The Vibration Absorber ‘Magic Tube’– Different Applications on Site

Johann Lenz, Henning Ledendecker and David Al Mouallem

February 14, 2023

The vibration absorber 'Magic Tube'–
- Different applications on site-

Dr.-Ing. Johann Lenz; Henning Ledendecker, M.Sc.
Machinery and Plants
KÖTTER Consulting Engineers GmbH & Co. KG
Rheine, Germany
j.lenz@koetter-consulting.com

David Al Mouallem, AMIOA
ADF - Acoustic Design Firm
Doha, Qatar

Abstract

In several industries increased vibrations often occur at pumps, compressors or turbines and at the connected piping systems. These vibrations can have different causes. Different measures can be used to decrease this vibration level.

In many cases additional supports are used to tighten the piping. But in some cases, it is not possible or very expensive and inconvenient to install additional piping supports or damping devices. Another very effective measure is the use of vibration absorbers[1]. Especially vibration absorbers with an implemented damping characteristic enable a significant optimization of the vibration characteristic in a broad frequency range.

In this paper the vibration absorber Magic Tube is presented. The vibration absorber is adjustable on site in a wide frequency range with different mass configurations. The design is introduced, and a case study shows the vibration reduction effect in the field.

1 Introduction

Resonances occur when a vibratory system is excited in its natural frequency. Often even a slight excitation – e.g., pulsations in piping – is sufficient to evoke critical dynamic stress of the structure.

A possible measure to eliminate resonance problems is the detuning of the system. By changing the stiffness or the mass, the natural frequency is shifted. However, to be effective, these measures have to be individually planned, manufactured and installed. That costs time and can lead to production losses and high costs at critical plant conditions.

KÖTTER Consulting Engineers (KCE) took this time and cost factor as a challenge to develop a solution for operators that can be implemented in a short time – even during a measurement: The patented device[2] is a variable two-dimensional vibration absorber that can be universally configured on site.

2 Usage of vibration absorbers

There are several different measures for vibration reduction. The most effective solution always depends on the root cause. In case of resonances with low damping (often at pipelines) a vibration absorber enables an effective vibration reduction. Vibration absorbers are not only used for industrial applications but also in everyday life, for example to improve unwanted vibrations at cabriolet roofs or to protect bridges and buildings against storms and earthquakes.

For analysing the dynamic effect of a vibration absorber, a dynamic model with two degrees of freedom is usually used, see figure 1.

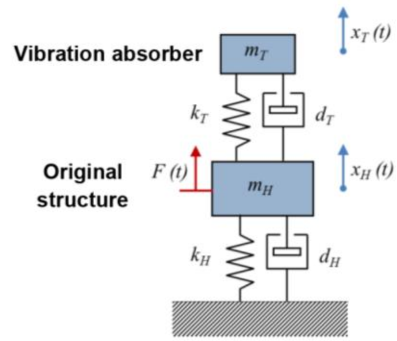

Figure 1: Principle dynamic system consisting of the original structure with a coupled vibration absorber.

The original structure without the coupled vibration absorber - as an example for various dynamic structures - has got one degree of freedom. If the system is excited harmonically with its natural frequency, resonance occurs. The resonance case (see dominant blue peak in figure 2) often leads to enhanced vibrations, which may end in cracks or other damages.

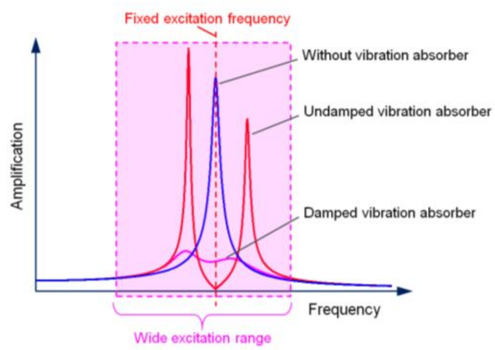

Figure 2: Response characteristics of an original structure and the modified behaviour with a vibration absorber.

In case of a fixed excitation frequency, additional stiffness or mass enables the tuning of the natural frequency and therefore it is a common measure to improve the dynamic behaviour. Another possibility with a nearly total compensation of single-frequency vibrations is the use of an undamped vibration absorber. An undamped vibration absorber is able to compensate the excitation forces by a counterforce. The result is visible in figure 2 (red curve). With an undamped vibration absorber, the response of the original structure is almost eliminated but only at one fixed frequency.

