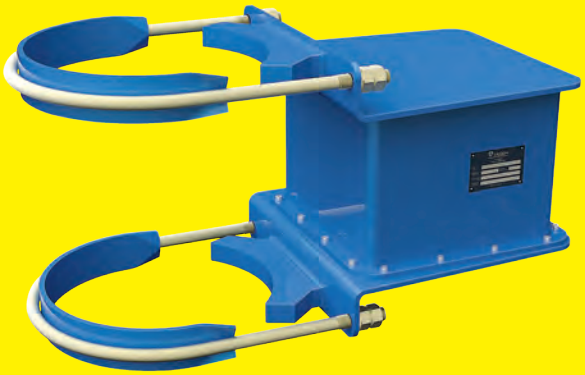
ISOLATING | DAMPING | TUNING



Spring Devices

Viscoelastic Dampers

Elastomeric Devices

Tuned Mass Dampers

Active Damping Devices

VICODA

Vibration Control › Devices › Applications

THE VIBRATION PROBLEM

VICODA has developed a full range of products for vibration control which can be used in different applications, to give for every vibration problem the optimum solution.

Today the reduction of vibrations is gaining importance due to safety, environmental and economic reasons.

During their service life structures and equipment may suffer from problems related to uncontrolled vibrations.

These may be originated by:

- **heavy machines such as engines, turbines and pumps**
- **shock loads, turbulent flows, incident-flows or pressure pulsations within piping systems**
- **external causes like wind, traffic loads or earthquakes**
- **railway lines within urban areas**

Uncontrolled vibrations and shock loads may lead to fatigue and high stress causing severe damage to structures and facilities. This may translate into high costs for:

- **repair of failed components**
- **downtime of machines**
- **maintenance**
- **environmental and safety issues**

Taking into account and solving these problems right from the beginning at the design stage will later avoid expensive and time consuming retrofitting operations.

For each vibration problem the optimum solution!

VICODA is an engineering company that can help you control vibrations in the following fields:

- **plant engineering**
- **buildings**
- **bridges**
- **railways**
- **seismic protection**

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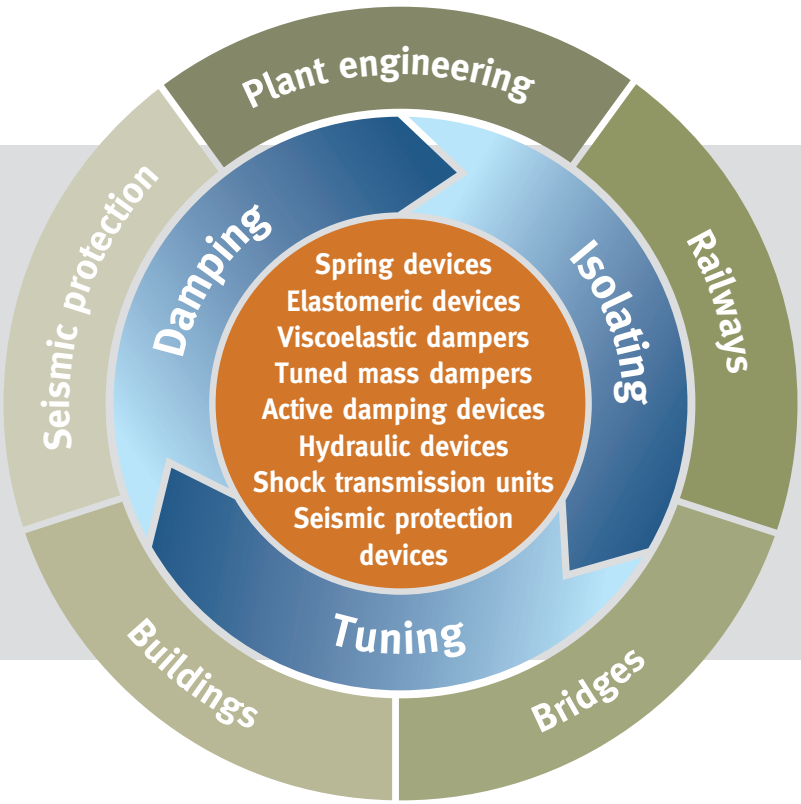


VICODA solutions provide:

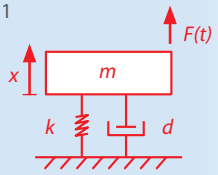
SAFETY
for personnel, facilities and environment

AVAILABILITY
for plants, installations and construction

LONGEVITY
for the amortization of investments

TECHNICAL BACKGROUND

Fig. 1



F_I = Inertia force [N]
F_D = Damping force [N]
F_E = Elastic force [N]
$F(t)$ = excitation force [N]

x = displacement [m]
$\dot{x}$ = velocity [m/s]
$\ddot{x}$ = acceleration [m/s²]
m = mass [kg]
k = stiffness [N/m]
d = damping parameter [N/(m/s)]
t = time [s]

Harmonic excitation force
$F(t) = F_0 \cdot \sin(2\pi f_E t)$
F_0 = force amplitude

Natural angular frequency [1/s]:
$\omega_0 = \sqrt{\frac{k}{m}}$

Natural frequency [Hz]:
$f_0 = \frac{1}{2\pi}\sqrt{\frac{k}{m}} = \frac{1}{2\pi}\omega_0$

Frequency ratio [1]:
$\eta = \frac{\omega_E}{\omega_0} = \frac{f_E}{f_0}$

Modal damping ratio [1]:
$D = \frac{d}{2m\,\omega_0}$

The modal damping ratio D describes the damping in a system. In case of a vibratory system as shown in Fig. 1 without external excitation but an initial displacement and

D = 0 the mass will continue oscillating like in an undamped system,

D ≥ 1 the system will be strongly damped so that the mass will return to its rest position without any overshoot,

0 < D < 1 the mass will oscillate with a decreasing amplitude and finally come to rest.

Basic physical model

Systems with vibration problems often are very complex. However, most of the fundamental effects may be discussed using a simple system with only one degree of freedom. Complex systems can be transferred into a set of these simple systems (modal reduction).

The basic model for an excited and damped oscillator with one degree of freedom is shown in Fig. 1.

An external force ***F(t)*** acts on the mass ***m*** that is connected to a spring with a stiffness ***k***. The resulting motion of the mass is damped due to a damping device with a damping parameter ***d***.

Starting from the equilibrium of the acting forces in the system

$$F_I + F_D + F_E = F(t)$$

the differential equation describing the motion of the mass is given by

$$m\ddot{x} + d\dot{x} + kx = F(t)$$

In case of no external excitation but an initial displacement of the mass leading to an elastic force of the spring, the solution of the equation describes a free, damped oscillation – the mass oscillates with its natural angular frequency ω_0 i.e. its natural frequency f_0.

In case of a sinusoidal (harmonic) excitation force with a frequency f_E the change in maximum displacement of the system (amplitude amplification) is given by

$$V = \frac{1}{\sqrt{\left(1-\left(\frac{f_E}{f_0}\right)^2\right)^2 + \left(2D\frac{f_E}{f_0}\right)^2}} = \frac{1}{\sqrt{(1-\eta^2)^2 + (2D\eta)^2}}$$

For slow excitations ($f_E \to 0$, $\eta \to 0$, $f_E << f_0$) the system response is *quasi-static* i.e. the motion of the mass directly follows the excitation force. There is practically no increase of the amplitude of the oscillation compared to the static displacement (Fig. 2).

For excitations with high frequencies ($f_E \to \infty$, $\eta \to \infty$, $f_E >> f_0$) the system cannot follow the excitation force. Therefore the amplitude of the oscillation is reduced and approaches zero (Fig. 2).

If the frequency of the excitation force f_E is close to the natural frequency of the system ($\eta \approx 1$, $f_E \approx f_0$) the motion of the mass is increased significantly. This effect is called *resonance*. The increase of the amplitude is depending on the damping parameter of the system, creating different amplitude amplification functions for different damping parameters (Fig. 2). In general the amplification function can be divided into two areas. For $\eta > \sqrt{2}$ the amplitude is always reduced even without a damping element – this area often is called *isolation* area.

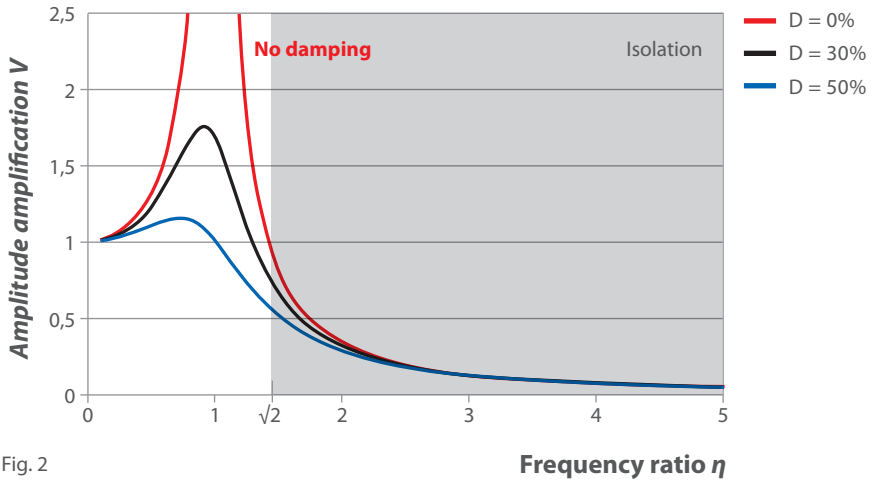


Fig. 2

VIBRATION CONTROL PRINCIPLES

In order to control the effects of unwanted vibrations in plants, machines or structures in general there are the following fundamental principles:

1.) **Isolation**
2.) **Damping – Dissipation**
3.) **De-tuning**
4.) **Energy transfer to a coupled oscillating system – Absorption**
5.) **Active vibration control**

ISOLATION

In case of vibration control by isolation it is avoided that energy from the source of vibration (exciter) is transferred to the vibratory system. Due to the reduced energy transfer the vibratory system will oscillate with a smaller amplitude compared to the non-isolated situation.

