# Causes, assessment and reduction of piping vibrations

Dipl.-Ing. Robert Missal
KÖTTER Consulting Engineers KG, Rheine
Germany

**Abstract**
The assessment of piping vibrations frequently presents an insolvable task for the user of piping systems. Plain and applicable directives, somewhat comparable with the approximate data for the assessment of machine vibrations, are non-existent because of the major differences in the installation and application conditions. On the basis of plain models, an option for estimating the vibration-related stresses in piping systems is portrayed. Causes and reduction measures related to piping vibrations are presented and discussed on the basis of practical examples.

**CV**
After his graduation Robert Missal studied mechanical engineering at the Fachhochschule Münster. Subsequently, he worked for one year as assistant in the laboratory for thermodynamics. Thereafter, he was employed as constructing engineer by a manufacturer of wind- and waterpower plants. Since 1987 he works at KÖTTER Consulting Engineers KG (formerly KÖTTER Beratende Ingenieure) as consulting engineer and specialist for worldwide troubleshooting missions in the field of technical acoustics and machine dynamics. He is responsible for his department as senior consultant since 1992. Mr. Missal also works in different standards and VDI-directives committees.

**Contact**
Dipl.-Ing. Robert Missal
Devision Machinery and Plants
KÖTTER Consulting Engineers KG
Bonifatiusstraße 400
48432 Rheine
Germany
Tel. +49 - (0) 5971 - 9710.25
Fax. +49 - (0) 5971 - 9710.50
E-Mail: missal@koetter-consulting.com
www.koetter-consulting.com

## 1. Introduction

Vibrations in piping systems are not an unusual occurrence in industrial plants. During the period of operation, users of these plants have frequently learned to assess and to accept these vibrations on the basis of experience values.

However, with new plants the user is confronted with the problem that these experience values are non-existent. This is the case during start-up when the first alarm indications occur, such as subjective major vibrations, noises caused by secondary effects, screws falling out and similar. An objective assessment of the situation is then required.

The description of the excitation and transmission mechanisms and the assessment of the situation is possible with the measurement of piping, machine and foundation vibrations as well as pressure pulsations in the pipe. In addition, effective reduction measures for improving the situation can be processed on the basis of the available measuring values.

## 2. Cause of piping vibrations

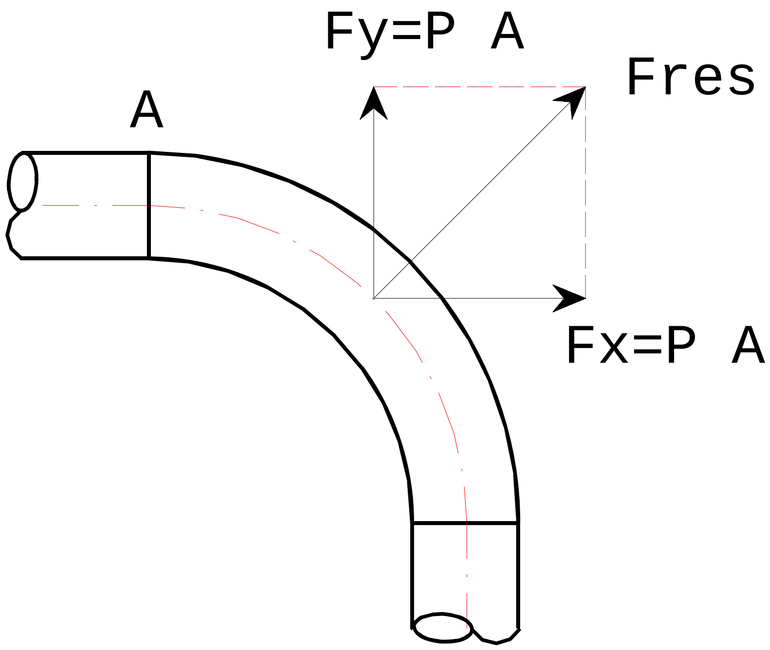
The cause of piping vibrations can be attributable to the mechanical coupling of vibrating components such as motors, compressors or the foundation. Then again, a vibration excitation can also occur as a result of pulsating media within the piping itself.