Today many machines are changing their operating speed in dependency of the operating conditions. Thus, there is a wide range of possible excitation frequencies.

In this case, tuning of the system often only changes the critical operating point but does not solve the resonance problem. The use of an undamped vibration absorber also leads to two new resonance cases. Therefore, other modifications are necessary.

One very effective measure in case of low dampened resonances is the use of a damped vibration absorber. In contrast to the undamped configuration of a vibration absorber no further dominant resonance frequencies of the new dynamic system occur. The typical optimized response behaviour is shown in figure 2 (violet curve). This absorber enables the machine operation in a very wide range without further restrictions.

3 Design of the vibration absorber Magic Tube

The design is made for high flexibility. The focus is on multiple parameter variation during the vibration investigation on site. All functionally relevant components are placed in an encapsulated casing, see figure 3.

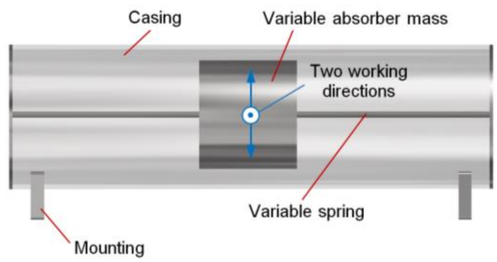

Figure 3: Design of the vibration absorber Magic Tube.

The main components of the absorber are the variable mass and the connected spring with its tuneable stiffness. Both are integrated in the casing that can be connected to any critical structure by changeable mountings.

The on-site installation can be divided in three main steps. At first, a suitable absorber mass is selected depending on the vibrating structure. Then the natural frequency of the absorber system is set with variable elements. In a final step the damping is adjusted via a matched damping oil. A damping ratio of up to 25 % can be reached with a special lamellar contour inside of the casing. The frequency range, at which the vibration absorber can be implemented, is from 6 Hz to 150 Hz.

Resonance induced vibration overshoots of pipelines are typical applications. These often have – because of their symmetry – two adjacent natural frequencies with orthogonally oscillating mode shapes. The two-dimensional mode of action also enables an effective reduction of the vibration level at more dimensional mode shapes.

4 Laboratory investigations

A test structure was set up for a first view on the vibration reduction potential of the vibration absorber Magic Tube. This test structure consisted of a piping section with an additional mass at the top, see figure 4. For the excitation of the structure an unbalanced motor with variable speed was used.

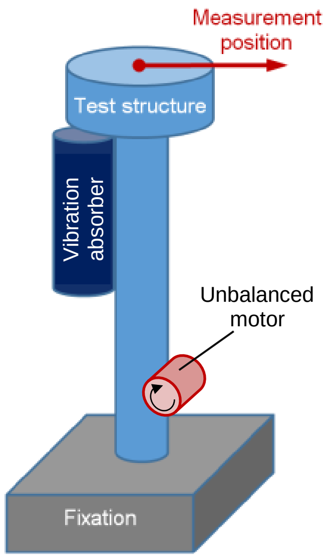

Figure 4: Test rig with experimental setup.

For the laboratory investigations the unbalanced motor was driven with a frequency sweep from 10 Hz up to 40 Hz. The dynamic behaviour of the test structure was analysed without the vibration absorber in a first step. For the original test structure two dominant resonance frequencies occurred at 25 Hz and at 26.5 Hz (see figure 5).

Vibrations of up to 115 mm/s RMS were measured in this initial state. The enormous vibration level resulted from the two original resonances (without absorber). These correspond to the already mentioned orthogonally oscillating mode shapes of the pipeline.

For the assessment of piping vibrations, the standard VDI 3842[3] and ISO 20816-8[4] are proven. In this case the VDI 3842 standard is used. This provides frequency dependent guide values for permissible piping vibrations which are based on measured values and experience in the petrochemical industry over a period of more than 25 years. The comparison with the measured effective vibration velocities shows an unacceptable vibration level, figure 5.