A good and simple model to discuss vibration isolation is given in Fig. 3. The model describes a damped oscillator with base excitation i.e. an induced motion of the base that was considered fixed in the model described in Fig. 1. The differential equation describing the motion of the mass is given by

$$m\ddot{x} + d\dot{x} + kx = d\dot{s} + ks = F(s(t))$$

In case of harmonic base excitation s(t) the solution of the equation is very similar to the results discussed for the basic physical model above. The amplitude amplification function is given by

$$V = \frac{\sqrt{1+\left(2D\frac{f_E}{f_0}\right)^2}}{\sqrt{\left(1-\left(\frac{f_E}{f_0}\right)^2\right)^2+2D\left(\frac{f_E}{f_0}\right)^2}}$$

Like in the basic physical model the amplitude is significantly reduced for high frequency ratios η or $f_E >> f_0$. The system can be considered *isolated* from the exciting force. However, in cases where the frequency f_E of the base motion is close to the natural frequency f_0 of the vibratory system there will be resonance.

In order to achieve isolation in this case a soft coupling has to be introduced between the system and the base as shown in Fig. 4.
In fact the vibration problem is now a problem with two degrees of freedom as a second system has been introduced. However, if the stiffness k' is much smaller than the stiffness of the original system k i.e. $k' << k$, the original system can be considered much more rigid than the vibratory system defined by the soft coupling. Therefore the model can be simplified to the model shown in Fig. 5. The natural frequency f_0' of the simplified system is defined by the total mass $m' = m^* + m$ and k'.
For a good isolation the coupling has to be defined in a way that the resulting natural frequency f_0' is much lower than the excitation frequency f_E of the base motion. The amplitude amplification function of the simplified system has its maximum at the new natural frequency f_0' (Fig. 6). Compared to the original system the amplitude amplification (Fig. 6) can be significantly reduced.

Isolation is a very effective way to avoid unwanted vibrations, as energy transfer from the exciter to the vibratory system is systematically prevented. However, in order to apply isolation as an approach to control vibrations it is often a prerequisite to consider it during the design of the system, e.g. during the design of a machine or a building and its foundation. A later implementation of an isolation solution to reduce unwanted vibrations is often costly.

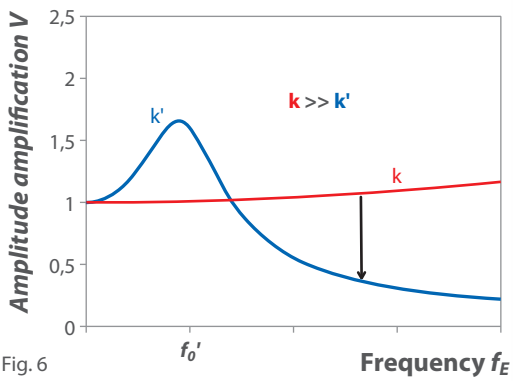
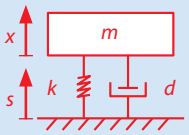


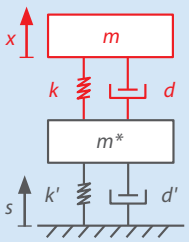
Fig. 6

Fig. 3



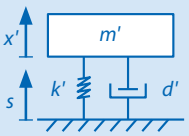
s = Displacement of base [m]
$\dot{s}$ = Velocity of base [m/s]

Fig. 4



Parameters of coupling system:
m^* = mass [kg]
k' = stiffness [N/m]
d' = damping parameter [N/(m/s)]

Fig. 5



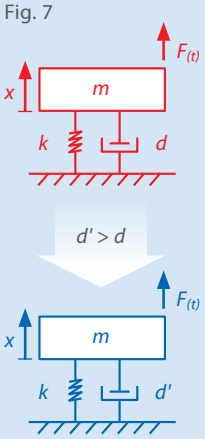
$m' = m + m^*$

$f_0' = \frac{1}{2\pi}\sqrt{\frac{k'}{m'}}$

$\eta' = \frac{f_0'}{f_E}$

DAMPING – DISSIPATION

Fig. 7



In case of vibration control by damping energy from an oscillating system is dissipated by a damping element (Fig. 7). In contrast to isolation here the system is excited first i.e. there is an energy transfer from the exciter to the vibratory system. Within the damping element a portion of kinetic energy of the vibration is converted into another form of energy like heat, thus reducing the amplitude of the oscillation.

Damping is only an effective method to reduce vibrations in the range of the system's natural frequency f_0 i.e. around resonance (Fig. 2).

By increasing the damping parameter the amplitude of the system in that area can be significantly reduced. The related change of the amplitude amplification function is shown in Fig. 8. Damping elements have to be designed in a way that they efficiently dissipate energy in that frequency range.

Other than in the case of isolation damping can often be applied as a retrofit measure to control vibrations. Viscoelastic dampers are a typical example for a damping device that can easily be added to e.g. an installed piping system.

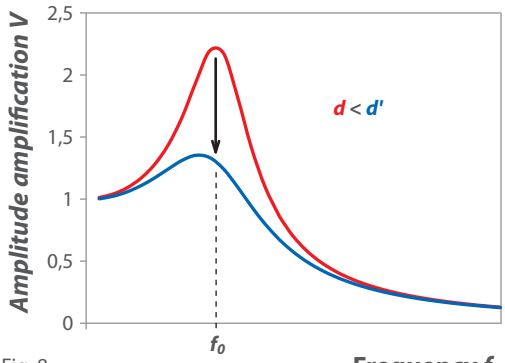
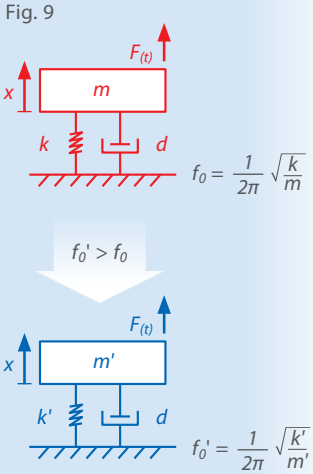


Fig. 8

DE-TUNING

Fig. 9



The basic characteristics of a vibratory system are defined by the oscillating mass, the damping as well as the elastic properties of the system. In case of an excitation around the natural frequency of the system, i.e. around resonance, the resulting amplitude of the oscillation is specifically high (Fig. 2). When changing the size of the oscillating mass or the elastic properties of the system the natural frequency of the system will change as well (Fig. 9).

By that change it can be achieved that the natural frequency of the system and the excitation frequency are no longer in the same range i.e. the system is no longer in resonance for the given excitation frequency and the amplitude of the oscillation is reduced. In fact the amplitude amplification function will shift depending on the natural frequency (Fig 10).

De-tuning can be done by changing the mass or – the typical approach – by stiffening the system i.e. increasing the stiffness k. Obviously with the new natural frequency resonance will occur at another excitation frequency. Therefore it has to be made sure that the new natural frequency is not matching any other possible excitation frequency. Like damping de-tuning can be applied as a retrofit measure. However, significant de-tuning is often only possible in a narrow frequency range as

$$f_0 \sim \sqrt{k}$$

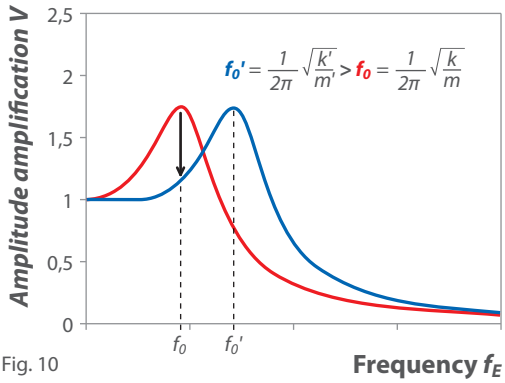


Fig. 10

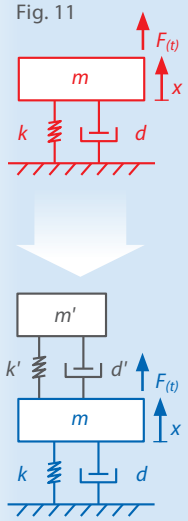
ENERGY TRANSFER TO A COUPLED OSCILLATING SYSTEM – ABSORPTION

A special way of reducing the amplitude of an oscillating system is to couple it to another vibratory system, the **t**uned **m**ass **d**amper (TMD), and tune the characteristics of this coupled system i.e. its mass, damping properties and stiffness so that it efficiently gets energy transferred from the original system (Fig. 11).

As a result the TMD will oscillate while the vibration of the original system is significantly reduced (Fig. 12). Vibration energy of the original system is *absorbed* by the tuned mass damper.

Passive TMDs are to be designed in a way that they efficiently absorb energy from the original system when it is oscillating at its natural frequency. The level of absorption depends on the initial damping of the original system. The damping element of the TMD, e.g. a viscoelastic or eddy current damper, allows the use of the TMD in a wider frequency range. TMDs often are a good retrofit measure, as they can easily be mounted on vibrating structures without any abutment.

Fig. 11



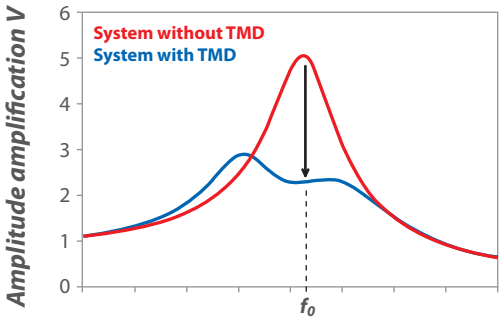


Fig. 12

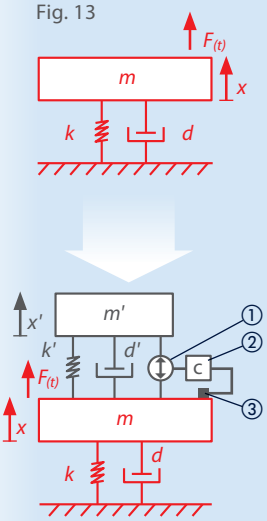
ACTIVE VIBRATION CONTROL

A very special and unconventional approach to control vibrations is to couple an *active* system to the original system (Fig. 13). These active systems are equipped with sensors and actuators and can counteract the oscillation of the original system.

By superposition of the vibrations of both systems the initial vibration of the original system can be significantly reduced.

Different from TMDs active devices can be adjusted to changing excitation frequencies. They are also a good retrofit measure e.g. for vibration problems in piping systems, as they can be easily mounted onto the piping system and significant vibration reduction can be achieved with only little mass *m'* of the active system.

Fig. 13



① Actuator
② Controller
③ Sensor

With all the above methods unwanted vibrations can be controlled and significantly reduced. The design of the best solution is an optimizing task for the VICODA engineers. Due to the comprehensive portfolio of vibration control products like spring elements or elastomeric bearings for isolation, viscoelastic dampers for damping, spring elements or rigid steel struts for de-tuning, TMDs or active devices, they are not limited to a certain method to solve a vibration problem but can choose the optimum solution and support it with the optimum products.