The rigid coupling to vibrating components is frequently encountered in practice (e.g. pneumatic control lines). These are rigidly attached to vibrating parts such as valves and piping so that there is an direct vibration excitation in this case. In addition, the distance between supporting locations is relatively large where control and measuring lines are concerned. These lines frequently break off when excitation frequency and natural frequency coincide.

The vibration excitation caused by pulsation in the medium is attributable to the discontinuous output of many compressors. Particularly with reciprocating compressors, the operation of the compressor produces pressure pulsations with amplitudes of more than 2 % of the static pressure in the piping. If the excitation frequencies caused by the compressor coincide with the acoustical resonance of the connected piping system, the pressure pulsations are additionally increased.

The pulsations generate forces at piping internals, pipe elbows etc. which frequently and excessively exceed the static requirements of piping supports.

Figure 1: Force effect with redirection through a 90° bend




A further possibility of vibration excitation can occur as a result of pressure fluctuations in the medium, as caused for example by periodic vortex shedding. As a rule, these vibrations are high-frequent, meaning, they are above 500 Hz, so that the movement of the piping is very small with these frequencies.

## 3. Measurement and assessment of piping vibrations

The measurement of piping vibrations can be difficult with subsurface piping or with piping for hot media. As a rule, however, it is not an insolvable problem. The vibrations can be relatively easily measured with the help of vibration sensors which are attached to the piping from the outside by means of an adhesive method, with a threaded union or with the help of a magnet.

The dominating frequencies can be determined by filtering of the vibration signals and analysis with the help of the Fast Fourier Transformation.

In general, the measurement of piping vibrations is not a problem. However, the assessment of the measured values is difficult. Standards and directives as known for example from the field of machine vibrations (VDI 2056 [1], DIN ISO 10816 [2]), are non-existent for the assessment of piping vibrations. Moreover, the approximate values of these standards just cannot be simply transferred to the sphere of piping vibrations because the type and number of the vibration-endangered components as well as the type of the vibration excitation are not comparable.

Nevertheless and in order to provide a quick and applicable assessment basis, some machine manufacturers have extended the VDI 2056 (withdrawn) by certain groups, according to which piping vibrations can be assessed. Such a manufacturer directive is the NEN 7-4-1 [3] of the reciprocating compressor manufacturer Neuman & Esser. For the machine group D „reciprocating machines on high-tuned solid foundations, piping rigidly secured to these machines", this directive specifies the following assessment stages:

Table 1: Assessment stages of the NEN 7-4-1, machine groups D and T

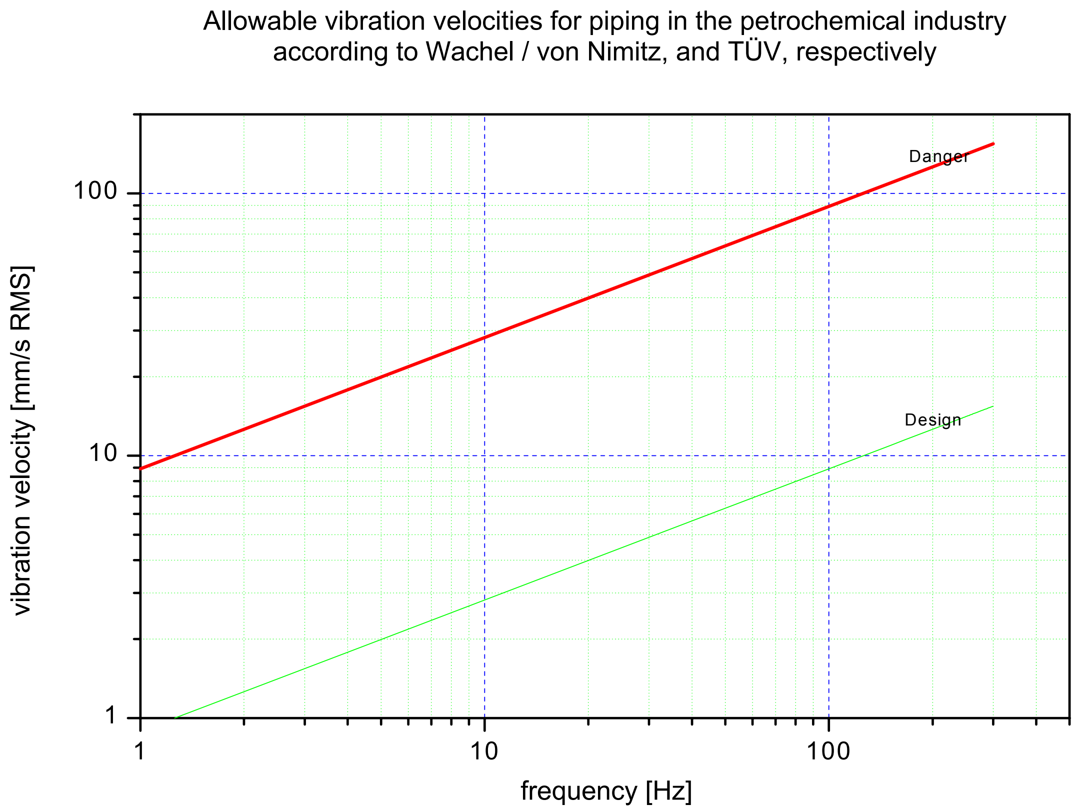
| assessment stage | vibration velocity in mm/s eff. | |
|---|---|---|
| | group D | group T |
| good | ≤ 4,5 | ≤ 2,8 |
| useable | ≤ 11,2 | ≤ 7,1 |
| still acceptable | ≤ 28 | ≤ 18 |
| not acceptable | > 28 | > 18 |

The machine group T refers to *„larger machines with only rotating masses and low-tuned foundations, pipelines in piping systems, where it is irrelevant whether connected to reciprocating, screw-type or centrifugal machines".*

For the assessment of the piping vibrations, the measurement of the vibration velocity as a sum value in a frequency range of 5 to 1000 Hz can only provide a rough orientation. That particular factor, which is generally responsible for a piping breakaway, is the stress in the piping wall. This stress is given by the static internal pressure in the piping, stresses resulting from the installation process, and additional stresses caused by piping vibrations.

In [4], values for allowable piping vibrations are stated which are based on experience from the petrochemical industry. Here, deviating from the above-mentioned directives, an assessment of the frequencies of the vibrations is made.

Figure 2: Allowable vibration velocities as a function of the vibration frequency [4]




The stresses additionally caused by piping vibrations depend on the material, the geometrical dimensions of the piping, the support and/or the fixation conditions of the piping as well as the deflection of the piping. The following equation applies for the vibration displacement, meaning, the deflection caused by vibrations:

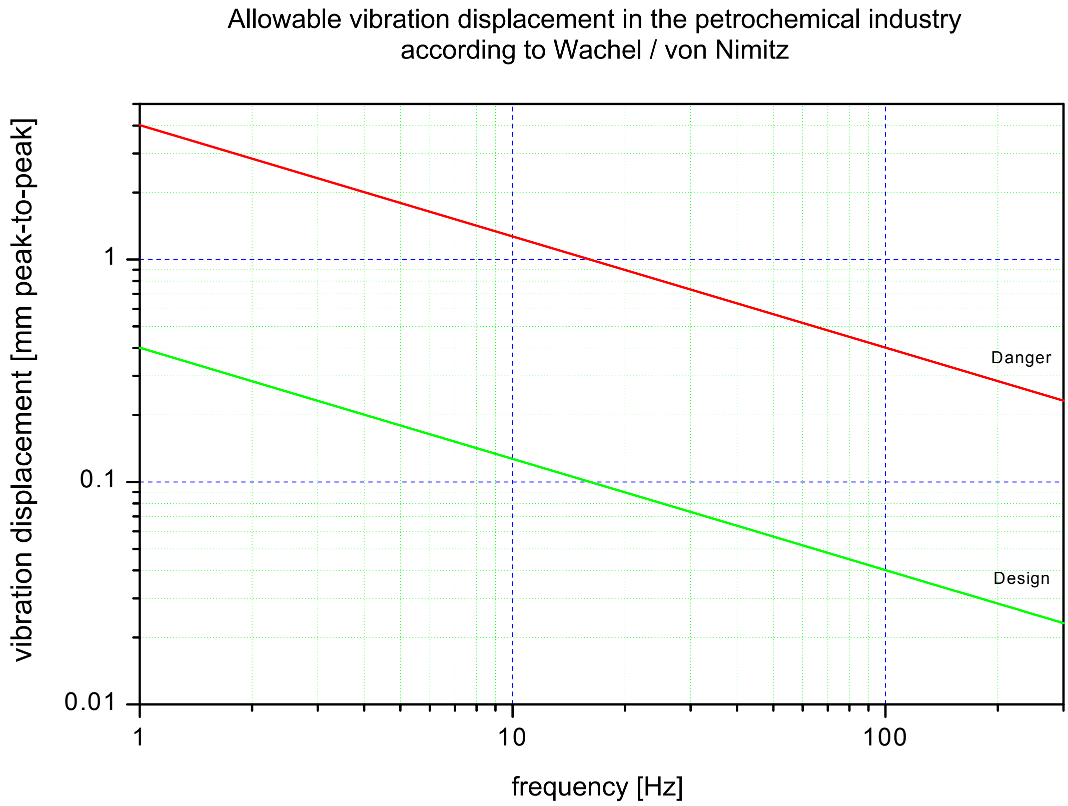
$$s = \frac{v}{2 \cdot \pi \cdot f} \tag{1}$$

with

| | | | |
|---|---|---|---|
| s | = | vibration displacement | [m] |
| v | = | vibration velocity | [m/s] |
| f | = | vibration frequency | [Hz]. |

On the basis of figure 2, therefore, also a presentation of the maximum vibration displacement as a function of the frequency is common practice.

Figure 3: Allowable vibration displacement as a function of the vibration frequency [4]




Piping vibrations lead to material fatigue particularly in such situations where the natural frequency of the piping and/or of the piping sections coincide with the excitation frequency. Subsequently and in this case, the stresses in the piping can be estimated as follows (the derivation of the formula can be seen in [4]):

$$\sigma_{max} = K_d \cdot s \cdot \frac{D}{L^2} \cdot SCF \qquad (2)$$

with

| | | | |
|---|---|---|---|
| $\sigma_{max}$ | = | maximum stress | [psi] |
| $K_d$ | = | deflection stress factor | [ - ] |
| s | = | maximum vibration displacement | [mils] |
| D | = | outer diameter of the piping | [in] |
| L | = | length of the piping | [feet] |
| SCF | = | stress concentration factor | [ - ] |

The values for $K_d$ are dependent on the piping routing. For different types of piping and for each first and second natural frequency in this case, the factors for various length conditions have been calculated and graphically plotted with the Finite-Element-program ANSYS.

If the maximum allowable vibration velocity is derived from, the result is the calculation equation for determining the maximum stress

$$v_{\max} = \frac{\sigma_{allow}}{K_v \cdot SF \cdot SCF} \qquad (3)$$

with

| | | | |
|---|---|---|---|
| $v_{max}$ | = | maximum vibration velocity at the piping | [in/s] |
| SF | = | safety factor | [ - ] |
| SCF | = | stress concentration factor | [ - ] |

For a material with an allowable stress of $\sigma_{allow}$ = 13000 psi, a $K_v$ of 318, a safety factor of 2 and a stress concentration factor of 5, the following results for the maximum allowable vibration velocity in the resonance

$$v_{max} = 4 \text{ in/s} \simeq 102 \text{ mm/s} \quad 0 - p$$

This estimation shows that, even with these high vibration velocities, stresses of only 90 N/mm$^2$ (13000 psi) are to be expected. The values of table 1 therefore, are to be categorised as being very conservative.