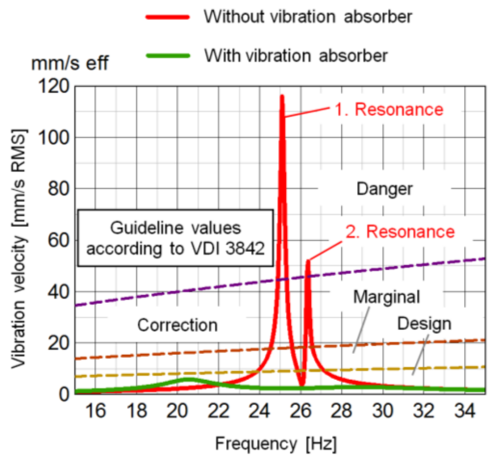

Figure 5: Reached vibration reduction for the experimental setup.

To reduce the increased vibrations, the vibration absorber was installed. The comparison with the original situation shows that the vibration level of the test structure could be reduced with a factor of more than 20. According to the assessment with VDI 3842 the vibration level now is in the design range, at which a flawless “operation” can be ensured from the vibration-technical point of view.

5 Case studies

After the successful laboratory investigations, the vibration absorber was installed on site several times.

5.1 constant speed

In one case, a metrological investigation of three reciprocating compressors for ethylene was carried out. Each of the vertical compressors has got two cylinders and two stages. The power of each unit is 1.5 MW and the fixed running speed is 498 RPM. In dependency from the ethylene consumption of the connected chemical plant many different operating conditions are possible. The suction and discharge pressures are changing continuously. Single and parallel operation of up to three compressors are also possible.

Within the scope of the investigation increased piping vibrations were found in the ethylene compressor station. The maximum piping vibrations of up to 51 mm/s RMS occurred in a section (DN 150, PN 100) located at a height of almost 4 m near the cooler of the second compressor stage.

A comparison to the guideline values according to VDI 3842 showed a clear exceedance of the allowable vibration velocities for the measured vibration frequency of 16.6 Hz, see figure 6.

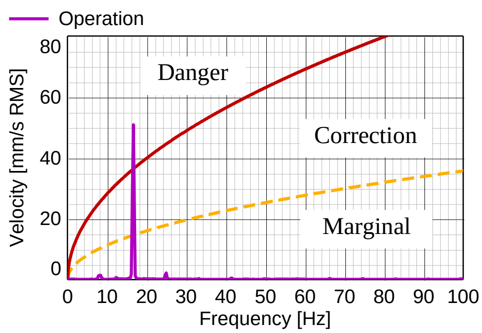

Figure 6: Measured spectrum of the local vibration level at the critical measurement position compared to VDI 3842 guideline values.

To analyse the reason for these increased vibrations, a harmonic response analysis was done during shut down. The mobility of the piping section was determined by bump tests. Figure 7 shows the recorded data of one bump test.

A single beat by a hammer was used to excite the structure, which vibrated with its natural frequency. To get information about the excitation force, a load cell was used. The vibrations were collected with a common vibration transducer. The recorded data could be used for the determination of the structural mobility (special type of transfer function) after some repetitions of the bumps.

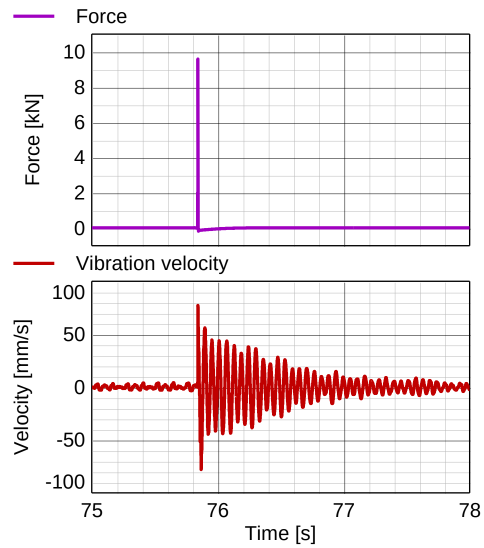

Figure 7: Measured force and vibration during a bump test.

The structural mobility is calculated by a mathematical method using spectral transformations from the time domain into the frequency domain. As a result, the structural dynamic behaviour of the system is described by a complex response with a frequency depending magnitude and phase. Figure 8 shows the spectrum of the magnitude for these bump tests.