SPRING DEVICES

ISOLATION

Spring devices are the prototype product to isolate a vibratory system from a source of vibration. They are to be positioned between the system and the source in order to minimize the energy transfer from the source to the system. The natural frequency of the vibratory system f_0, defined by the spring devices and the load they carry, has to be designed in a way that it is significantly lower than the main natural frequency f_E of the vibration source.

FIELD OF APPLICATION

Spring devices can be used for solving a wide range of vibration isolation problems. A typical industrial application is the base isolation of heavy duty machinery like presses or forge hammers. In power plants spring devices are installed to isolate coal mills, gas- or diesel engines, turbines, pumps or condensers. Compressors or big fans are other examples where the isolation with spring devices avoids that the vibration induced by this equipment – the source of the vibration – is transmitted to the environment (isolation at source).

Spring devices can also be used to isolate a vibratory system against vibrations induced by the environment (isolation of recipient). Examples for this are the base isolation of precision equipment or structures like buildings against vibrations induced e.g. by traffic.

In any case base isolation with spring devices is a state of the art approach to reduce the effects of unwanted vibrations thus allowing:

- protection against shocks and continuous excitation for man, machines, structures and building
- control of operational vibrations within the admissible limits
- reduction of wear and tear of machines, equipment and structures.

As a consequence, with a good base isolation concept for e.g. a big hydraulic press, the foundation of the machine may be designed in a significantly lighter way (Fig. 14). This concept allows the installation of vibrating equipment in almost every location and it allows to considerably reduce the cost of the foundation itself.

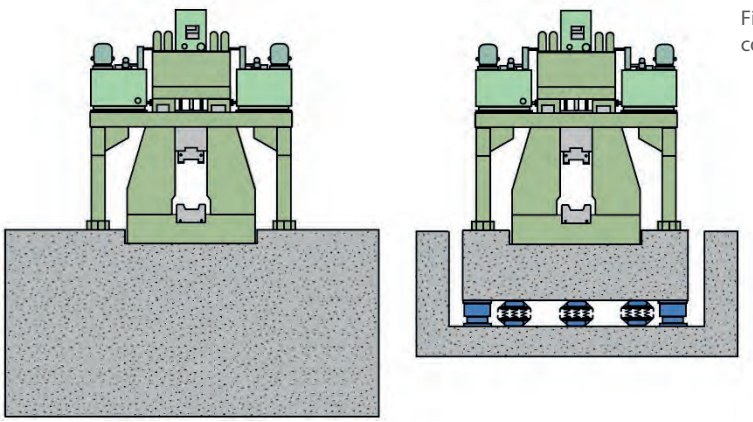


Fig. 14 - Different base isolation concepts for a hydraulic press

The above example shows that the range of applications of spring devices is big and so is the related product portfolio.

DESIGN

The basic design of a spring device consists of a set of helical coil steel compression springs connected to a top and bottom plate (Fig. 15). These plates are fixed to the vibratory system and the vibration source through different anchor systems or by screwless fastening. For screwless fastening the spring devices may be fitted using adhesive pads that in addition allow the decoupling of the system from structure-borne noise.

Springs

The springs are the core elements of a VICODA spring device. Being an affiliate of LISEGA, the world market leader for pipe supports, VICODA benefits from LISEGA's more than 50 years of experience in the design and manufacturing of spring based constant and spring hangers. The full portfolio of standardized, high quality LISEGA springs is available for VICODA's design engineers. However, to comply with all the typical requirements of base isolation additional standardized springs were added.

Over time – especially under higher loads and at higher temperatures – the stiffness of conventional helical coil springs may decrease (spring relaxation). In order to avoid any negative effects due to that effect VICODA uses pre-relaxed springs guaranteeing the constant functionality of springs over their lifetime.

In order to ensure high quality corrosion protection VICODA springs are given a special surface treatment based on a cathodic immersion process (CIP) that was adopted from automotive industry: first the peeled surface of springs is steel ball blasted and then zinc-phosphated, subsequently a 2-component epoxy resin coating is applied by electro-immersion and burnt in at ~200°C.

In order to cover the wide range of applications VICODA has developed a modular approach for the design of spring devices using a limited number of standardized springs.

As discussed in the technical background the natural frequency f_0 of a vibratory system is given by

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

Applying this to the system defined by the spring device and the mass it carries, it can be concluded that the natural frequency of the system is directly depending on the spring compression (travel) due to the weight of the mass (dead load):

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{k}{m}} = \frac{1}{2\pi}\sqrt{\frac{m \cdot \frac{g}{x}}{m}} \approx \frac{1}{2}\frac{1}{\sqrt{x}}$$

Therefore, only by checking the travel of a spring device at dead load the natural frequency of the system can be determined (Fig. 16). For good base isolation this natural frequency has to be significantly lower than the main frequencies of the vibration source.

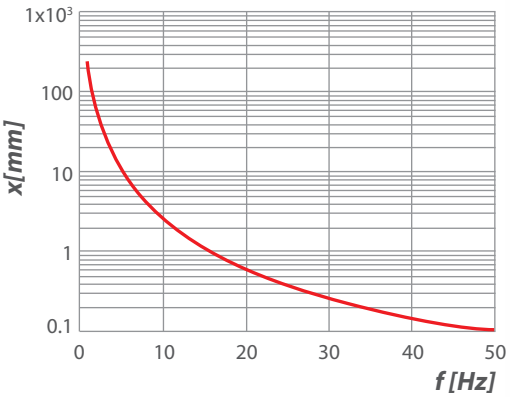


Fig. 16 - Travel of a spring device and the resulting natural frequency

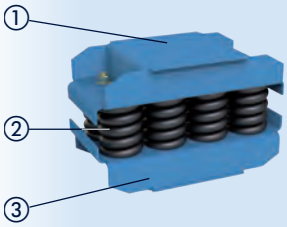
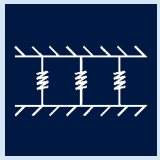


Fig. 15

① Top plate

② Helical coil

③ Bottom plate



Damping Elements

In general spring devices only have very low damping properties due to natural material damping of steel components. Whenever higher damping characteristics are needed, viscoelastic damping elements can be integrated into the spring devices.

Typical examples for the need of higher damping characteristics are impulsive excitations as induced e.g. by forge hammers. In this case a damping element is needed to quickly convert the significant kinetic energy of the hammer into heat thus rapidly stopping the motion of the hammer.

Another example is the existence of different operational conditions of the vibration source leading to changing frequencies f_E, e.g. during ramp-up of a turbine. Starting from a rotation frequency of 0Hz the turbine will accelerate up to its continuous operational frequency. As a base isolation spring device is designed in a way that the vibratory system has a natural frequency that is lower than the continuous operational frequency the system will be exposed to frequencies close to its natural frequency f_0 during the ramp-up phase. In order to avoid significant resonance effects during that phase a safety damping element has to be added. In this case the energy to be converted into heat is significantly smaller than in the case of the forge hammer.

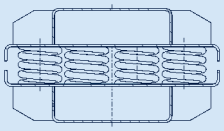
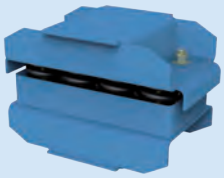


Fig. 17 - Spring device type SL



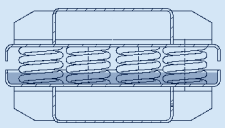
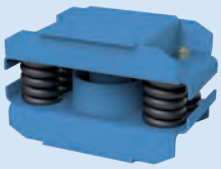


Fig. 18 - Spring device type SM



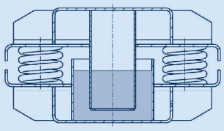


Fig. 19 - Spring device type SH

Preloading and locking of spring devices

An important feature of spring devices is the function of locking them at any spring compression (Preloading). Preloaded spring devices are installed in a locked position by that acting as solid supports. At the end of the installation – which may be the installation of heavy machinery or the erection of a building – the spring devices are *activated* i.e. they are unlocked. From that moment on the devices act as vibration control elements.

This feature of spring devices allows for an easy adjustment or maintenance, as it is not necessary to lift the supported machine, structure or building.

For any alignment, maintenance or replacement of spring devices without that feature it is indispensable to first lift the supported machines or structure leading to significantly higher related cost.

Basic types of spring devices

VICODA spring devices are designed based on a modular approach allowing for cost-effective manufacturing and quick delivery times. The fundamental types of VICODA spring devices are:

- **Spring devices without damping elements – type SL**
  The design of these spring devices consists of a set of springs connected to a top and bottom plate (Fig. 17). By combining VICODA standard springs these devices allow an optimum adjustment of the natural frequency of the vibratory system and by that effective base isolation for a given excitation frequency of the vibration source.

- **Spring devices with safety damping elements – type SM**
  Different from the spring devices of Type SL, the springs of the devices of type SM are embedded in a highly viscous fluid (Fig. 18). This adds a safety damping characteristic to these devices.

- **Spring devices with viscoelastic damping elements – type SH**
  These spring devices provide a viscoelastic damping element (Fig. 19) comparable to the standard VICODA viscoelastic dampers. These damping elements allow a significantly higher damping than the safety damping elements.

VICODA spring devices are available with or without the feature of preloading.

All spring devices are delivered with high quality corrosion protection in line with international standards. For extreme operational conditions VICODA can provide adequate special surface protection using its surface treatment center.

CONFIGURATION OF SPRING DEVICES

The basic parameters needed for the definition of a base isolation solution with spring devices are:

- the load [kN], given by the weight of the supported machine, structure or building
- the main excitation frequencies [Hz] of the vibration source
- operating temperatures [°C]– in case of damping elements
- installation dimensions [mm] and geometrical limitations.

Even though the definition of the general solution is rather simple the many combinations of different numbers of standardized springs and damping elements make it hard for a non-expert to define the optimum solution for his specific problem.

That is why VICODA engineers are available to help you configure the optimum solution using the VICODA modular system.

If and to what extend unwanted vibrations may affect a system can be determined at a rather early stage in new built projects. As good base isolation can reduce the stress on a structure significantly, the structure itself might be designed in a much lighter way. Therefore an early discussion of your problem with VICODA engineers and their early involvement in the process can help you to identify cost-effective overall solutions.

TESTING, INSTALLATION AND MAINTENANCE

All VICODA spring devices are subject to a final test before shipment. LISEGA, VICODA's parent company an experienced designer and manufacturer of dynamic and static test stands, has provided all the know-how for the related test equipment and processes.

