration phenomena. They may have identical signal content but still reveal very different information about the physics behind what is happening. Figure C.4 shows one example of two different ways of displaying a signal – in the time range and in the frequency range. Our recommendations and what we must say here about suitability should not be taken as unqualified restrictions. In many cases phenomena, which are hard to distinguish in one form of display are a simple matter to interpret in another.

There are advantages in viewing peak and overall values derived from a time signal, rather than post FFT processing. Consideration of overall and peak, and crest factor (the ratio of peak to r.m.s.) can give important indications of the signal content without recourse to Spectra.

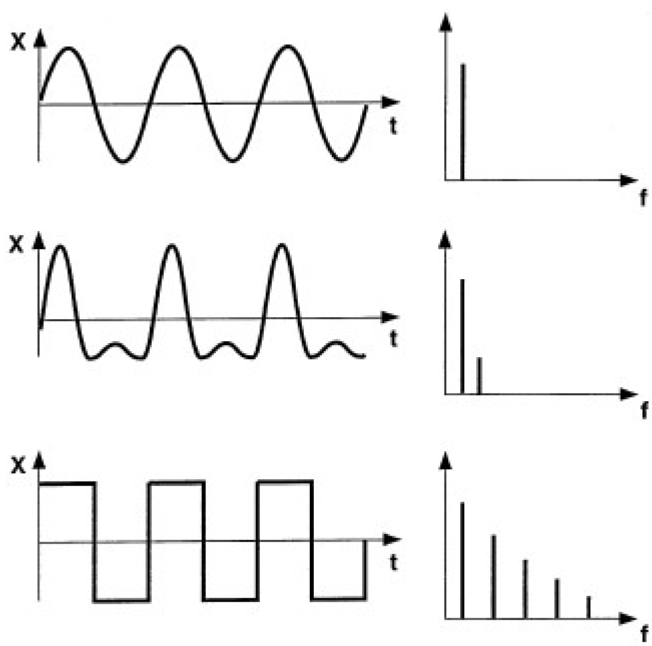


Figure C.4 Displaying measured values in the time range (left) and in the frequency range (right)

C 2.6 Displaying in the time range

When a vibration signal is plotted as a curve against time this reveals whether one is dealing with stationary, periodic, or quasi-periodic beats e.g. parallel running compressors with a difference in speed, see also figure C.5.
This also simplifies monitoring for interference or intermittent connections. Such problems would not be easily detectable from plots of r.m.s. or peak values. If time curves are to be recorded using continuous line recorders these must be able to reproduce accurately oscillations with frequencies up

to multiples of the rotational frequency. Interpretation is made more difficult by averaging temporally diverse vibration patterns.
Recording the time curve of the r.m.s. or peak value of a vibration is recommended when the vibration characteristics change, for example, with compressor load, speed, warming up after starting, or when other equipment is switched on.
In most cases, it will then be necessary to record additional operating parameters with the same recorder. Comparison of the time curves may provide information on the causes of such changes in the vibratory pattern.

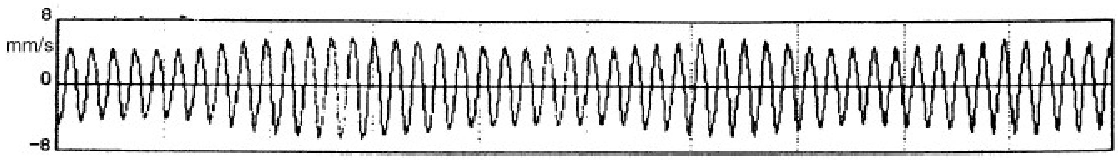


Figure C.5 Time curve for vibration with beating caused by two parallel running reciprocating compressors

C 2.7 Displaying in the frequency range

C 2.7.1 Frequency analysis, frequency diagram

Frequency analyses are a major aid in the interpretation of vibration measurement results. They provide information about the source of vibration excitation and often reveal components of the vibration signal which are important in diagnosis. As can be seen from figure C.6, when a signal is plotted as a time function these components are virtually indistinguishable while in a spectral analysis diagram, on the other hand, they are clearly apparent. In many cases, specific frequency components can be assigned to individual exciters or compressor parts.
Irregularities or damage will result in changes in the spectrum. They can be detected at an early stage and frequently linked with causes. This includes problems such as unbalances of mechanical or thermal origin, self-excited vibration, brushing of compressor parts against each other, changes in alignment, defective bearings and gears, faulty mechanical connections, cracks in shafts and so on.

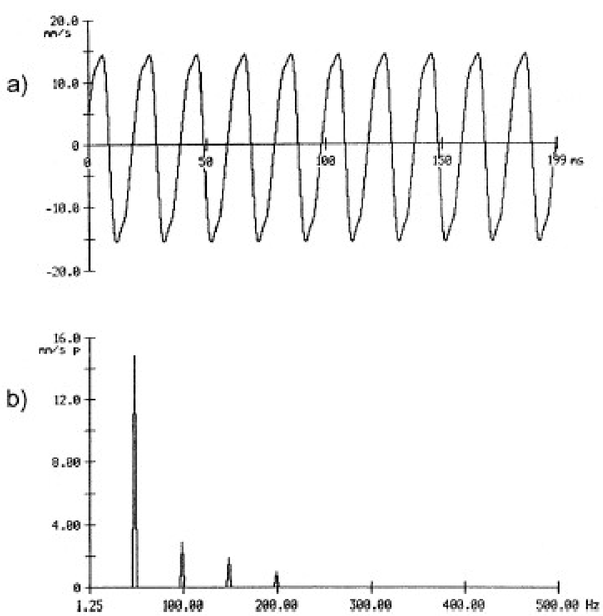


Figure C.6 Small components of the vibration signal are hard to detect in time-curve diagrams (a) but become clear in frequency analyses (b)