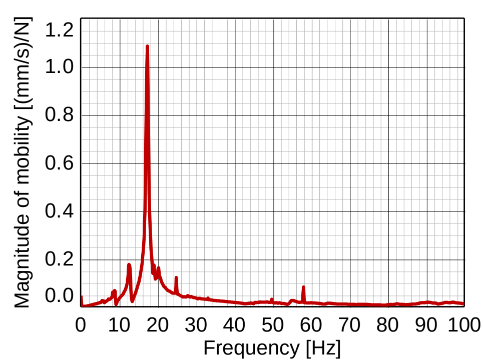

Figure 8: Magnitude of the local mobility at the critical vibration position as a result of the measured bump tests.

The critical piping section had a dominant natural frequency of 17 Hz. This natural frequency is close to the measured vibration frequency of 16.6 Hz during operation, which is the second harmonic of the compressor speed. Thus, the increased vibrations occur due to this structural resonance. A low modal damping ratio D < 2 % could also be detected by analysing the mobility.

To reduce this vibration level, several measures were possible. The challenge was to optimize the mobility at the dominant excitation frequency of 16.6 Hz. One possible solution was the detuning of the system by additional stiffness or additional mass. Additional stiffness was quite difficult in this case because of several other piping nearby. Furthermore, the strut would have to be stiff enough to function in a height of 4 m. Additional mass was another possibility to shift the natural frequency downwards. This was not preferred because in the original condition the natural frequency was slightly above the excitation frequency. The additional mass would have had to be very heavy for the natural frequency to fall far enough below the critical excitation frequency.

Thus, another solution was suggested in this case: The installation of a vibration absorber.

One advantage of the vibration absorber Magic Tube is that it can be applied on any structure and does not need any support to its surrounding. So, it did not matter that the critical vibrations occurred in a height of 4 m where support is difficult. Another advantage corresponds to the realization of a damped vibration absorber: There is no more dominant resonance peak in the frequency spectrum when using the damped vibration absorber to solve vibration issues. This is an additional advantage for sometimes critical start-up conditions when machines must pass through resonance frequencies.

To get a well working design of the absorber, it is necessary to know the modal parameters of the original structure. Therefore, a curve fitting tool is used to calculate the relevant structural parameters: stiffness, damping and mass of the piping section.

For the design of the vibration absorber the modal mass of the original structure and its damping ratio are the most important design values. The damping ratio was already known (D < 2 %) and the calculated modal mass was about 180 kg.

Knowing these parameters, the vibration absorber was configured on site with a mass of 14 kg, a natural frequency of 16.6 Hz and a damping ratio of about 7 %. These values are based on our long-term experience and recommendations from technical literature[5,6]. Subsequently, bump tests were carried out to check the influence on the modified structure.

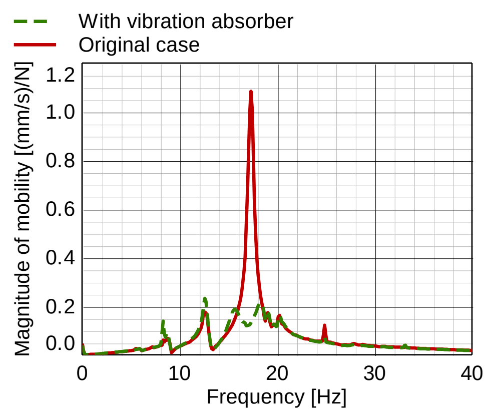

Figure 9: Comparison of the mobility with and without damped vibration absorber.

Figure 9 clearly shows the improvement due to the influence of the vibration absorber. There was no longer a dominant resonance peak. Thus, the modified structure was suitable for continuous operation.

To check the function and the performance of the vibration absorber over a longer period of operation, a long-term measurement was installed for more than one month. Figure 10 shows a timeline of the RMS vibration value for the previously critical measurement position. The improved vibration level was far below the guideline value of the VDI 3842 for permissible vibrations. The different vibration levels during this long-term measurement were a result of the varying operating conditions of the three reciprocating compressors.

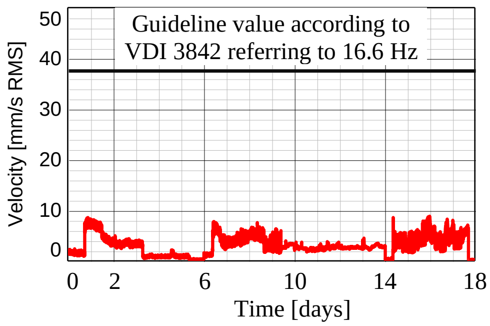

Figure 10: Vibration level during long-term measurement (time excerpt).