Installation of spring devices can be supported and supervised by experienced VICODA service personnel.

Due to the high quality of the spring devices they can be considered maintenance-free in case of normal operating conditions and the typical lifetimes of the supported structures. However, when equipped with damping elements a visual inspection every 10 years is recommended.

For these maintenance inspections, for replacements in general and for any kind of base isolation related trouble shooting the VICODA experts are available.

SPRING DEVICES – TYPES SL, SM, SH

STANDARD SPRINGS

The standardized VICODA spring portfolio covers most of the typical applications of base-isolating spring devices. Typical natural frequencies of the isolated system consisting of the spring device and the supported machine, structure or building are in the range of up to 8Hz. By combining a set of standardized VICODA springs in a device an optimum device for a vibration isolation task may be configured.

Starting from the given dead load of the system the set of standard springs has to be chosen in a way that the total stiffness of the device will create the intended natural frequency. As discussed the total stiffness of the spring device, its travel under dead load and natural frequency are linked. The standardized VICODA spring devices cover all the typical applications as shown in Fig. 20. As an example when using a spring device of type SL 102 121-2, consisting of 12 standard springs with a total stiffness of 47.76kN/mm to support a system with a dead load of 955kN, the system will have a natural frequency of 3.5Hz and the travel under dead load will be 20mm.

Another option to carry the same load can be the use of spring type SL 107 161-12 which will provide a natural frequency of 2.8Hz. For even higher isolation, spring device SL 108 101-4 can be used, ensuring a natural frequency of 2.46Hz.

In most of the cases there will not be one single device to support and isolate a system but a set of spring devices. In this case of course the natural frequency of the system will be defined by the totality of spring devices used. In order to find the best solution to a given vibration isolation problem an optimization has to be done with regard to the number of standard spring devices and the type of standard spring devices i.e. the number of springs used per device.

As the number of possible combinations of standard springs and standard spring devices is rather high VICODA engineers are ready to help you with the identification of the optimum solution. Based on the general configuration parameters defined on page 11, they will configure a solution using the modular VICODA spring device portfolio based on standardized springs.

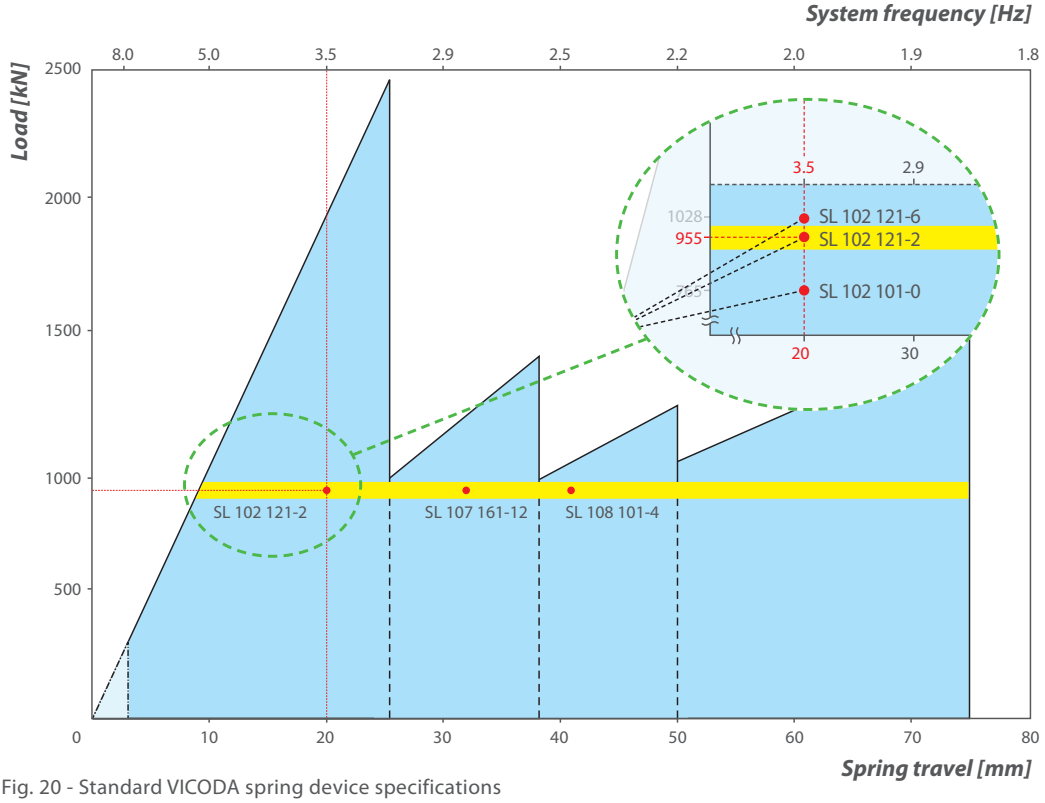


Fig. 20 - Standard VICODA spring device specifications and selection of device SL 101 121-2

The modular VICODA spring device portfolio is optimized with regard to manufacturing thus allowing quick delivery times and cost effective production. As a prerequisite for that the complete technical documentation including design and production drawings, bill of materials, special certifications as well as customer documentation is made available in the VICODA IT systems.

As an example the typical technical data-sheets for spring devices of type SL 102, SL 107 and SL 108 are shown in Fig. 21.

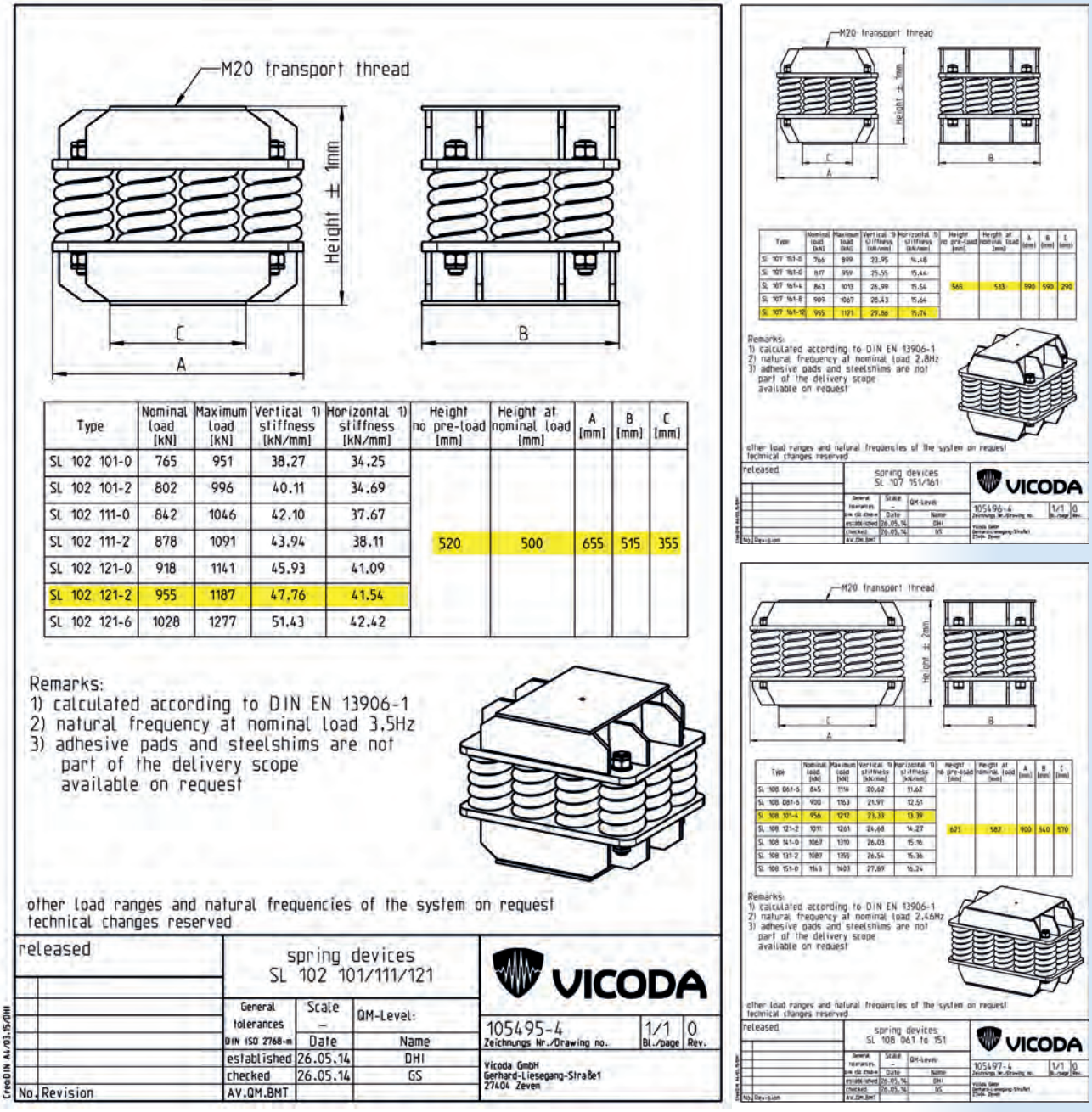


Fig. 21

VISCOELASTIC DAMPERS

DAMPING – DISSIPATION

Viscoelastic dampers reduce vibrations by converting kinetic energy into heat thus damping the motion of the system. Damping, as a method to reduce unwanted vibrations, is most effective in cases where the vibratory system is excited with a frequency close to its natural frequency. VICODA's viscoelastic dampers are designed in a way that they can be used in a wide frequency range.

FIELD OF APPLICATION

Viscoelastic dampers may be used to solve very different vibration problems. A typical example is the damping of operational vibrations in an industrial piping system e.g. in a power plant. Whenever the piping systems cannot be isolated from the source of vibration, damping might be an adequate measure to reduce the motion of the piping to an acceptable level.

Another example is the damping of shock-like forces induced e.g. by a large forge hammer. In this case the damping device needs to absorb a significant amount of energy in a very short time, so that the system comes to rest as quickly as possible allowing the start of the next operation of the hammer with minimum recovery time.

Viscoelastic dampers may also be used together with spring devices e.g. for the base isolation of a smaller diesel engine foundation in a power plant (Fig. 22).

During start-up, the engine will continuously increase its speed up to the operational frequency. Spring devices are designed in a way that at operational frequency the turbine foundation is well isolated from its surroundings. In fact the natural frequency of the system consisting of turbine, foundation and spring devices has to be much lower than the operational frequency. Therefore when starting up the turbine it will increase its frequency from zero to the operational frequency thus passing the resonance range of the system around its natural frequency – that is where it needs a viscoelastic damping element to damp the motion of the system.