Narrow-band analyses can be carried out with various kinds of analogue measuring instrument. For this purpose, they are fitted with a manually or automatically tuneable filter of a defined absolute bandwidth. The amplitudes of the individual frequency components can be displayed or also recorded. Frequency analyses of this type require a period during which the vibration signal must not change. The higher the desired frequency resolution, the longer the time required.
If the measuring devices are equipped with a tracking filter which can be tuned to the rotational frequency or its harmonics, this will allow order analyses to be carried out as well.
Faster frequency analyses can be carried out with digital FFT analysers (fast Fourier transformation), especially when the analysers are equipped with transient buffers for the short-term storage of a signal. FFT analysers display the spectra as a line diagram of limited resolution.
A common figure might be 400 lines for the selected frequency range. This yields a bandwidth for the individual line – for example, 0,25 Hz for the frequency range between 0 and 100 Hz at 400 lines. The consequence of the finite sampling time of the digital device in FFT analysis is the following effect. If the frequency of the input signal does not coincide precisely with a line, this blurs the spectrum – in other words, all adjacent lines will also be less markedly activated.
The only way to exclude this effect is to precondition the signal mathematically by means of a so-called window function. Here it is essentially only the lines directly adjacent which are still present. However, the more these window functions prevent the blurring effect the more they tend to dampen the amplitude of the spectral components.
For this reason, with FFT analysers it should be possible to activate at least one window function which would minimize amplitude distortion (flat top, for example) and another which would ensure optimum frequency (Hanning, Hamming, for example). The two demands are mutually exclusive. It should also be noted that most windows strongly attenuate the signal components at the beginning and end of measurement. One-off occurrences such as shock responses should therefore be processed without windowing but this is normally not required for reciprocating compressor systems.

In general, representation of a spectrum is restricted to plotting the individual amplitudes against frequency. In most cases this provides enough information for an analysis. But it is not until an additional phase spectrum is plotted (the phases of the individual spectral components plotted against frequency) that the signal information is accessible in its entirety. Most modern FFT analysers display the two variables simultaneously. The amplitudes of the individual spectral components can be shown linearly or logarithmically.
Linear display of amplitude has a limited dynamic range. For this reason, only the larger components in the spectrum are visible. In the case of logarithmic display, a dynamic range of 60 to 80 dB is possible. Compared with linear display, much more detail is evident. Which form of diagram is best, will therefore depend on the task to be performed.
Logarithmic display is more suitable for the early identification of problems (crack detection, for example) while linear display is better suited to revealing the origins of dominant frequency components.

The frequency scale, too, can be linear or logarithmic. A linear frequency scale is suitable for a constant filter bandwidth – in other words, the same resolution covers the entire frequency range, such as provided by an FFT/DFT (FFT: Fast Fourier Transformation; DFT: Discrete Fourier Transformation) analysis, for example. Harmonics and side bands appear as equidistant lines and are thus easily recognizable. Using a logarithmic frequency scale means that a very large frequency range in a single spectrum can be presented (3 to 4 decades). However, one disadvantage is that with a constant absolute bandwidth the graphical resolution becomes poorer in the high frequency region. Harmonics and side bands are scarcely recognizable. Table C.2 shows combinations of different possible input, signal and digital processing options:

Table C.2 – Typical signal processing options

Ref.	Input	Raw Signal		Processed e.g. FFT / DFT		
		Time Trace	r.m.s	Spectra	r.m.s.	Crest factor
1	accelerometer	acceleration	m/s^2, g	acceleration	m/s^2, g	Yes
2	accelerometer	acceleration	m/s^2, g	velocity†	mm/s	-
3	accelerometer	acceleration	m/s^2, g	displacement†	mm, micron	-
4	velocity	velocity	mm/s	acceleration‡	m/s^2, g	-
5	velocity	velocity	mm/s	velocity	mm/s	Yes
6	velocity	velocity	mm/s	displacement†	mm, micron	-
7	displacement	displacement	mm, micron	acceleration‡	m/s^2, g	-
8	displacement	displacement	mm, micron	velocity‡	mm/s	-
9	displacement	displacement	mm, micron	displacement	mm, micron	Yes

Key:
† = integrated,
‡ = differentiated,

Annex D General guidelines for troubleshooting in case of unusual, abnormal or excessively high vibrations

D 1 General

High vibrations of a reciprocating compressor system are often caused by improper installation of pipe and vessel supports, loose parts, poor mounting of the compressor to the foundation, poor or no bracing of small lines with heavy masses etc. There are situations in which it is difficult to determine the root cause of the vibrations and for those cases a more extensive performance procedure is necessary. This Annex gives some guidelines how to investigate the source of unusual, abnormal or excessively high vibrations and how to approach such problems.

D 2 Figure 1 — The “Source-and-Effect-Chart”

The Source-and-Effect-Chart explains the basic correlation between the compressor being the source (not necessarily the cause) of the vibrations and its effect on material stress across the compressor system (piping, accessories, instrumentation and the machine itself). The mechanical excitations (loads) are: unbalanced mass forces, crank shaft torsional forces and moments, crosshead guide forces (in vertical direction), gas loads in cylinders (in rod direction) causing dynamic material “stretching”.

