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# vibrafoam | vibradyn

## Technical information about the product and about vibration isolation


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### Why vibration isolation

Industry, transport and residential construction increasingly coming closer together. This proximity results in impairments due to noise and vibrations.

### Which problems occur

Without appropriate measures, buildings, the people who live in them, machines and machine foundations or sensitive components are defenceless against vibrations from the immediate surroundings.
Undesirable or excessively powerful vibrations can also occur in buildings or industrial plant. Secondary airborne noise also increases, since structural elements such as ceilings or walls are also stimulated.

### Solution

PURASYS **vibra**foam and PURASYS **vibra**dyn provide effective protection against vibrations and shock. These high-tech PUR elastomers can be used as full surface, point or strip bearings between the structural components matching the relative component geometry or as tailor-made moulded parts. We can offer you 13 standard materials (5 for PURASYS **vibra**dyn) as well as the possibility of producing special types in many colours and thicknesses. Our team of highly qualified employees will support you or will draw up individual solutions after detailed analysis.

### Possible ways to isolate receiver and source

In vibration technology, a distinction is made between receiver and source. As a basic principle measures may be carried out on the interference source (rail operations, industrial plant), for example through the use of mass-spring systems, ballast mats or using isolating machine foundations. Isolation of vibrations can also be achieved at the receptor (buildings next to the railway, precision machines in industrial operations), for example through the use of elastic building foundations or specific isolation of certain areas or levels in the building. Source isolation is generally much more efficient but cannot always be carried out retrospectively. We can therefore offer you also effective and economic solutions for vibration isolation at the receptor.

### Benefits of vibration isolation

**• for buildings**
Reliable vibration protection for a building or for parts of a building against external interference sources and their vibrations (also insulation of footfall), improvement of market value (respectively building value), enhanced life and workplace quality and a viable solution for the future and the anticipated increase in comfort standards that will be aspired to

**• for machines**
Isolation against disruptive machine vibrations, higher precision performance, less wear, longer machine service life, better working conditions

**• for machines and industrial components**
The benefits can be many and varied. For example, units or components can run more quietly, can produce with less wear and, at the same time, can become more long-lasting and resistant against chemicals and oils. PURASYS **vibra**foam and PURASYS **vibra**dyn can be useful as a high quality seal or as a structural component tolerance compensator with extremely high resilience


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Made in Germany
0,1 mm

## "Due to its properties, PURASYS **vibra**foam is suitable for almost any application."

PURASYS **vibra**foam is a cellular elastomer made of a special kind of polyetherurethane. Elastomer springs are used in mechanical engineering and in the construction sector to isolate and/or damp vibration levels. PURASYS **vibra**foam elastomers exhibit outstanding characteristics as both pressure and compression-loaded springs.
For almost every application, there are 13 basic types of PURASYS **vibra**foam available, ranging from SD 10 to SD 1900 (Fig. 1). The desired requirements can be achieved easily through an appropriate selection of PURASYS **vibra**foam types, support surface area and construction height.

PURASYS **vibra**foam is available as mats for maximum floor coverage, but can also be obtained in the form of technical moulded parts.

If necessary, special types with exactly matched strength can be produced. This defines special properties for the material. In contrast to non-cellular elastomers, the fine cellular structure of PURASYS **vibra**foam contains enclosed volumes of gas. This makes the material volume-compressible in response to static as well as to dynamic loads. It is therefore suitable for use on large surface areas in constructions made of locally mixed concrete.

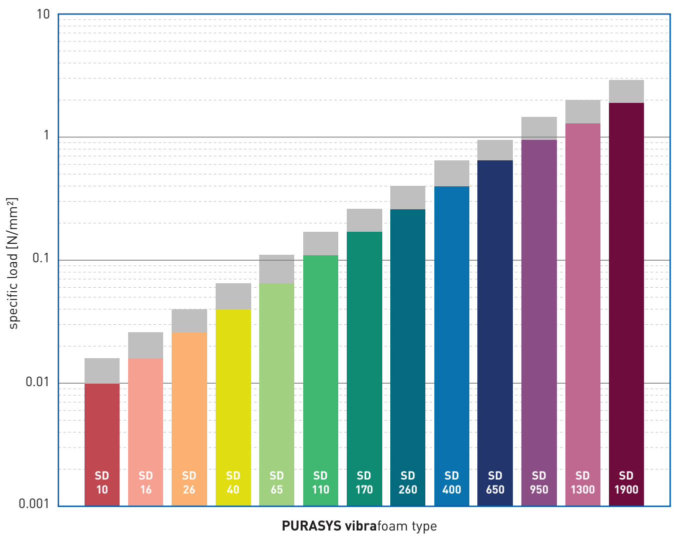



*Fig. 1: The PURASYS **vibra**foam materials*

### The static load deflection curve of
PURASYS **vibra**foam

Fig. 2 shows the quasi-static load deflection curve from a pressure test conducted on PURASYS **vibra**foam material.