The calculation performed above applies only for an excitation of the piping in the range of the natural frequency. If excitation and natural frequency do not coincide, an additional correction variable must be included.

A similar course for the assessment of piping vibrations is described in [5]. The formula is stated as follows for the calculation of the maximum stresses caused by vibrations:

$$\sigma_{max} = f_M \cdot f_\sigma \cdot f_\Phi \cdot v_{max} \cdot r_a \cdot \sqrt{\frac{E \cdot \mu}{I}} \qquad (4)$$

with

| | | | |
|---|---|---|---|
| $\sigma_{max}$ | = | maximum stress | [N/m²] |
| $f_M$ | = | correction factor for the detection of single masses | [ - ] |
| $f_\sigma$ | = | stress increasing factor | [ - ] |
| $f\phi$ | = | natural form characteristic value | [ - ] |
| $v_{max}$ | = | maximum vibration velocity | [m/s] |
| $r_a$ | = | radius of the piping, extern | [m] |
| E | = | elasticity module of the piping | [N/m²] |
| $\mu$ | = | mass occupancy of the piping | [kg/m] |
| I | = | surface moment | [m$^4$] |

With the introduction of correction factors which consider the properties of the piping, an attempt is made here also to draw conclusions on the stresses caused by vibrations.

The assessment is then made by comparing the calculated stresses with the allowable stresses for the individual material, application conditions and the load case. With the methods described here, the stress in the piping is adopted in each case as an assessment quantity. Subsequently, the significant question is this: why not measure the stresses directly.

Stresses are normally measured in a component with the use of strain gauges. The strain gauges are adhesively applied to locations on the piping where the highest stresses are expected. However, the sticking and wiring of the strain gauges is very time-consuming. For this reason and in general terms, the location of the piping is to be determined at which the highest stresses occur. This is performed with for example orientating vibration measurements and theoretical consideration.

If these locations are known, the situation can be detected and assessed by means of a measurement of the actually occurring stresses during plant operation. It is particularly the measurement of the actual stresses in the individual case which provides the exact result.

## 4. Reduction of piping vibrations

Exact knowledge of the excitation and transmission mechanisms is required for the refurbishment of the vibration situation in piping. If the piping vibrations are subject to the coincidence of acoustical or structure natural frequency with the excitation frequency of the connected machine as a first step, an improvement of the situation can be obtained by means of shifting of the resonance frequency.

The structure natural frequency of a piping can be changed by attaching additional supports. The support must have a sufficient rigidity here because no adequate shifting of the natural frequency can be otherwise obtained. A further measure could be, for example, additional supporting and/or relocating of armatures and valves which can have a considerable influence on the local natural frequency due to their concentrated mass.

If acoustical natural frequencies are responsible for the piping vibrations, the pulsation in the piping can be reduced by means of

- change of the piping lengths or
- change of the piping internals - e.g. relocating of orifices - or
- application of additional pulsation-reducing measures.

Such a measure can be, for example, the installation of pulsation-damping-plates according to the KÖTTER-principle. With the installation at a previously calculated location in the piping system, acoustical resonances can be destroyed and pulsation reductions of up to 95 % can be realised. The pulsation-damping-plate has been optimised over the past 15 years from acoustical and flow-technical aspects. For this reason, the remaining pressure loss is significantly less compared to a conventional orifice with a comparable acoustical effect.

## 5. An example from practice

Excitation in the range of the mechanical natural frequency

In a reciprocating compressor plant, piping vibrations at pipes between the 1. and 2. stages have been the subject of complaint. Measurements of pressure pulsations and piping vibrations confirm at first the high vibration level at local piping sections.