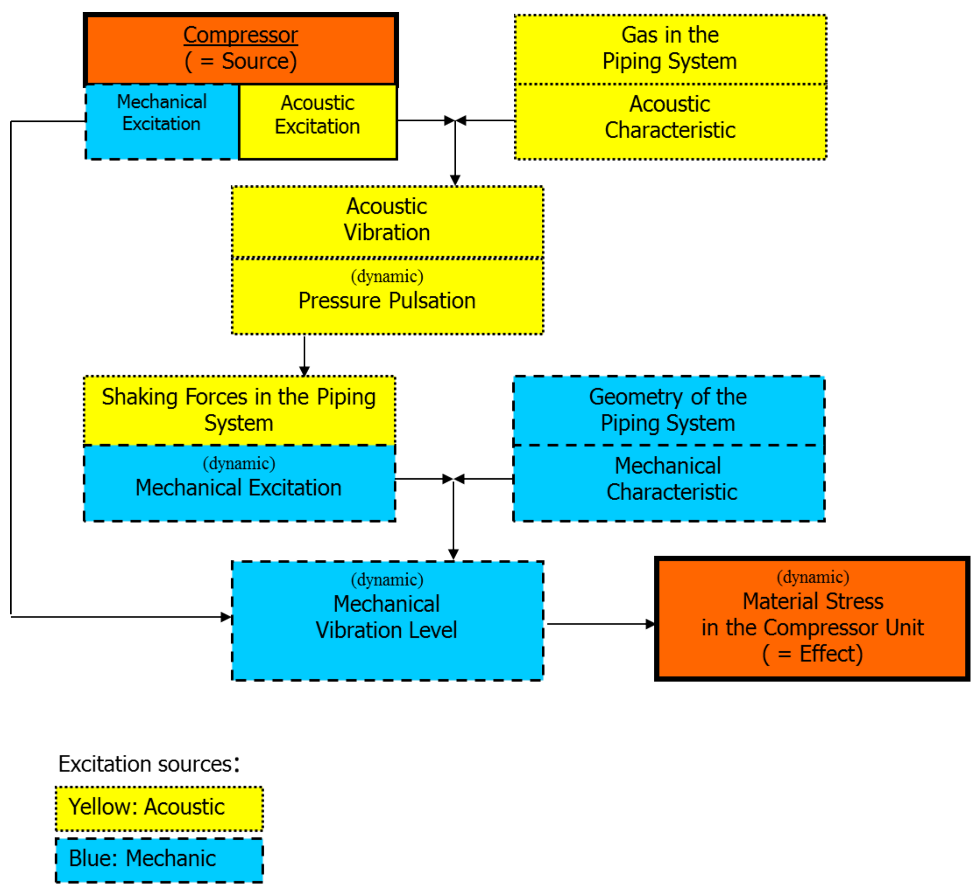


Figure D.1 — Source-and-Effect-Chart

D 3 Step by step procedure to determine the root cause

Without special tools, only the “Mechanical Vibration Level” from the Source-and-Effect-Chart is perceptible. Through with the following step-by-step procedure it is, within certain limits, possible to identify the nature of the vibrations and their cause. The suggested procedure is as follows:

1. Review operating history and vibration data of the compressor system~~;~~ when vibrations have increased and if the condition has changed (such as process and/or load conditions). Any changes to the pipeline, support layout, vessels, etc. is important; if potentially being linked with the vibration occurrences.
2. Check relevant design documents; if a pulsation and mechanical response study according to an applicable standard, e.g. API Standard 618 or ISO 13707, had been performed that may indicate critical conditions for particular process and/or load conditions.
3. Obtain information from end user if frequent compressor valve failures have been encountered. Liquid formation or carry over can be detrimental to the valve performance. Improper valve performance may also be caused through poor design and sizing and/or gas pulsation inside the cylinder gas passage. Valve flutter can be the source for high frequency pulsation which consequentially introduces high frequency vibration into the pipeline.
4. Conduct an initial “walk down” survey of the compressor and the piping to identify locations of abnormal vibrations. Look for the following:
    a. High vibration levels (displacement, velocities and accelerations) and/or high vibration frequencies.
    b. Check the compressor frame for relative movement between:
        - frame base and foundation/grouting.
        - mating surfaces (e. g. cylinder-distance piece, distance piece-crosshead guide and crosshead guide-frame).
    c. Failed piping (cracks or repairs), particularly small bore connection piping of vessels and branch-off lines.
    d. Support structure showing cracks or repairs, signs of damage or which have come loose; missing or broken clamps.
    e. Abnormal noise such as metallic rattling, knocking or hammering that may be indicating some kind of abnormal impact load.
5. Measure vibrations with a mobile vibration meter at points of interest found during the “walk down” survey:
    a. At first only the vibration values, in terms of displacement, velocity and acceleration, are required to obtain an overview of the vibration situation.
    b. Frequency spectra shall be recorded at those locations where vibrations are abnormally high in amplitude or frequency.
    c. The vibration mode shape shall be investigated to detect potential high stress locations (e. g. if vibrations vary significantly along a short distance) and to identify potential mechanical weakness where the stiffness is supposed to be high (e. g. loose connections). To obtain the lower mode shape, a series of vibration values are taken along a straight line.
6. At the time of the vibration measurement it is important to record the actual compressor operating and process conditions.
7. Verify that compressor valve cover temperatures are normal.

8. Verify if the vibrations are related to the load condition as follows:
    a. Loaded, e. g. through suction valve lifters (if installed), or valves removed.
    b. Low load condition, e. g. during start-up- when discharge pressure is equal or close to the suction pressure.
    c. Compressor speed, e.g. determine if vibrations are linked to the compressor speed and its harmonics.
    d. Flow control, e.g. gas recycle, reverse flow control, speed etc.
    e. Process conditions e.g. suction and discharge pressures and temperatures etc.