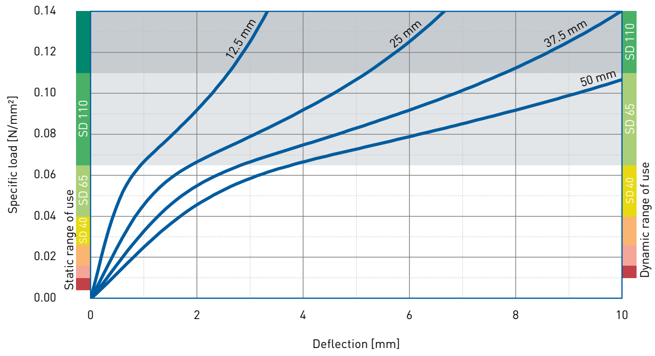



*Fig. 2: Quasi-static load deflection curve of a PURASYS **vibra**foam material (SD 65)*

Under low compression, the material exhibits an almost linear characteristics curve. The long-term static loading of these flexible bearings should lie within this range. The left scale shows the optimum static application range for each type of PURASYS **vibra**foam.

As loading on these bearings increases, the spring characteristic curve trends downwards (light-grey area). PURASYS **vibra**foam reacts in a very soft way to additional static and dynamic forces. In this dynamic application range, vibration isolation is at an optimum level. The right-hand scale indicates the optimum dynamic range for each type of PURASYS **vibra**foam.

As compression levels rise, the characteristic curve follows a progressive line (dark grey area). Due to the specific properties of PURASYS **vibra**foam, the material is unaffected by brief peak loads. The polymer structure also makes it possible, after brief high peak loads, for the material to return almost to its original position. The compression set defined in EN ISO 1856 is less than 5% for most types of PURASYS **vibra**foam (please refer to the product data sheets for more precise details).

## The dynamic properties

Fig. 3 shows the relationship between the quasi-static and the dynamic modulus of elasticity (for 10 Hz and 30 Hz) at given load levels.

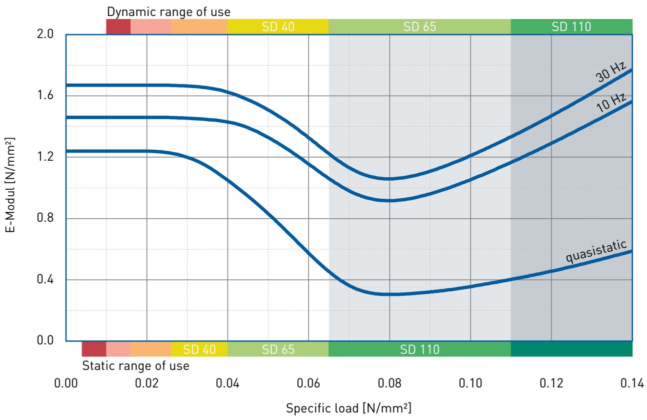



*Fig. 3: Modulus of elasticity of a PURASYS* ***vibra****foam material (SD 65)*

Due to its polymer structure, the intrinsic damping in PURASYS **vibra**foam causes the dynamic modulus of elasticity to exhibit higher values than the static modulus of elasticity. Depending on frequency and compression level, the strength reinforcement factor of PURASYS **vibra**foam materials measures 1.5 - 4.

The characteristic curve shown here for the quasi-static and the dynamic modules of elasticity indicates a minimum in the central dynamic application area. Despite slight spring compression action, the material at this minimum still exhibits optimum vibration-isolating properties.