Unfortunately sometimes dampers are proposed as an alternative to shock arresters (snubbers). The typical function of shock arresters is to make sure that mechanical connections are protected in case of shock-like forces acting on these connections. Therefore these devices have to provide a rigid link parallel to the connection to be protected that can transmit these forces instead of the connection. An example for this is e.g. the connection of a steam generator in a nuclear plant to the piping that must be protected in case of a seismic event. Here viscoelastic dampers are not the right choice to protect the connection as they do not provide the necessary rigidity with the necessary reaction time.

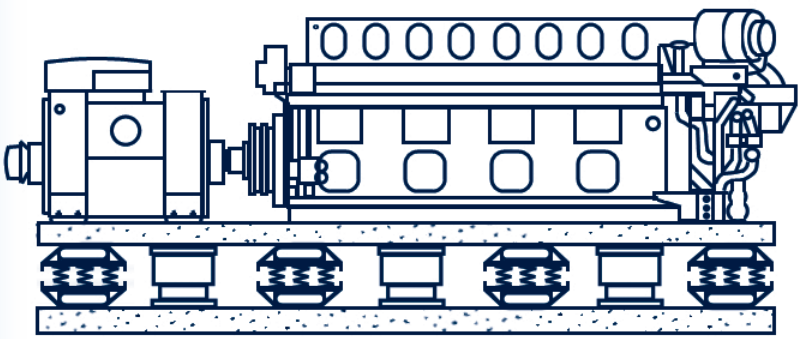


Fig. 22 - Diesel engine supported by spring devices and viscoelastic dampers

DESIGN

VICODA viscoelastic dampers consist of a metal housing that is filled with a viscous fluid (Fig. 23). The piston which is connected to the top plate can move in all directions within the housing (Fig. 24).

Either top or bottom plate of a VICODA viscoelastic damper can be connected to the vibrating system. The respective other plate has to be mounted on a rigid abutment. Because of this viscoelastic dampers do not statically support the system that they are connected to, but they allow that by movement of the piston within the viscous fluid kinetic energy is dissipated i.e. transferred into heat. This free movement of piston / top connection plate against housing / bottom connection plate makes it possible that viscoelastic dampers can follow thermal displacements (travel) of the system they are connected to. In addition, they allow damping of vibrations in all six degrees of freedom i.e. of all linear and torsional vibrations.

The model of *the ideal damper* is a simple but in most cases an adequate way to describe viscoelastic dampers. In this model the damping force F_D, that the viscoelastic damper develops to counteract the movement of the piston, is directly linked to the velocity of the piston

$$F_D = d\dot{x}$$

Typically the damping parameter d is depending on the frequency the system is vibrating with i.e. $d = d(f)$. The frequency dependence of the damping parameter of a damper is determined by tests (Fig. 25). Due to the design of VICODA viscoelastic dampers vertical and horizontal damping parameters are different.

The model implies that slow movements of the piston only lead to small damping forces. Therefore slow displacements of the piston due to e.g. thermal extensions lead to negligible damping forces. However, continuous operational vibrations leading to a displacement Δx of the piston lead to continuous energy dissipation during every cycle of an oscillation (Fig. 26).

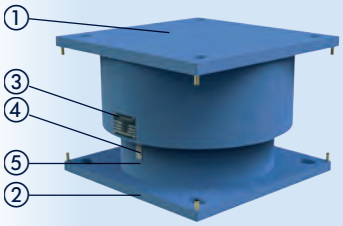


Fig. 23

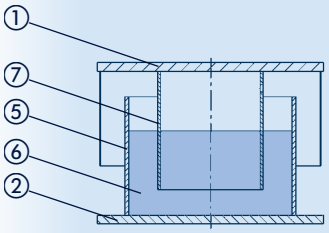


Fig. 24

① Top connection plate
② Bottom connection plate
③ Serial number plate
④ Position indicator
⑤ Housing
⑥ Fluid
⑦ Piston

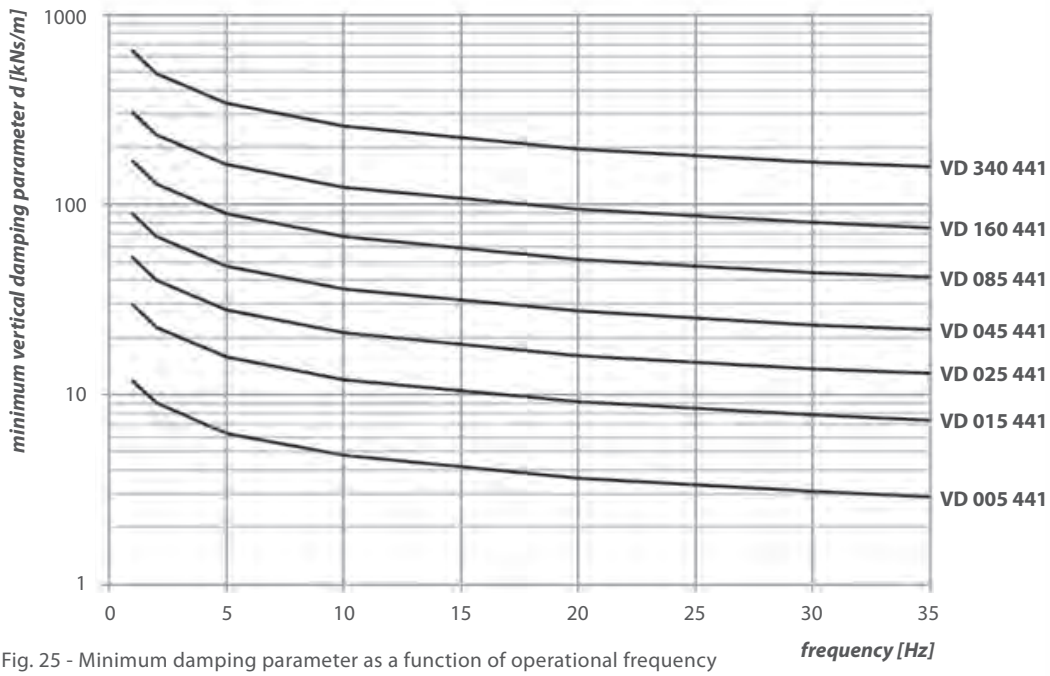


Fig. 25 - Minimum damping parameter as a function of operational frequency for different VICODA viscoelastic dampers

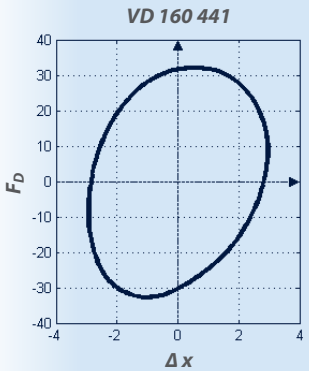


Fig. 26 - Hysteresis of a viscoelastic damper

Fig. 27 - Viscoelastic damper with locking device (red) in neutral (above) and in offset position (below)

Viscous fluids and basic types

Apart from the piston design the most important element to define the damping characteristics of a viscoelastic damper is the selection of the fluid. In general the viscosity of fluids is temperature dependent. Hence for an optimum design of a viscoelastic damper the operational temperatures of the damper have to be considered. The operational temperature will always be a result of the expected ambient temperatures during operation and the heat that will be created within the damper due to its operation.

VICODA is providing three types of fluids with different characteristics depending on temperature and frequency:

- **Viscoelastic dampers type VD** with bitumen based fluids allowing high damping parameters (up to d = 554kNs/m at 5Hz) for operational temperatures from +20°C to 80°C in a limited temperature range of ΔT = 10°C
- **Viscoelastic dampers type VM** with polybuten based fluids allowing medium damping parameters (up to d = 438kNs/m at 5Hz) for operational temperatures from –10°C to +40°C
- **Viscoelastic dampers type VL** with silicone oil based fluids allowing lower damping parameters for operational temperatures of –30°C to +110°C

All VICODA viscoelastic dampers are delivered with high quality corrosion protection in line with international standards. For extreme operational conditions VICODA can provide adequate special surface protection using its surface treatment center.

CONFIGURATION OF VISCOELASTIC DAMPERS

The basic parameters needed for the definition of an adequate damping solution with VICODA's viscoelastic dampers are:

- operating temperatures [°C] i.e. the highest and lowest expected operational temperature that the viscoelastic damper will be exposed to during continuous operation
- the dominant resonance frequencies [Hz] of the vibrating system
- the damping parameters [kNs/m] required
- travel [mm] between installation and operating condition
- installation dimensions [mm] and geometrical limitations.

With these parameters a viscoelastic damper from VICODA's standard portfolio may be identified using the tables in the following section. In order to choose the optimum solution for each problem and to identify the most suitable damping device VICODA engineers are available to help you.

TESTING, INSTALLATION AND MAINTENANCE

All VICODA viscoelastic dampers have been thoroughly tested during their design phase. On request final testing before shipment is possible in the VICODA test labs.

All VICODA viscoelastic dampers are delivered with a transport locking device that has to be removed during its installation and before putting it into operation (Fig. 27).

In cases where a significant travel between installation and operational condition is to be expected the damper can be delivered in an offset position. By this it can be achieved that the damper will operate in a centric position (Fig. 28). It has to be made sure that the dampers are always handled in an upright position to avoid leakage of the fluid. Installation of the dampers on site can be supported and supervised by experienced VICODA service personnel.

Especially dampers using bitumen as viscous medium may have a significantly higher viscosity at installation temperature than at operating temperature. In order to allow smooth travel from installation to operational position of the damper heating elements may be added to heat up the fluid during installation and by that reduce its viscosity.

Although they do not statically support the vibratory system, viscoelastic dampers need a rigid abutment. There are many ways of installing them, connecting either top or bottom plate of the damper to the vibratory system (Fig. 29).

Special components to connect the viscoelastic damper, e.g. to piping, may be designed and delivered as required (Fig. 30).

Due to the high quality of VICODA's viscoelastic dampers they can be considered maintenance-free for 40 years. During this period of time the fluids can be considered as non-aging in case of normal operating conditions. However a visual inspection every year and after exceptional operating conditions – e.g. higher operating temperatures over a longer period of time – is recommended.

For these maintenance inspections, for replacements in general or for any kind of related troubleshooting VICODA experts are available.