The following conclusions can be drawn from the "Source and Effect Chart":

A. If vibrations are high during a no-load-condition (e. g. all suction valves lifted) they are likely to be caused from the "Mechanical Characteristic" of the compressor or the pipe and vessel system. This is caused by the fact that a mechanical natural frequency coincides or is close to the frequency of excitation which is called a mechanical resonance condition. An impact test as described in chapter D.4 is then advised as the next activity.
B. If the vibrations vary with load, the vibration response can be a mix of pulsation-induced unbalanced shaking forces (yellow section of the chart in figure D.1), mechanical loads and resonance. Further investigations are then required as outlined below:
    - In pure recycle mode over all stages the actual compressor load is not influenced. The position of the bypass valve changes the bypass flow rate and the acoustical characteristics (yellow section of the chart).
    - If high vibrations can easily be restrained, e. g. through temporary supports for trial purposes (bracing with wooden bars), they are likely to be related to a mechanical resonance. An impact test is then advised.
    - If high vibrations cannot easily be reduced through temporary supports or if vibrations are transferred to other locations across the compressor unit, they are likely to be caused by high forces in the system. If the system is poorly restrained, the dynamic loads can be distributed in such a way that they can create unacceptable dynamic stresses elsewhere which can lead to a fatigue failure. Modifications without detailed analysis may require comprehensive testing to validate.

D 4 Impact Test

Impact testing is a tool to investigate the mechanical natural frequencies and damping of a structure. Impact testing involves hitting the part of equipment of interest, (e. g. with a rubber hammer), and measure the vibration frequency response (the frequency spectrum need to be shown). The frequency peaks reveal the mechanical natural frequencies. This measurement is typically done when the compressor unit is not running. Typical impact test locations to check mechanical natural frequencies are:

- Compressor components (predominantly the cylinders).
- Vessels
- Piping
- Other accessories (e.g. small bore connection piping, drains etc.).

Note: One should be aware that other vibration sources (e. g. pumps, drivers, compressors) could affect the impact test of the compressor system of interest.

In case of other vibration generating machines (compressors, pumps, drivers etc.) that are running and are transmitting vibrations into the relevant investigated compressor system, their influence must be detected. Two impact tests need to be carried out: the first without any impact (bump) and the second with an impact (bump) on the compressor part which is showing too high vibrations.
The comparison of both can show if vibrations peaks for specific frequencies are initiated through the other sources (machines) or from the impact test; thus, being able to differentiate outside excitation from if mechanical natural frequencies and if the latter cause a vibration problem.
If possible, also the time signal shall be recorded to determine the damping ratio from the time signal between vibration peaks and their decreasing amplitude.
If supports and/or clamps had been released or removed with the intention to solve or ease the vibration issue the vibration measurements and impact testing shall be performed for (1) the fixed and (2) released (removed) condition to verify the effect of that modification.

D 5 Measuring the effect of other vibration generating machines on the compressor of interest

Other vibration generating machines (pump, compressors, combustion engines etc.) which are near the compressors, can transfer vibrations to the compressor of interest.
If the measured vibration levels of the compressor of interest exceed the allowable levels and if it is expected that one or more of the other machines have a large contribution on the overall vibrations of the compressor of interest, the contribution of the other vibration generating machines shall be investigated.
This can be done by stopping the compressor or one or more of the other vibration generating machines.
The effect of the vibrations on the compressor of interest generated by all other machines can be investigated by stopping (if possible) the compressor of interest. The effect of the vibrations on the compressor of interest generated by one of the machines can be investigated by stopping (if possible) that particular machine while the compressor is running.
However, this procedure is in many cases not allowed or feasible due to process constraints, safety and integrity reasons of the compressor, machines or complete plant.

D 6 Further investigations

If investigations as outlined above do not indicate the root cause, the following activities may be required:

1. Additional measurements:
    a. Modal analysis (investigation of structure mechanical natural frequencies; mode shapes and damping)
    b. Piping and vessel pulsations.
    c. PV-diagnosis.
    d. Strain gauge measurement to identify the material stress level.
2. Extended engineering:
    Performance of a pulsation and mechanical response study according to the highest level of the applicable standards, e.g. API Standard 618, or revision of the existing study, to match actual measured values with original results. Extend the study scope if actual condition had initially not been investigated.

Finite element models can be used to explore design options to solve a mechanical problem. The models should be correlated with the field data as experience has shown that they are often inaccurate due to wrong or missing boundary conditions.

One should be aware that a modification at a certain location may adversely affect vibrations in other parts of the system. Also, be aware of the effect of variable speed compressors, unallowable vibrations can occur at other frequencies (shift in mechanical resonance conditions).

Attention shall be paid to vibration compensating through secondary masses. A secondary mass, such as a separator level gauge, a cooling water pipe or a large isolation valve, may be excessively vibrating. If the vibrations of that comparably small equipment are restraint and a major part of the compressor unit starts to vibrate instead or even the machine itself, the secondary mass of the level gauge is acting as a vibration compensator which is a clear indication of mechanical resonance.