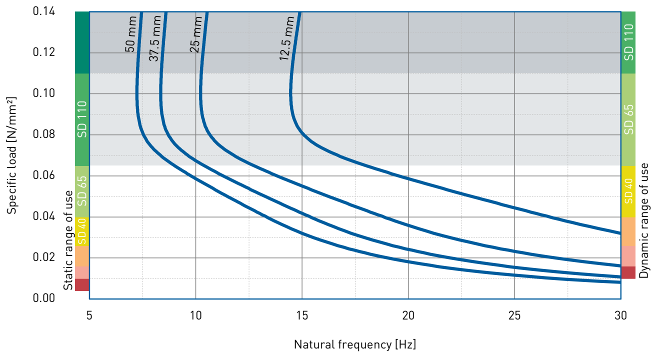
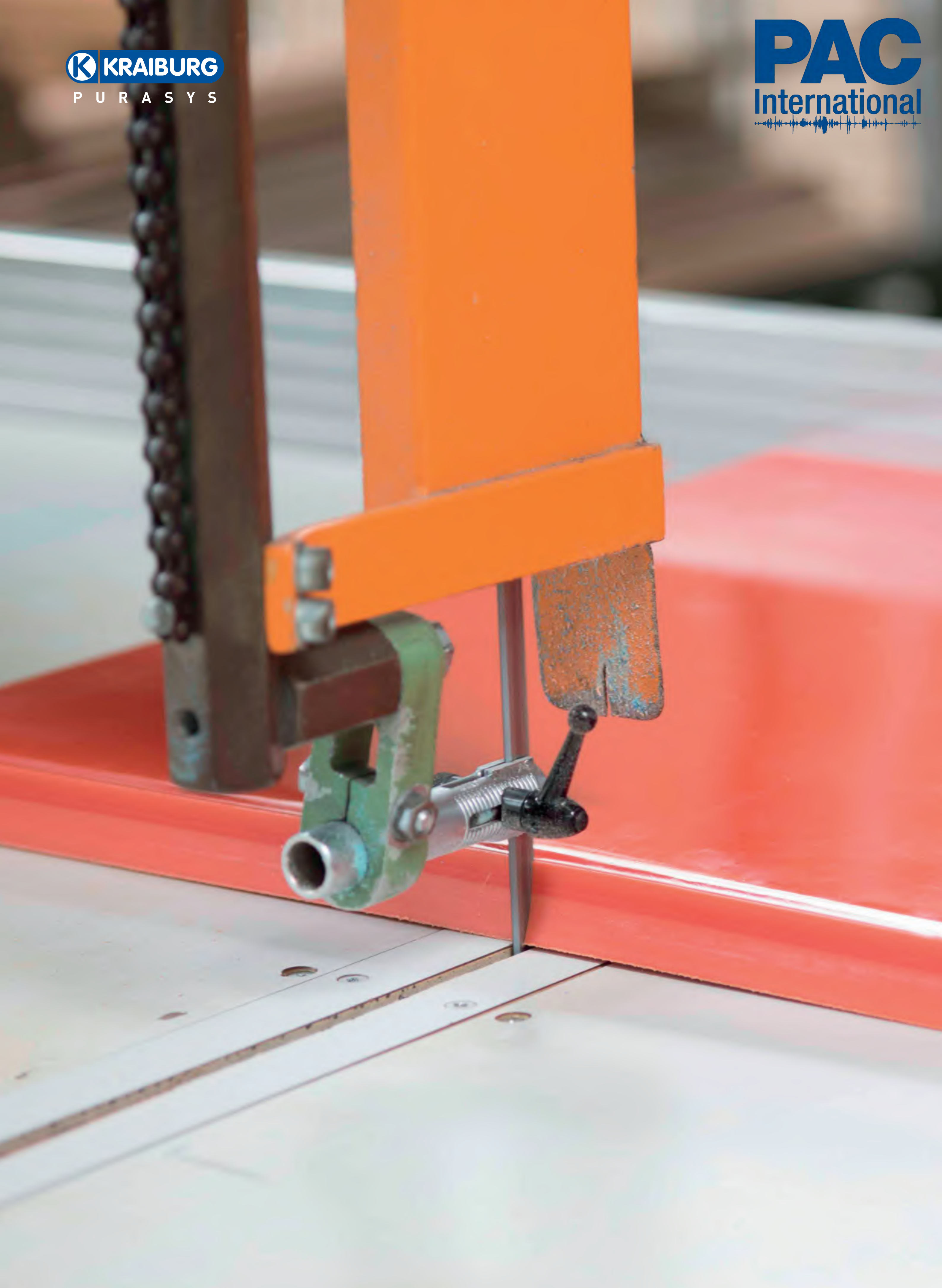