Figure 4: Time sequence of the pressure pulsations in the suction line (1. stage)

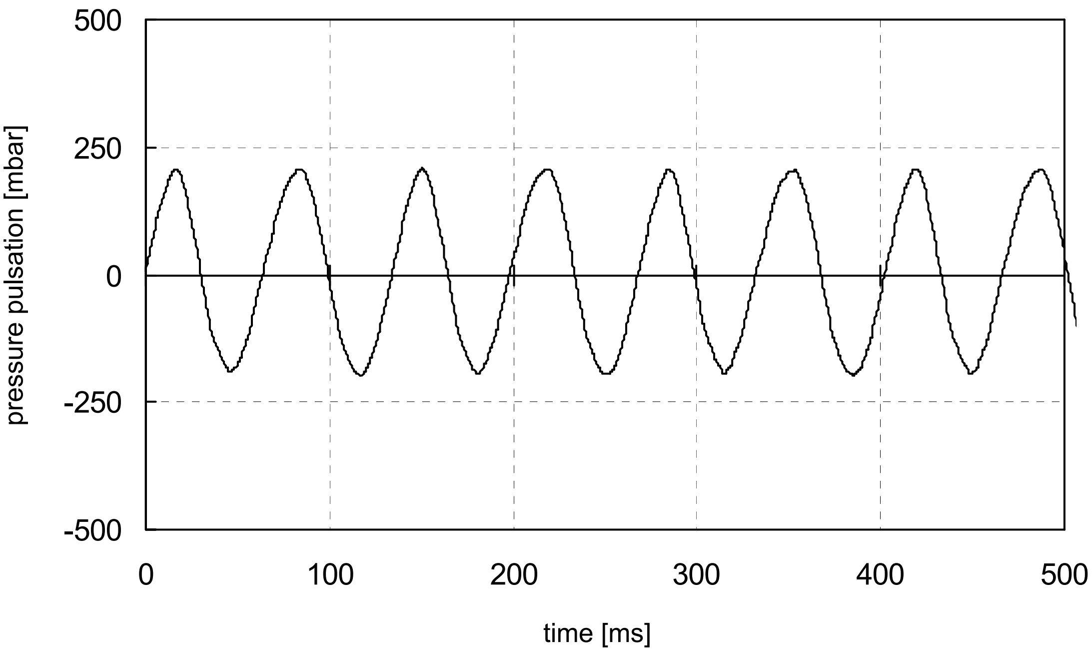