Fig. 28 - Viscoelastic damper in neutral and offset position

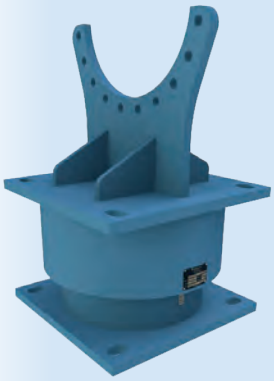


Fig. 30 - Special connection elements

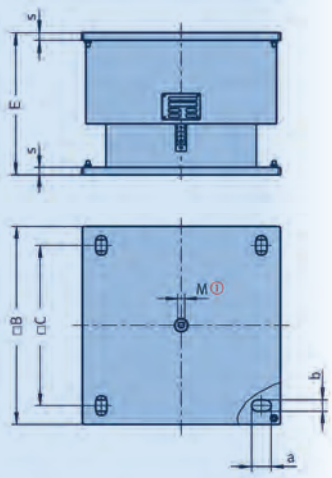
Fig. 29 - Installation examples

Damping of horizontal piping

Damping of horizontal piping

Damping of vertical piping

VISCOELASTIC DAMPERS – TYPE VD



① M = transport thread

Nominal load = maximum force that the damper can transfer at operational temperature

Maximum offset to neutral position is 40mm for all viscoelastic dampers of type VD shown here.

Specifications for dampers with larger offsets are available upon request.

Nominal load, dimensions, weight

Type	Nominal load [kN]	E [mm]	B [mm]	C [mm]	s [mm]	a [mm]	b [mm]	① M	Weight [kg]
VD 005 441	0.3	240	225	200	8	34	14	-	12
VD 015 441	2.5	240	270	215	8	34	14	-	16
VD 025 441	5	240	290	230	8	34	14	-	19
VD 045 441	10	240	340	270	10	38	18	M16	31
VD 085 441	20	280	390	320	12	42	22	M16	51
VD 140 441	30	320	440	350	15	46	26	M16	84
VD 160 441	40	335	470	380	18	46	26	M16	109
VD 185 441	60	350	510	410	20	53	33	M16	149
VD 225 441	80	390	535	430	25	59	39	M16	191
VD 340 441	100	405	580	460	30	59	39	M16	246

Vertical damping parameter [kNs/m]

Type	Nominal load [kN]	5 [Hz]	10 [Hz]	15 [Hz]	20 [Hz]	25 [Hz]	30 [Hz]	35 [Hz]
VD 005 441	0.3	6.2	4.7	4.0	3.6	3.3	3.0	2.9
VD 015 441	2.5	15.7	12.0	10.2	9.1	8.3	7.7	7.3
VD 025 441	5	27.8	21.1	18.0	16.0	14.7	13.7	12.9
VD 045 441	10	47.3	36.0	30.7	27.4	25.0	23.3	21.9
VD 085 441	20	89.3	67.9	57.9	51.6	47.3	44.0	41.4
VD 140 441	30	143.9	109.4	93.2	83.2	76.2	70.9	66.7
VD 160 441	40	162.7	123.7	105.4	94.1	86.1	80.1	75.4
VD 185 441	60	189.4	144.0	122.7	109.5	100.3	93.3	87.8
VD 225 441	80	229.9	174.8	148.9	132.9	121.7	113.2	106.5
VD 340 441	100	340.2	258.7	220.4	196.7	180.1	167.6	157.7

Horizontal damping parameter [kNs/m]

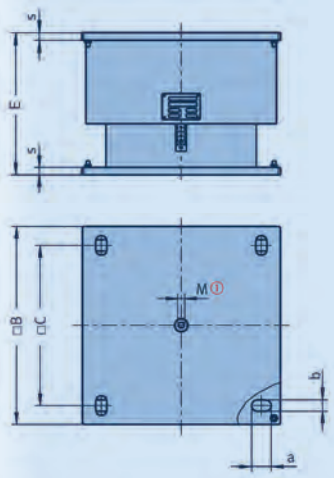
Type	Nominal load [kN]	5 [Hz]	10 [Hz]	15 [Hz]	20 [Hz]	25 [Hz]	30 [Hz]	35 [Hz]
VD 005 441	0.3	5.9	4.5	3.6	3.0	2.5	2.2	2.1
VD 015 441	2.5	14.0	10.8	8.6	7.0	6.0	5.4	5.1
VD 025 441	5	23.7	18.3	14.6	12.0	10.2	9.1	8.6
VD 045 441	10	37.4	28.8	22.9	18.9	16.1	14.4	13.6
VD 085 441	20	94.0	72.5	57.8	47.5	40.6	36.3	34.2
VD 140 441	30	148.7	114.5	91.3	75.1	64.2	57.4	54.1
VD 160 441	40	229.9	177.1	141.2	116.2	99.2	88.7	83.6
VD 185 441	60	293.0	225.8	180.0	148.1	126.5	113.1	106.6
VD 225 441	80	367.3	283.0	225.6	185.6	158.5	141.8	133.7
VD 340 441	100	554.1	427.0	340.4	280.0	239.2	214.0	201.7

VISCOELASTIC DAMPERS – TYPE VM

Nominal load, dimensions, weight

Type	Nominal load [kN]	E [mm]	B [mm]	C [mm]	s [mm]	a [mm]	b [mm]	① M	Weight [kg]
VM 010 331	5	270	260	195	10	38	18	M16	21
VM 020 331	10	270	295	230	10	38	18	M16	30
VM 030 331	15	280	335	265	15	42	22	M16	48
VM 055 331	25	290	425	340	20	46	26	M16	106
VM 100 331	40	300	540	440	25	53	33	M16	193
VM 175 331	50	380	590	480	30	59	39	M16	288

Maximum offset to neutral position is 30mm for all viscoelastic dampers of type VM . . . 331 shown above.



Nominal load, dimensions, weight

Type	Nominal load [kN]	E [mm]	B [mm]	C [mm]	s [mm]	a [mm]	b [mm]	① M	Weight [kg]
VM 010 551	5	390	325	260	10	38	18	M16	39
VM 020 551	10	390	360	290	10	42	22	M16	49
VM 030 551	15	410	420	345	20	46	26	M16	107
VM 055 551	25	410	525	420	20	46	26	M16	158
VM 100 551	40	490	590	470	25	53	33	M16	282
VM 175 551	50	500	730	590	30	59	39	M16	489

Maximum offset to neutral position is 50mm for all viscoelastic dampers of type VM . . . 551 shown above. Specifications for dampers with larger offsets are available upon request.

Vertical and horizontal damping parameters are the same for VM . . . 331 and VM . . . 551

Vertical damping parameter [kNs/m]

Type	Nominal load [kN]	5 [Hz]	10 [Hz]	15 [Hz]	20 [Hz]	25 [Hz]	30 [Hz]	35 [Hz]
VM 010 ..1	5	10.2	7.7	6.6	5.9	5.4	5.0	4.7
VM 020 ..1	10	19.5	14.9	12.7	11.4	10.4	9.7	9.1
VM 030 ..1	15	29.7	22.7	19.4	17.3	15.8	14.8	13.9
VM 055 ..1	25	54.9	41.9	35.8	32.0	29.3	27.3	25.7
VM 100 ..1	40	103.1	78.7	67.2	60.0	55.0	51.2	48.3
VM 175 ..1	50	174.8	133.4	113.8	101.8	93.3	86.9	81.8

Horizontal damping parameter [kNs/m]

Type	Nominal load [kN]	5 [Hz]	10 [Hz]	15 [Hz]	20 [Hz]	25 [Hz]	30 [Hz]	35 [Hz]
VM 010 ..1	5	11.6	8.7	7.3	6.5	5.9	5.5	5.2
VM 020 ..1	10	24.0	18.1	15.3	13.6	12.4	11.5	10.8
VM 030 ..1	15	41.0	30.8	26.1	23.2	21.1	19.6	18.4
VM 055 ..1	25	100.0	75.3	63.7	56.6	51.7	47.9	45.0
VM 100 ..1	40	217.3	163.5	138.5	123.1	112.3	104.2	97.8
VM 175 ..1	50	438.3	329.9	279.4	248.3	226.6	210.2	197.4

In general offset is possible in vertical and horizontal directions.

ELASTOMERIC DEVICES

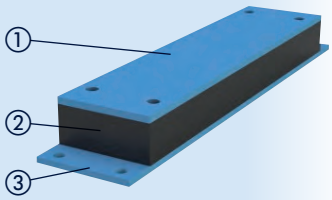


Fig. 31

① Top plate

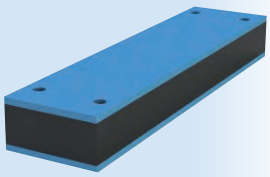
② Rubber middle section

③ Bottom plate

ISOLATION

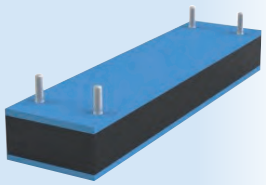
Similar to spring devices, elastomeric devices are products used for the isolation of a vibratory system from the source of vibration. A special characteristic of elastomeric devices is that they always provide some internal damping. Like spring devices they are to be positioned between the vibratory system and the source of vibration. For an efficient isolation the natural frequency of the vibratory system, defined by the elastomeric devices and the load they carry, has to be significantly lower than the main excitation frequency of the vibration source.

FIELD OF APPLICATION



There are many applications where elastomeric devices represent the most efficient way to solve a given vibration isolation problem.

A typical example is the isolation of an emergency power generator of a building like a hospital that is located at the top of building (isolation at source). The backup power supply will be instantly started when a power cut occurs. During the startup and the operation of the generator the isolation by elastomeric devices has to make sure that the operational vibration of the generator is not transmitted to the building thus not impairing the functioning of medical instruments or laboratory equipment.



Elastomeric devices are also frequently used to isolate complete rooms or even buildings from external excitations (isolation of recipient). Examples range from small rooms with sensitive production or measurement equipment to complete concert halls that are isolated against e.g. the vibrations induced by nearby railway tracks.

Another big area of application is the isolation of production equipment such as machine tools. Here the use of small standard elastomeric devices provides a very simple and cost efficient isolation solution.

The protection of complete building structures against seismic events like earthquakes is also possible with special elastomeric devices. The design of these devices requires a thorough upfront analysis of the situation.

Because of the vibrational characteristics of the VICODA elastomeric materials, the natural frequencies of the typical vibratory systems defined by the elastomeric devices and the load they carry are in the range of 6 to 20Hz, thus above the range of spring devices. This difference in natural frequency often is decisive when choosing the appropriate product for an isolation problem.