D.7 Temporary improvements & testing

A very quick and adequate way to test if a modification will work in the field is to solve the vibration issue with temporary wooden supports as follows:

- Apply temporary wood bracing and wood wedges for trial purposes to verify if and how elevated vibrations can be reduced (or not) or if the movement of piping and accessories is only shifted to other locations – or even amplified.
- Such an approach has ever since been quite a practical attempt on the way of fighting elevated vibrations. It is not meant to replace the engineering practice of sound simulations and calculations but often does give hints towards the cause of a vibration occurrence.
- In doubt and/or next to no success all modifications in terms of temporary wood bracing (or others) must be removed to set equipment back to the condition as found when the investigation started.
- If a comprehensive vibration check can prove that a certain temporary support or bracing renders a general reduction of the vibration level, such modification may be converted into a permanent (steel) support. A follow-up vibration check measurement shall then show that the modification is successful. Such action must be communicated with the end user and must be based on his agreement.
- In case of obvious danger from the elevated vibrations the unit must be shut-down.
- In any case the end user must be informed about any finding which he might not yet have become aware of.

Note: The above activities need to be performed with great care and caution including adequate vibration measurement to ensure that the situation is not worse compared with that “before”.

If the wooden beams help, a permanent solution can be designed. The rules of thumb for supports and natural frequency shifting are:

- The stiffness of a support shall be 4 times the stiffness of the equipment (e. g. piping) which needs to be restrained. Simple example: A steel beam made of a 100 mm pipe column (or equivalent H-steel) can hardly restrain a 200 mm pipeline with elevation of some 2 meters above ground level.
- An additional mass to shift a mechanical natural frequency shall be approximately 20% of the equipment mass for which the frequency shifting is intended. Simple example: To shift the

natural frequency of a cylinder with mass of 800 kg at least an additional mass of 160 kg would be required; everything less is not worth trying. Attention: Make sure any additional mass load can be carried by the equipment; check with OEM if not sure.

D 8 Coupling Issues

Frequent or abnormal tear and wear of rubber coupling sleeves

- Excessive rubber sleeve damage can lead to bearing damage either in the motor or of the compressor - including an outboard bearing, if installed.
- The condition of the sleeves can be identified through regular vibration measurements, e. g. with a hand-held vibration meter; including FFT measurement capabilities to see frequency spectra.

In case of repeated coupling damage the true drive train load pattern shall be investigated through:

- Torsional analysis.
- Torsional load measurement with a strain gauge.
- Rod load strain gauge measurements.

D 9 Foundation Issues

Compressor foundations may suffer from long term effects due to:

- Oil contamination.
- Environmental impact (such as ice formation during winter time and/or aggressive atmosphere).
- "Wear and Tear" from normal loads, e.g., dynamic, thermal, etc. causing material fatigue of the grouting and the concrete underneath.
- Deteriorated anchor bolts or grout creep can cause the bolt to lose preload with consequential cracks.

If a compressor frame starts to rock on its foundation, any attempt to prevent movement must be conducted with care as follows:

- In case of oil penetration between frame bottom and foundation top (includes shims, jack screws and other pieces on which the frame may be resting) the oil penetration should be removed e.g. by removing the contaminated part
- In case of visible foundation or grouting cracks or cracked-off material, the compressor frame may suffer from "soft foot". Any evidence of ""soft foot" shall be corrected otherwise anchor bolt torque up may result in unallowable frame twisting.
- In case of cracked anchor bolts investigate the design for load carrying capabilities and proper stretch.

- Prior to any kind of anchor bolt torque-up it is vital to check the anchor bolt material, its true dimension and condition (if corroded) and the preload derived from the required/applied torque (or hydraulic pump pressure) specified in the manual.

  Note: the preload capability of anchor bolts may be reduced due to fatigue, corrosion etc. The actual torque-up must be accompanied by dial gauge measurement between top of frame footing and top and foundation to ensure that frame is not twisted, e. g. due to a. m. "soft foot".

D 10 Mating Surfaces

Unacceptable vibrations of parts of the system in close vicinity of the compressor, or of compressor itself can be caused by too flexible connection of mating surfaces typically “cylinder to distance piece”, “distance piece to crosshead guide” and “crosshead guide to frame”.

Extremely high relative movement of mating surfaces can occur in compressors which are not weather protected or if they are exposed to an aggressive atmosphere e.g. salty air near the sea, content of H_2S e. g. from units nearby or from gas wells etc. This occurs predominantly, but not only, to aged machines. In case water and/or aggressive atmosphere penetrated between mating surfaces, corrosion may occur which means that the mating parts will no longer be able to provide the high stiffness which is required for clamped parts and dynamically loaded fasteners. Checks should be carried out as follows:

- Check the relative movement of mating surfaces which are supposed to be tight together. This should be done by measurement of the displacement on either side of the surfaces which are showing relative movement. If possible the values should be compared with earlier measurements.

- Verify if the relative movement contributes to abnormal vibrations at the outer end of the structure e. g. at the cylinder heads.

- Check if (repeated) bolt cracks have been experienced in the past. In that case the mechanical load on the bolts should be calculated utilizing true process data and should be checked with the compressor vendor.

Annex E Small Bore Connections (SBC)

E 1 General
This annex describes procedures and guidelines for the measurement and classification of mechanical vibrations of small bore connections (SBC).

Small bore connections often refer to a cantilevered pipe appendage with a lumped mass attached to it, such as a relief valve on a pipe nipple, vents or drains on pulsation bottles, or level gages on scrubbers.

Although the vibration of the mainline pipe or vessel may be acceptable, the vibration may be magnified for the SBC, depending on the geometry of the cantilevered attachment. If the SBC is resonant, the vibration can be amplified as much as 30 times higher or more than the process piping. At these high amplitudes, the risk of SBC failure is very high.

The vibration values are defined primarily to classify the vibration of the SBC piping and to avoid fatigue failure. The difference between the highest and lowest vibration velocity level, including the correct phase differences, determines the maximum cyclic stress levels. The guidance values for acceptable overall vibrations are for that reason based on the difference in vibration levels of the two locations as defined in E.2.1 (relative vibration values). The measurements shall be carried out for that reason on two locations.