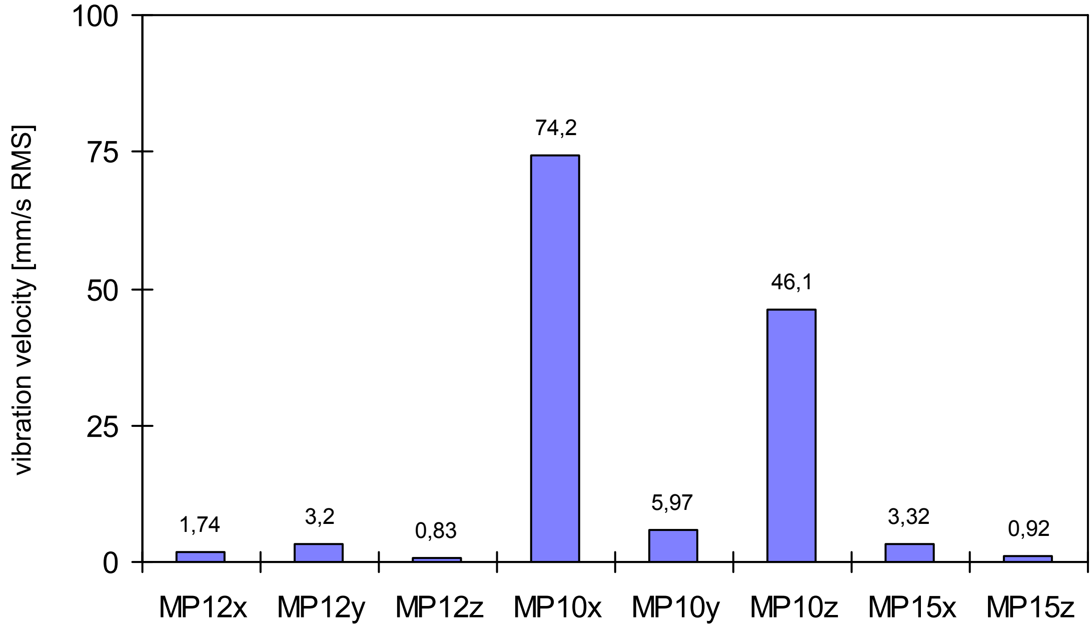
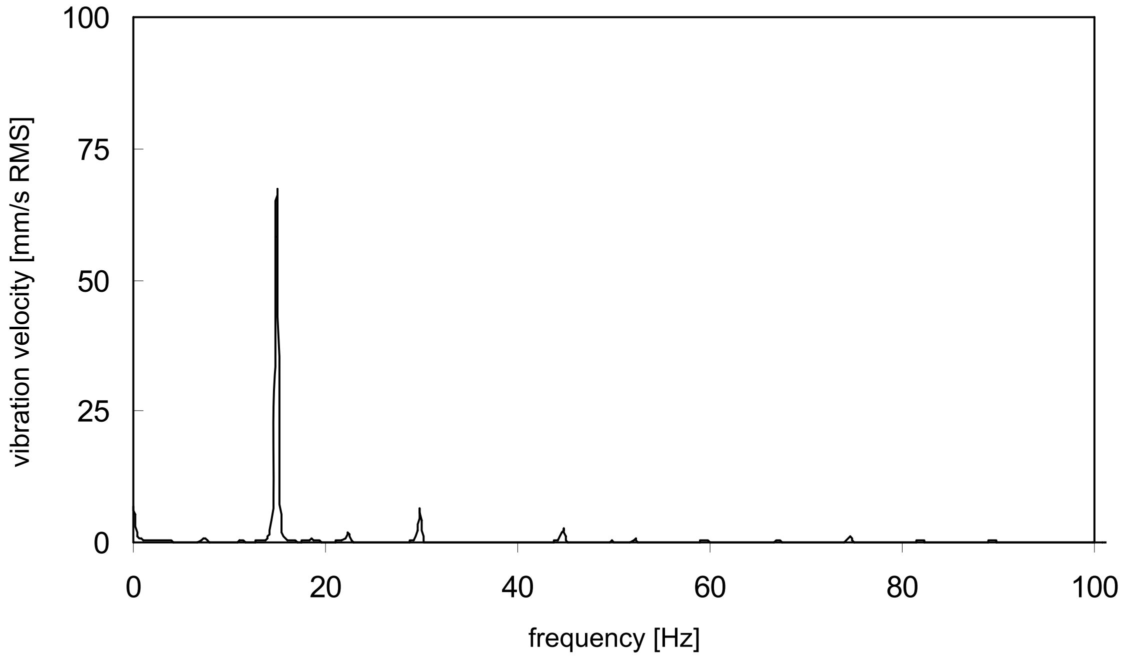