DESIGN

Like all VICODA products, also VICODA elastomeric devices are based on a modular design approach. The basic design of an elastomeric device consists of a rubber pad that is vulcanized to an upper and a lower steel plate. For the anchoring of the devices the plates may be equipped with threaded holes or bolts (Fig. 31).

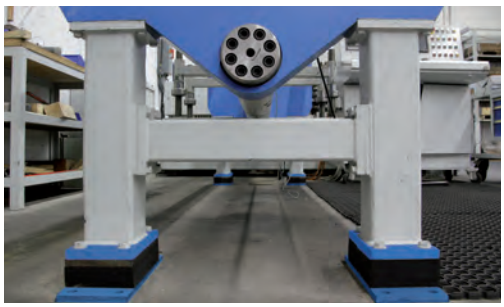
Rubber materials

The vertical stiffness of an elastomeric device and its damping characteristics are depending on the rubber material and its thickness. The typical VICODA rubber compound used for the elastomeric devices of type RR has a Shore *A* Hardness of 55 and a related damping ratio *D* of 4-5%.

However, in order to cover the wide spectrum of elastic as well as damping requirements other synthetic or caoutchouc based rubber materials are available. In aggressive weather environments chloroprene rubber provides the required resistance. When resistance against oil or grease is required acrylonitrile butadiene based rubber is available.

Arrangement of elastomeric devices

VICODA standard elastomeric devices may be used to create very different elastic properties. Like in the case of spring devices in a parallel arrangement of elastomeric devices the total stiffness of the system will be the sum of the stiffnesses of all devices in the arrangement (Fig. 32). In a serial arrangement the (vertical) thickness of the system is increased thus reducing the total stiffness of the arrangement (Fig. 33).



CONFIGURATION OF ELASTOMERIC DEVICES

For the definition of the vibration isolation solution using VICODA standard elastomeric devices the following parameters are necessary:

- the load [kN] given by the weight of the supported machine, structure or building and the load distribution
- the main excitation frequencies [Hz] of the vibration source
- installation dimensions [mm] and geometrical limitations
- special environmental requirements like exposure to sunlight, ozone, oil or grease
- operating temperatures [°C]
- special dynamic requirements.

With these data an elastomeric device solution for a vibration isolation problem can be defined using the modular VICODA product portfolio. However, like in the case of spring devices the large number of different possible rubber materials, loads etc. makes it hard for a non-expert to define the optimum solution for his specific problem.

Therefore VICODA experts are available to help you configure the optimum solution. A typical configuration process for the VICODA elastomeric devices of type RR is described on page 23.

INSTALLATION AND MAINTENANCE

When all relevant requirements have been taken into account during the configuration process, VICODA elastomeric devices do not require any maintenance during their typical service life. However, as the rubber material is subject to aging, a regular visual inspection of the devices is recommended. The intervals are depending on the rubber material used and the environmental conditions that the devices are exposed to.

For the installation of elastomeric devices, for inspections and for any kind of replacement work, VICODA experts are available.

Parallel arrangement of elastomeric devices:

Total stiffness k_{tot}:

$$k_{tot} = k_1 + k_2 + k_3 + k_4$$

Deflection:

$$x = \frac{F}{k_{tot}} = \frac{F}{k_1 + k_2 + k_3 + k_4}$$

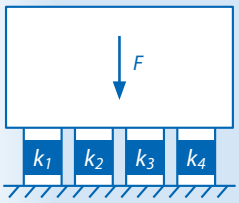


Fig. 32

Serial arrangement of elastomeric devices:

Total stiffness k_{tot}:

$$\frac{1}{k_{tot}} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \frac{1}{k_4}$$

Deflection:

$$x = \frac{F}{k_{tot}} = \frac{F}{k_1} + \frac{F}{k_2} + \frac{F}{k_3} + \frac{F}{k_4}$$

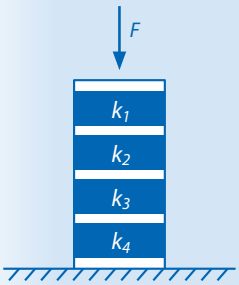


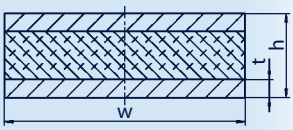
Fig. 33

ELASTOMERIC DEVICES – TYPE RR

For the VICODA elastomeric devices of type RR a standard rubber compound with a Shore A Hardness of 55 is used. They are available in 6 different widths from 40 to 200mm and with thicknesses ranging from 20 to 110mm. The length of a device is adjustable to the required loads.

VICODA offers a wide variety of elastomeric devices with different sizes and characteristics.

As an example the below table shows the standard devices of type RR:



Elastomeric Devices Type RR

VICODA Ref.	Dimensions w [mm]	h [mm]	l [mm]	t [mm]	Values are valid for 10 mm length: Stiffness $k_{x,y}$ [N/mm]	Stiffness k_z [N/mm]	Max. load $F_{x,y}$ [N]	Max. load F_z [N]	Natural freq. at F_z f_0 [Hz]	Weight Unit [kg]	Hardness [Shore A]
RR .08 041	40	20	Cut to needed length; IMPORTANT: data on stiffness, max. load and min. natural frequency relate to 10mm length)	5	27	452	100	350	18.0	0.035	55
RR .08 071	40	35		10	21	209	100	300	13.2	0.069	55
RR .08 081	40	40		10	18	115	100	220	11.4	0.071	55
RR .08 091	40	45		10	12	79	100	200	9.9	0.073	55
RR .08 101	40	50		10	11	63	100	200	8.8	0.075	55
RR .10 071	50	35		10	24	275	125	400	13.0	0.089	55
RR .10 091	50	45		10	14	121	125	320	9.7	0.094	55
RR .10 111	50	55		10	10	67	120	250	8.1	0.100	55
RR .10 141	50	70		10	6	38	100	250	6.2	0.108	55
RR .14 061	70	30		10	56	1263	180	650	21.9	0.118	55
RR .14 091	70	45		10	24	260	180	520	11.2	0.129	55
RR .14 101	70	50		10	20	157	180	400	9.8	0.133	55
RR .14 111	70	55		10	17	124	175	350	9.4	0.137	55
RR .14 131	70	65		10	13	75	175	350	7.3	0.145	55
RR .14 161	70	80		10	8	53	144	350	6.2	0.156	55
RR .20 091	100	45		15	53	1376	240	800	20.7	0.247	55
RR .20 121	100	60		15	26	372	225	730	11.3	0.263	55
RR .20 141	100	70		15	19	195	225	600	9.0	0.274	55
RR .20 151	100	75		15	16	141	215	550	8.0	0.280	55
RR .20 161	100	80		15	14	122	210	500	7.8	0.285	55
RR .20 181	100	90		15	12	84	200	500	6.5	0.296	55
RR .30 121	150	60		15	40	809	380	920	14.8	0.403	55
RR .30 161	150	80		15	28	310	380	840	9.6	0.436	55
RR .30 201	150	100		15	19	142	380	760	6.8	0.469	55
RR .40 141	200	70		15	48	902	560	1600	11.8	0.608	55
RR .40 181	200	90		15	31	350	560	1480	7.7	0.652	55
RR .40 201	200	100		15	27	218	560	1360	6.3	0.674	55
RR .40 221	200	110		15	23	191	550	1240	6.2	0.696	55

w = Width
h = Height
l = Length
t = Thickness of top and bottom plate

$k_{x,y}$ = Stiffness in horizontal direction
k_z = Stiffness in vertical direction

Configuration example:

A rotating machine running at 3,000rpms (50Hz) with a mass of 2,000kg is to be supported by four elastomeric devices of type RR, so that the surroundings are isolated from the vibration of the machine. The load distribution on the four devices is assumed to be even. Because of geometrical limitations the height of the devices should be less than 50mm.

The expected load on every device is:

$$F = \frac{2{,}000}{4}\,kg \cdot 9.81\,\frac{m}{s^2} = 4.9kN$$

Due to the geometrical limitation only the devices with a height of up to 45mm are suitable for this specific application.

When choosing a device RR.20091 with a maximum load capacity of 800N per 10mm length, the required length of the device is

$$l = \frac{4.9kN}{800N/10mm} = 61.25mm$$

the total stiffness of this device is

$$k = \frac{61.25}{10} \cdot 1{,}376\,\frac{N}{mm} = 8{,}428\,\frac{N}{mm}$$

and the related deflection of the device will be

$$x = \frac{F}{k} = 0.58mm$$

The resulting natural frequency would be

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{k}{m}} = 20.7Hz$$

As desired the natural frequency is significantly lower than the excitation frequency of 50Hz induced by the rotating machine. When choosing a device RR.14091 with a maximum load capacity of 520N per 10mm length of the device the related date would be:

$l = 94mm$
$k = 2.45kN/mm$
$x = 2mm$
$f_0 = 11.2Hz$

In this case the natural frequency of the system is lower than in the case of the RR.20091 devices. The softer support of the machine provides an even better isolation. The final choice of the elastomeric device will be driven by the geometrical limitations and the reachable natural frequencies.

TUNED MASS DAMPERS

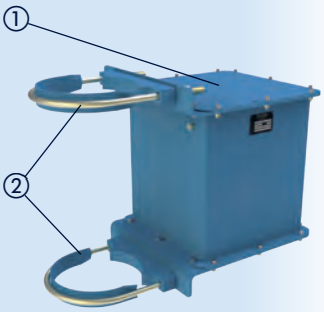


Fig. 34 - Tuned mass damper

① Oscillating unit

② Standard connection elements

ENERGY TRANSFER TO A COUPLED OSCILLATING SYSTEM – ABSORPTION

Tuned mass dampers (TMDs) are systems that can be coupled to a vibrating system and efficiently absorb energy from it. By that the vibration amplitude of the original system can be reduced significantly. TMDs are designed in a way that they efficiently absorb energy of vibrations around the natural frequency f_0 of the vibrating system. They are *passive* vibration control products as they only *follow* the motion of the original system.

FIELD OF APPLICATION

There are applications for tuned mass dampers in many industries from plant construction to civil engineering.

A typical application in plant engineering is the vibration control of piping systems in power plants or industrial plants oscillating e.g. because of rotating pumps.

An example taken from civil engineering is the control of vibrations of high-rise buildings or towers induced by wind. Another application is the control of vibrations of wind turbine generators that are due to the rotation of their blades.

These very different vibration control problems show that TMDs often are products that are specially designed for a specific application.