E 2 Locations and direction of measurements

E 2.1 Location
The difference in vibration measurements shall be taken on the small bore connection at the point of highest vibration (i.e. the anti-node point) and the connection of the small bore connection on the mainline piping. The highest vibration is typically the closed point in the small bore connection (e.g. closed valve, end flange, pressure safety valve, etc.) or at a point in between the mainline piping and the 1st pipe support. The measuring locations for typical small bore connection configurations are shown in Figure E.1. and are as follows:

- 1, 2 measurement locations for a small bore connection mounted at the top of the mainline piping, location 1 is at the top of mainline piping as close as possible at the branch;
- 3, 4 measurement locations for a pressure safety valve connection, location 3 is at the top of mainline piping as close as possible at the branch;
- 5, 6 measurement locations for a small bore connection mounted at the bottom of the mainline piping, location 5 is at the bottom of the mainline piping as close as possible at the branch;
- 7, 8 measurement locations for a small bore connection mounted at the bottom of the mainline piping, location 7 is at the bottom of the mainline piping as close as possible at the branch.

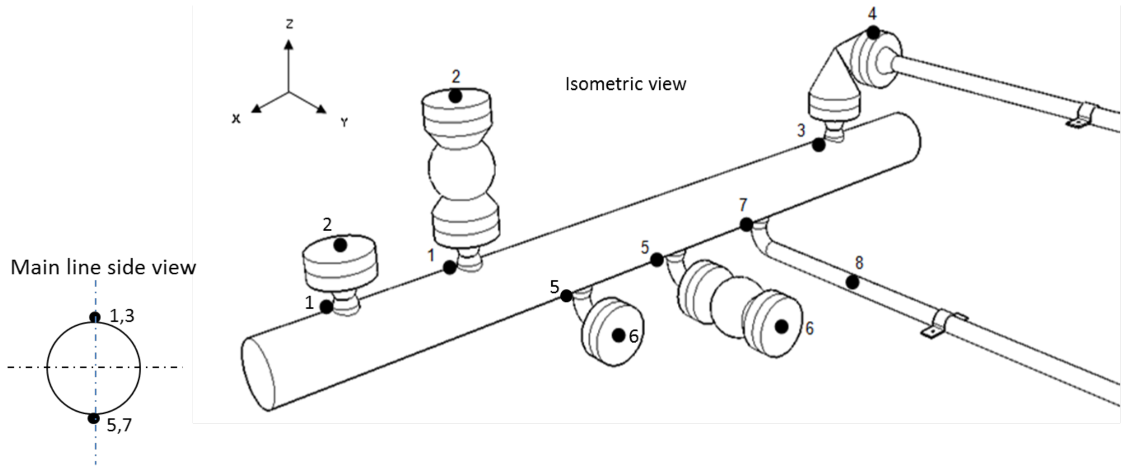


Figure E.1 Measurement locations of several typical small bore connections

E 2.2 Direction of measurements

The measurements should be carried out in three mutually perpendicular X, Y and Z directions.

E 3 Guidance values for acceptable overall relative vibrations for the small bore connections

All measured vibration levels in shall be within the values for acceptable overall relative vibration as summarised in Tables E.1, E.2 and E.3.

The guidance vibration levels from the Tables E.1, E.2 and E.3 are relative values, i.e. the differences in vibrations on the mainline at the branch of the small bore connection and the location of highest vibration of the small bore connection, see Figure E.1.
The correct phase difference between the vibrations of the two measuring points shall be considered. This shall be done by subtracting the time-domain signal of the mainline piping from the small bore connection. This means that both transducers shall have the same (time) reference signal. This can be achieved by using multiple channel equipment and if this equipment is not available, a trigger signal shall be used for both transducers.

Table E.1 – Guidance values for acceptable overall relative vibration displacement values for small bore connections

Compressor system part	**r.m.s. vibration relative displacement values [mm]**		
	Evaluation zone boundary		
	A/B	B/C	C/D
Small bore connection piping	0,202	0,302	0,454

Table E.2 – Guidance values for acceptable overall relative vibration velocity values for small bore connections

Compressor system part	r.m.s. vibration relative velocity values [mm/s]		
	Evaluation zone boundary		
	A/B	B/C	C/D
Small bore connection piping	12,7	19,0	28,5

Table E.3 – Guidance values for acceptable overall a relative acceleration values for small bore connections

Compressor system part	r.m.s. vibration relative acceleration values [m/s^2]		
	Evaluation zone boundary		
	A/B	B/C	C/D
Small bore connection piping	16,0	23,9	35,8

E 4 Curves with overall limits of relative vibration velocity values

The curve with the overall limits of relative vibration velocity values for small bore connections is shown in Figure E.2. The guidance vibration velocity values from Figure E.2 are relative values, i.e. the differences between the vibration velocities including the correct phase difference at two measurement locations as shown in Figure E.1. The correct phase difference between the vibrations of the two measuring points shall be taken into account, see E.3.