The maximum effective vibration level is now about 8 mm/s RMS. Figure 11 shows the installation position and compares the original case with the optimized case to visualize the effect on the vibration level.

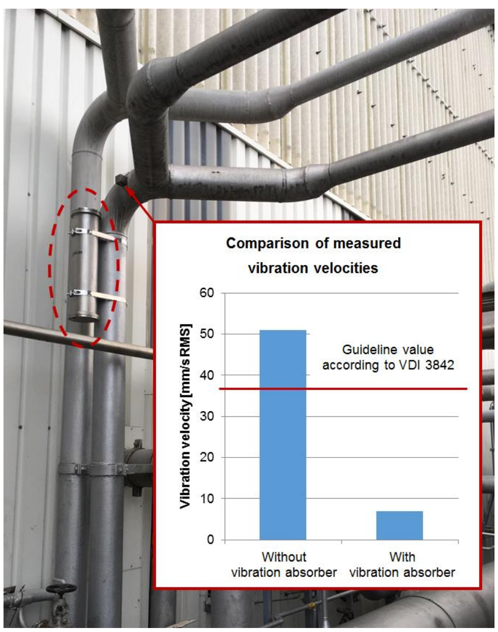

Figure 11: Installation and measurement position of the vibration absorber with comparison of vibration velocities.

5.2 variable speed

In another case, a metrological investigation was carried out at a 4-cylinder reciprocating compressor for a chemical process. The compressor has a power of 1.25 MW and a variable speed range from 155 RPM to 320 RPM.

This plant had increased vibrations at the discharge piping downstream of the first stage pulsation damper. The vibrations were in horizontal direction and additional supports were not an option because there was no solid structure to attach additional supports.

Figure 12 shows the position of the high vibrations and the conditions on site.

Furthermore, no shut down of the plant was possible. The production had to continue and no shut down bump tests could be performed.

First, the effective vibration velocities were measured for different operating conditions with different compressor speeds.

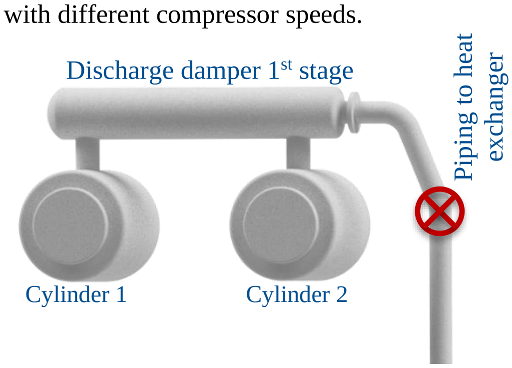

Figure 12: Position of the high vibrations (marked in red) and the conditions on site.

Figure 13 shows the effective vibration velocity as a function of compressor speed. A typical behaviour for resonance problems can be seen here. The periodic working principle of a compressor leads to vibrations with all harmonics of the compressor speed. Therefore, many different harmonics coincide with the natural frequency of the structure during speed variations.

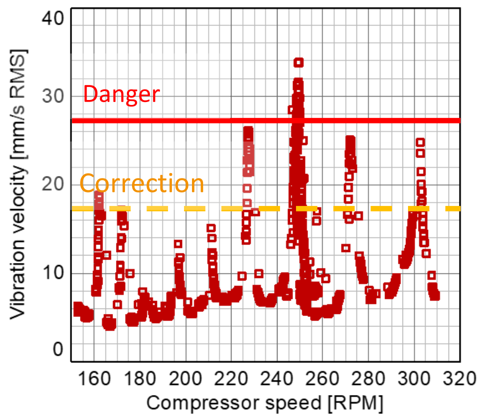

Figure 13: Effective vibration velocity as a function of compressor speed and assessment according to DIN ISO 20816-8.

According to the customer's request, the measured condition was evaluated in compliance with the DIN ISO 20816-8 guideline. The vibration velocity is partly above the limit of the danger zone at the considered position.

Since no shutdown was possible, bump tests were performed during operation. Figure 14 shows the result of the bump tests already analysed in the form of the structural mobility (red curve). The marked harmonics of the compressor were not derived from the force applied by the modal hammer and were ignored for further analysis.