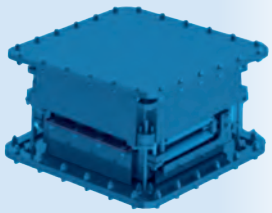


Fig. 35 - Oscillating unit

DESIGN

Tuned mass dampers in general are vibratory systems consisting of a mass, an elastic and a damping element being coupled to the original vibrating system. The process to design a TMD always follows a similar logic. However, depending on the specific application the size of the mass, as well as the sizes of the elastic and damping element of the TMD can vary in a broad range.

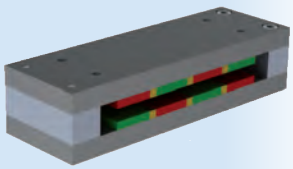


Fig. 36 - Eddy current damper

VICODA can provide TMD solutions for all typical applications. Depending on the vibration problem different designs like trolley type TMDs or pendulum TMDs are available.

TMD TYPE TP

Especially for the vibration control of piping systems **TMD** has been developed consisting of an oscillating unit and standard connection elements to the pipe (Fig. 34).

Oscillating unit

There are two preconfigured standard oscillating units available allowing to control almost all typical piping vibration problems (Fig. 35). Due to the modular design of the oscillating units its mass, its elastic and its damping parameter are adjustable. The smaller unit provides a mass of up to 100kg, the bigger unit a mass up to 350kg. Final fine-tuning of all parameters can easily be done during commissioning on site by VICODA experts thus allowing a precise calibration of the TMD and by that a highly effective reduction of the pipe vibration.

Damping elements

Damping of the TMD is provided by an eddy current damping unit (Fig. 36). Compared to conventional damping elements eddy current dampers have a number of special advantages:

- damping is independent from environmental conditions especially from temperature
- damping is independent from specific vibration characteristics of the piping system allowing efficient damping also in different operating conditions
- due to contactless energy transfer to the damping element the oscillating unit is durable, maintenance free and highly reliable.

Connection elements

LISEGA, VICODA's parent company, is the world-market leader of pipe support systems. Therefore, VICODA is benefitting from LISEGA's more than 50 years of experience in design and production of all types of pipe surrounding connection parts.

Standard connection parts are available for pipe diameters from 100 to 2000mm.

With one VICODA TMD a one or two-dimensional vibration can be controlled. For three-dimensional vibrations two tuned mass dampers are required.

An important advantage of the TMD is that it can be mounted directly onto the pipe without any rigid abutment like it is needed for e.g. a viscoelastic damper (Fig. 37). Due to this feature the tuned mass damper does not interfere with thermal movements of the piping system. It is also due to that feature that it can be perfectly used as a retrofit measure.

The VICODA TMD is designed to efficiently reduce vibrations with typical frequencies in the range of 2–25Hz and it is reliably operating in the temperature range from –40°C up to +80°C.

As pure mechanical products without any electrical components tuned mass dampers are suitable for application in areas where explosion prevention is mandatory. As passive vibration control elements not depending on power supply TMDs keep their functionality also in situations where this supply cannot be reliably secured e.g. in case of a seismic event.

For aggressive environmental conditions e.g. in a chemical plant VICODA TMDs can be delivered with special durable paints and coatings.

CONFIGURATION OF TUNED MASS DAMPERS

For the definition of a vibration control solution using VICODA TMDs the support of VICODA's engineers is mandatory. The first step in the solution process is always a detailed analysis of the vibration problem. In case of retrofit measures additional on-site mesurements are recommended.

Wölfel Beratende Ingenieure, a founding partner of VICODA has provided the respective know-how.

The optimum design will then be defined in close cooperation with the customer.

INSTALLATION AND MAINTENANCE

Installation of VICODA TMDs can be provided by VICODA's experienced service staff. Within their normal service life tuned mass dampers are maintenance-free.

Fig. 37 - Model of a TMD installed on piping system

ACTIVE DAMPING DEVICES

ACTIVE VIBRATION CONTROL

Active damping devices are coupled to a vibrating system and counteract the motion of this system. In contrast to *passive* vibration control the motion of an *active* damping device is created by acceleration of its reaction mass through actuators causing reaction forces in the original vibrating system thus reducing the vibration.

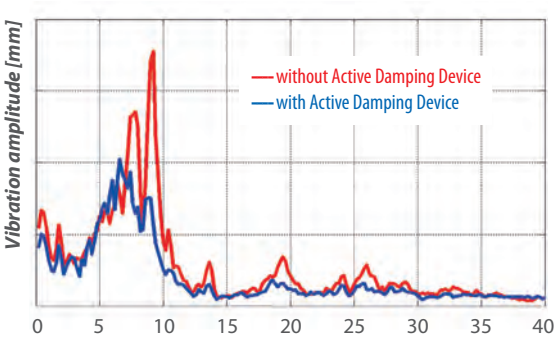


Fig. 39

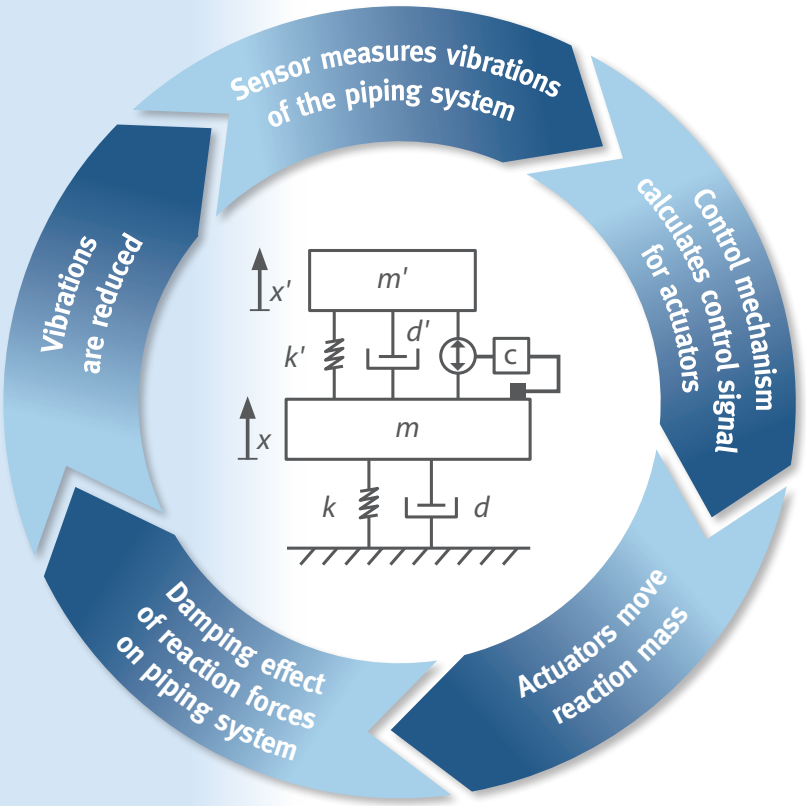


Fig. 38 - Functional principle of an active damping device

FIELD OF APPLICATION

VICODA has developed a standard active damping device for vibration control in industrial piping systems. These devices are especially suitable in cases where the vibration of the system needs to be reduced at various frequencies, e.g. due to different operational conditions of the plant.

DESIGN

The VICODA standard active damping device is equipped with sensors detecting the motion of a vibrating system and with powerful linear actuators accelerating the reaction mass of the devices (Fig. 38).

In the control unit of the device the actuator signal is continuously and automatically calculated based on the sensor signal thus securing an efficient counteraction of the damping device and reduction of the vibration. An example for the efficient vibration reduction over a wide frequency range is shown in Fig. 39.

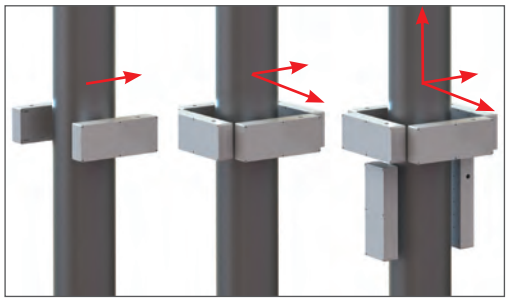


Fig. 41 - Active vibration control in one, two or three dimensions

With one basic module of the VICODA active damping device (Fig. 40) it is possible to create counteracting motion in one dimension.

Therefore multi-dimensional vibration problems require the respective number of modules (Fig. 41, 43).

Like a tuned mass damper, VICODA active damping devices can be mounted directly onto the pipe without any rigid abutment (Fig. 42) like it is needed for e.g. a viscoelastic damper. Therefore they do not interfere with thermal movements of the piping system. It is also due to that feature that they can be easily used as a retrofit measure.

Because of its special design using powerful actuators the mass of an active damping device is much lighter than the mass typically used in a comparable passive tuned mass damper.

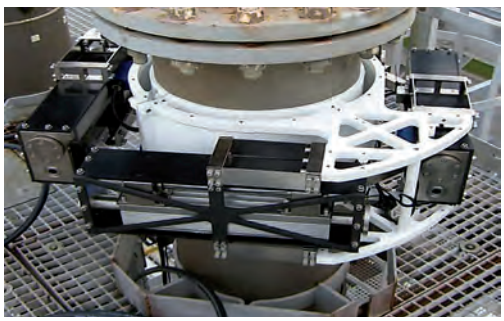


Fig. 42 - Active damping device installed directly on a gas piping system

The robust control cycle of the devices allows an easy adaptation to changing operational conditions in the plant. If required the detected information about the vibration can be transmitted to a central monitoring or control system.

For aggressive environmental conditions e.g. in a chemical plant VICODA active damping devices can be delivered with special durable paints and coatings. Devices to be used in environments where explosion prevention is compulsory are available upon request.

CONFIGURATION OF ACTIVE DAMPING DEVICES

The definition of a vibration isolation solution using active damping devices requires the support of VICODA's experienced engineers. A detailed analysis of the vibration problem and – in case of retrofit measures – on-site measurements are always the first step in the solution process.

The optimum design will then be defined in close cooperation with the customer

INSTALLATION AND MAINTENANCE

Installation and maintenance of VICODA active damping devices can be provided by VICODA's experienced service staff.

Maintenance requirements depend on the specific operating conditions of the active damping device.

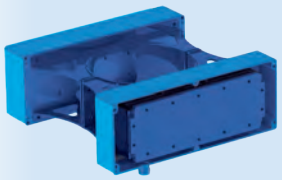


Fig. 40 - One double module of an active damping device

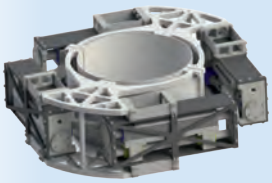


Fig. 43 - Active damping device for vibration control in two dimensions

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