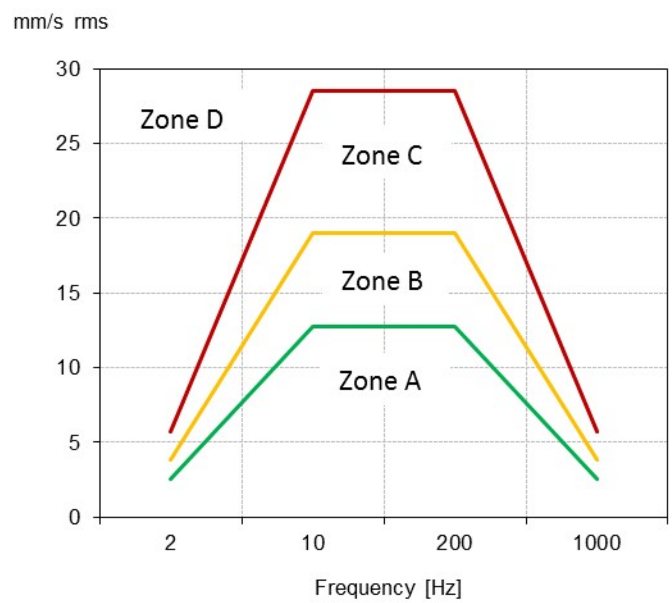


Figure E.2 — Overall relative vibration velocity curve for small bore connection

Annex F: Root-mean-square value, peak value and crest factor

F.1 Introduction

The root mean square (r.m.s.) value as used in these guidelines tends to be a much more consistent measure as sinusoidal data dominate the energy content in this type of calculation.
A disadvantage of the r.m.s. value is that short duration spikes in the vibration signal (especially in the accelerations), which can be audible in nature (knocking), do not have much energy and therefore are not necessarily represented in an r.m.s. measurement. Large-amplitude acceleration spikes also have small vibration displacement and vibration velocity values due to the integration of the measured vibration acceleration signals.
Spikes in a vibration signal can indicate e.g. possible large local stresses in a part of the compressor system or can indicate looseness of internal or external components. The peak value of a vibration signal is for that reason a better measure to detect looseness and to represent the stress since the peak value is proportional to the cyclic stress in the material.
There is not a simple mathematical relation between r.m.s. and peak value; however, crest factors can be a useful way to convert an r.m.s. vibration value into a peak value. The r.m.s. vibration value from these guidelines can be converted to a peak vibration value if the crest factor is measured.
In this annex, root mean square, peak and crest factor values are explained, and can be used to indicate the relationship of r.m.s. vibration value to peak vibration value.

F.2 Root mean square value, peak value and crest factor

F.2.1 Root mean square value

The root mean square or r.m.s. value $\overline{U}$ of a vibration signal is given by:

$$\overline{U} = \sqrt{\frac{1}{T}\int_0^T u^2(t)\,\mathrm{d}t} \tag{F.1}$$

Where:

- $u(t)$ is the time dependent vibration signal;
- T is the averaging time.

NOTE The r.m.s. value defined by Formula (F.1) is also named true r.m.s. value.

F.2.2 Peak value

The peak value $\hat{U}$ of a vibration signal $u(t)$ is the maximum value during a specified time interval.

NOTE 1 The peak value of a vibration is usually taken as the maximum deviation of the vibration from its mean value. A positive peak value is the maximum positive deviation and a negative peak value is the maximum negative deviation.

NOTE 2 One should be aware that other conventions for peak values are used which are not in accordance with International Standards, like:

— True Peak (tP) [also named peak (p) or zero-to-peak (0-p)] is defined as the difference between the maximum and minimum value of a vibration signal during a specified time interval divided by 2 and in this respect the True Peak definition differs from the peak value definition in ISO 2041.

— Pseudo Peak (pP) [also named calculated peak (cP) or derived Peak (dP)]: pP = $\sqrt{2}\ \overline{U}$. Pseudo peak is based on the conversion between peak and r.m.s. for a single sine wave. While Pseudo Peak has no direct mathematical relationship with a complex waveform, it is often used as a quick reference between r.m.s and peak. Such relationship can be useful but one should be aware that a Pseudo Peak value can be smaller (e.g. triangle, saw tooth), equal (pure sine) or larger (e.g. square, block) than a True Peak value.

F.2.3 Crest factor

The crest factor C_F of a vibration signal is the ratio of the peak value $\hat{U}$ to the r.m.s. value $\overline{U}$:

$$C_F = \frac{\hat{U}}{\overline{U}} \qquad \text{(F.2)}$$

NOTE 1 One should be aware that several other definitions of crest factor can be used, e.g. more than one (true) peak value is commonly applied.

NOTE 2 Crest factors based on true peak values typically range from 2 to 4 based on measurements for reciprocating compressor systems if minimal impacting is occurring. They can be several times larger if objects are responding to strong impact forces. Other types of positive displacement compressors or pumps can have larger crest factors.

NOTE 3 Objects, that are dominated by a pure tone, (e.g. small bore attachments) resonance near a sinusoid crest factor of $\sqrt{2}$.

Annex G: Measurement of vibration values on the crosshead guide

G.1 General

Accelerometers are often mounted on the crosshead guide for condition monitoring purposes of internal compressor components. The vibration is measured in the direction of the force exerted by the crosshead on this guide, which is in the vertical direction of a horizontal compressor. Experience on horizontal compressors has shown that the vibration values measured on the crossheads guide can be used in addition to the vibration values of other locations to judge the integrity of the compressor. The procedures for measuring the vibration values on the crosshead guide are outlined in this annex and are limited only to horizontal compressors.
Crosshead guide impact measurements (typically as vibration acceleration) are subject to mounting, compressor loading, speed, rotational direction, transducer type, structure resonance, and sampling techniques. Therefore, it is often used as a trending tool with ALARM and TRIP (shutdown) limits determined from normal running conditions. The limits provided below are interpolated from the standard guidelines from 2 Hz to 1000 Hz and can be used for guidance, but can be subject to change depending on the actual conditions and design of the machine.