Figure 5: Vibration velocities (root-mean-square values) at the piping at various measuring points



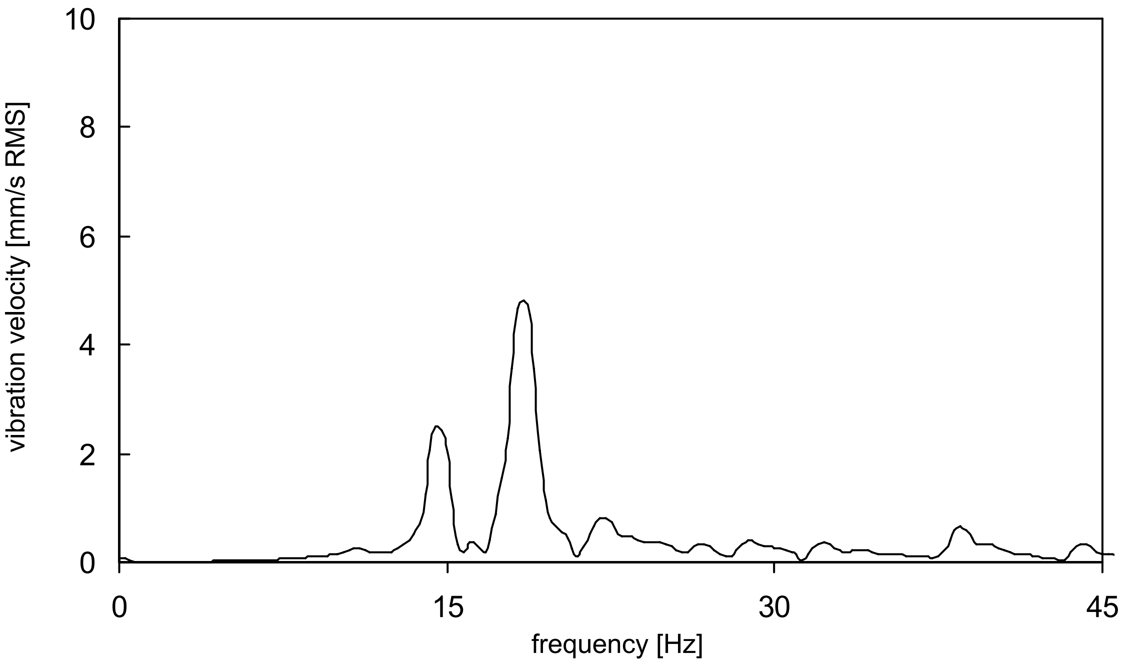

A dominating frequency of approx. 15 Hz can be seen in figure 4. The speed of the reciprocating compressor is 446 rpm. As the cylinders are double-acting, there are 2 pressure surges per rotation, meaning 14.9 Hz. The pressure pulsations lead to vibrations at the lines which lead to considerable vibrations at the measuring point 10 in x- and z-direction (figure 5) in particular. The frequency of the vibration is also 15 Hz.

Figure 6: Spectrum of the vibration speed at the measuring point MP 10 in x-direction




It can be derived from the FFT-analysis of the vibration speed (figure 6) that the vibration of the line in this direction is practically mono-frequent. Impact tests have shown that the natural frequency of the line lies in the range of the excitation frequency of the reciprocating compressor.

Figure 7: Spectrum of the vibration at the piping in the zone of the measuring point 10 with an impulse excitation and with the reciprocating compressor shut down



As the vibration increase in this case was attributable to the coincidence of structure-mechanical natural frequency of the piping and excitation frequency of the reciprocating compressor, an additional support in the zone of the measuring point MP 10 was proposed as a reduction measure. As an alternative to this, a reduction of the excitation, meaning the pressure pulsations in the line - e.g. by means of pulsation damper plates - would also have led to the objective. However, the belated installation of the support involved a substantially less work effort and this reduction measure was favoured.

## 6. Literature


[1] VDI 2056, October Edition 1964
Assessment scales for mechanical vibrations of machines (withdrawn)

[2] DIN ISO 10816
Evaluation of vibrations of machines by means of measurement at non-rotating parts, Part 1 to 6

[3] NEN 7-4-1
Extension VDI 2056, Groups T, D, S

[4] Wachel, Morton, Atkins
Piping Vibration analysis
Proceedings of the nineteenth turbomachinery symposium

[5] Bietenbeck, Petruschke
Assessment of piping vibrations on the basis of vibration speeds