Using a curve fitting algorithm, the structural parameters can again be determined, and an absorber can be configured - in this case for broadband excitation.

Eigenfrequency:	39 Hz
Mass:	12.2 kg
Damping ratio:	20 %

After installation of the vibration absorber, another bump test shows that no dominant resonance occurs anymore with this new structural system (green curve). The Magic Tube works in a wide frequency range of the excitation.

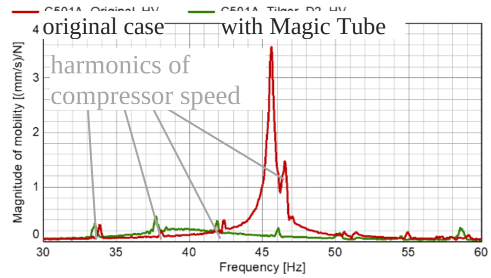

Figure 14: Magnitude of the local mobility at the critical vibration position with and without the Magic Tube.

A verification of the effect of the vibration absorber was also carried out during operation. Figure 15 clearly shows the effect achieved as a function of the compressor speed.

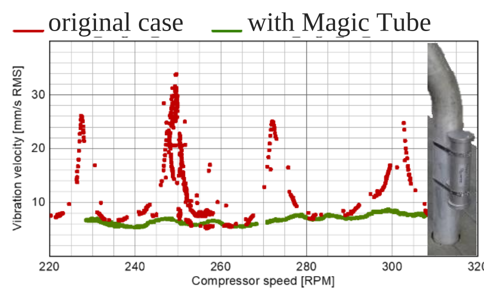

Figure 15: Comparison between the operational effective vibration velocities as functions of the compressor speed with and without Magic Tube.

5.3 other applications

In addition to the typical application of the Magic Tube on piping, it is also successfully used, for example, on vessels, blowouts, or small pipe connections. For this purpose, the product range was extended by a smaller version (XS). A larger series of the Magic Tube (XL) is also being planned.

A variety of application examples of the Magic Tube for different frequency ranges and structural dimensions can be seen in Figure 16.

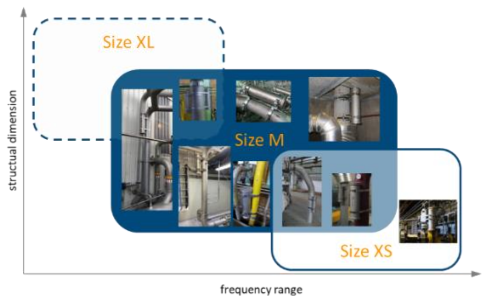

Figure 16: Product range and application examples of the Magic Tube for different frequency ranges and structural dimensions.

6 Conclusion

In this article the vibration absorber Magic Tube has been presented. Its general design was chosen to achieve high flexibility regarding absorbing parameters: natural frequency, absorber mass and absorber damping.

After some laboratory investigations an installation at a piping of a reciprocating compressor with fixed speed has been performed. In this case, the critical vibration level was reduced from about 51 mm/s RMS down to 8 mm/s RMS. The second case study shows that also at systems with variable rotational speeds a significant reduction of the vibration level can be achieved.

Other areas of application for the vibration absorber are all areas of machinery and plants which are presented briefly.

References

[1] Lenz, J.: Three-dimensional vibration absorber to eliminate high vibrations in pipelines. In: Gas Machinery Conference. 6th October 2008, Albuquerque, New Mexico.

[2] KÖTTER Consulting Engineers GmbH & Co. KG: Schwingungstilger oder Schwingungsdämpfer. Inventor: Patrick Tetenborg und Johann Lenz. DE, Patentschrift: DE102014006193B4. 2014.

[3] VDI 3842: Schwingungen in Rohrleitungs-systemen. 2004-06.

[4] ISO 20816-8: Mechanical vibration - Evaluation of machine vibration by measurements on non-rotating parts - Part 8: Reciprocating compressor systems. 2018-12.

[5] Petersen, C.: Dynamik der Baukonstruktionen. Vieweg & Sohn Verlagsgesellschaft. Braunschweig / Wiesbaden. 1996.

[6] Den Hartog, J. P.: Mechanische Schwingungen. 2. Auflage. Springer Verlag. Berlin. 1952.