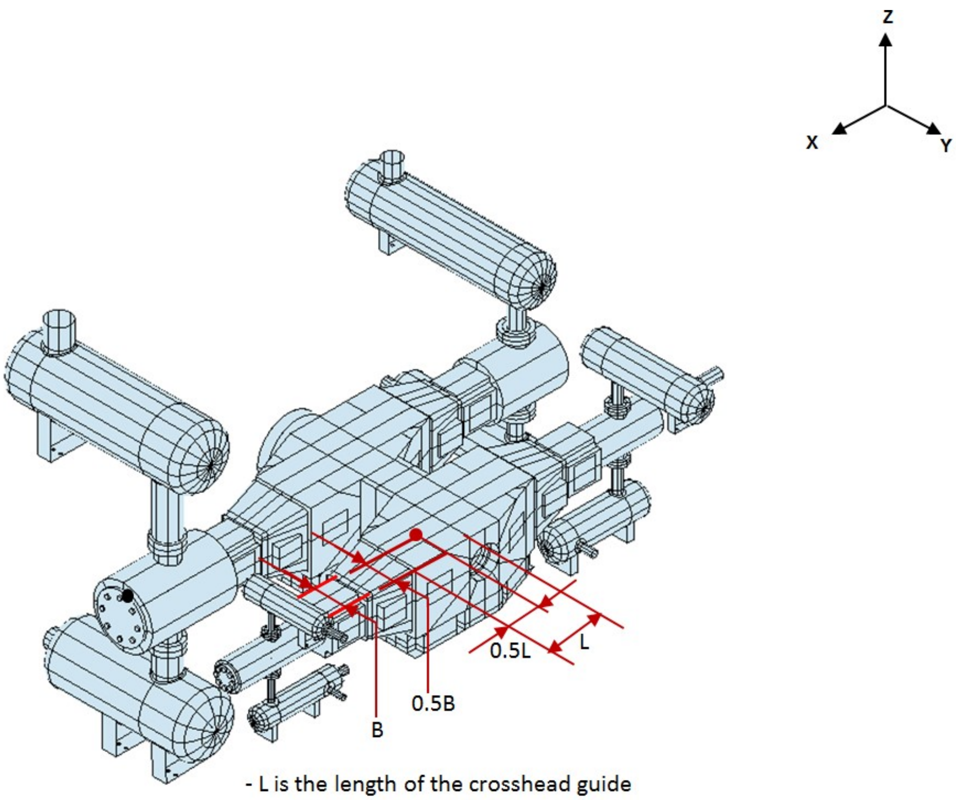
G. 2 Locations and direction of measurements

G.2.1 Location

Each crosshead guide for horizontal compressors as follows: on the marked location indicated by the compressor OEM. If there is no marked location, on top in the centreline of the throw halfway along the length of the crosshead guide, see Figure G.1. This location needs to be a stiff structural part which might need verification on the OEM machine drawings.

G.2.2 Direction of measurements

In the direction of the force exerted by the crosshead on this guide which is the vertical direction of a horizontal compressor.



NOTE The measurements should be carried out on the crosshead guide of each throw.

Figure G.1 — Crosshead guide locations on a horizontal compressor

G.3 Guidance values for acceptable overall vibrations for the crosshead guide

The vibration displacement limits are generally not used on crosshead guides and for reason only the vibration velocity and vibration acceleration levels have been given.
Unless the compressor manufacturer has guidance values for the vibration velocity and vibration acceleration levels for the relevant compressor, the guidance levels of Table G.1 Table G.2 can be used.

Table G.1 — Guidance values for acceptable overall vibration velocity values of the crosshead guide

Compressor system part	**r.m.s. vibration velocity values horizontal compressor [mm/s]**		
	Evaluation zone boundary		
	A/B	B/C	C/D
Crosshead guide	6,0	9,0	13,5

Table G.2 — Guidance values for acceptable overall vibration accelerations values of the crosshead guide

Compressor system part	r.m.s. vibration acceleration values horizontal compressor [m/s^2]		
	Evaluation zone boundary		
	A/B	B/C	C/D
Crosshead guide	7,5	11,3	17,0

G.4 Curve with overall limits of vibration velocity values

The curve with overall limits of vibration velocity values for the crosshead guide of a horizontal compressor is shown in Figure G.2.

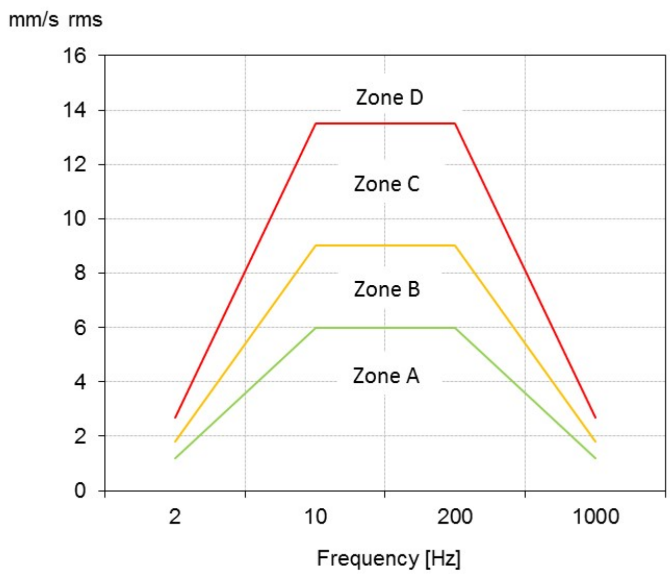


Figure G.2 — Overall vibration velocity limiting curve for the crosshead guide of a horizontal compressor

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