*Fig. 4: Natural frequencies of a PURASYS* ***vibra****foam material (SD 65)*

The dynamic characteristics of the modulus of elasticity is frequency-dependent. In practice, a good approximation for most applications is to select the dynamic modulus of elasticity for 10 Hz. Fig. 4 shows the computed natural frequency of a system comprising a compact mass and a flexible mounting made of PURASYS **vibra**foam, dependent on load (basis: dynamic modulus of elasticity at 10 Hz). The desired natural frequency of the system can be achieved through an appropriate choice of construction height.

## The damping characteristics

PURASYS **vibra**foam materials are damped spring elements. This means that, when PURASYS **vibra**foam materials are subjected to alternating dynamic loads, a proportion of the mechanically introduced energy is converted into heat. The damping characteristics are described by the mechanical loss factor $\eta$.
For PURASYS **vibra**foam materials, these values are between 0.09 and 0.25 (please consult the product data sheets for more precise details).



"Due to its superlative dynamic properties, PURASYS **vibra**dyn is also suitable for exceptionally challenging applications."

PURASYS **vibra**dyn is a closed-cell elastomer and it is made of a special kind of polyetherurethane. Thanks to its structure, this material absorbs almost no fluids and can therefore be used in pressing groundwater.

There are 5 basic types of PURASYS **vibra**dyn, S 75 to S 1500, to suit virtually any application scenario (Fig. 5). The desired requirements can be achieved easily through an appropriate selection of PURASYS **vibra**dyn types, support surface area and construction height.

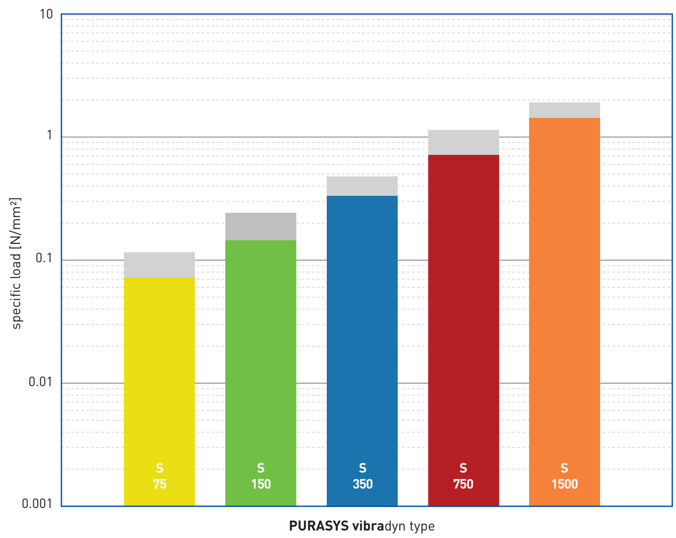
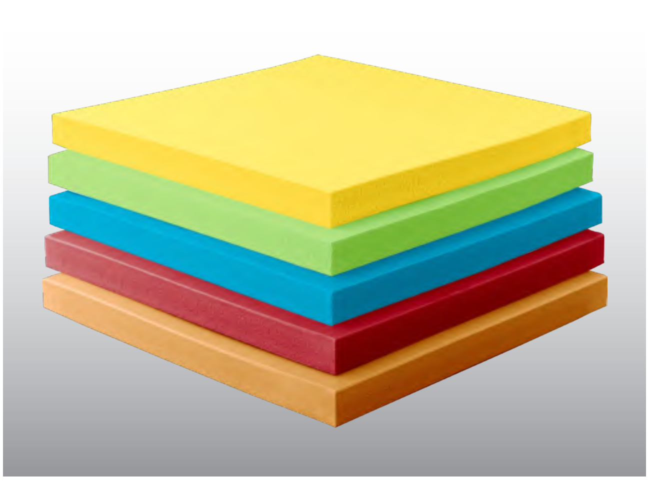



*Fig. 5: The PURASYS **vibra**dyn materials*



## The static load deflection curve of PURASYS **vibra**dyn

Fig. 6 shows the quasi-static load deflection curve from a pressure test conducted on PURASYS **vibra**dyn material.

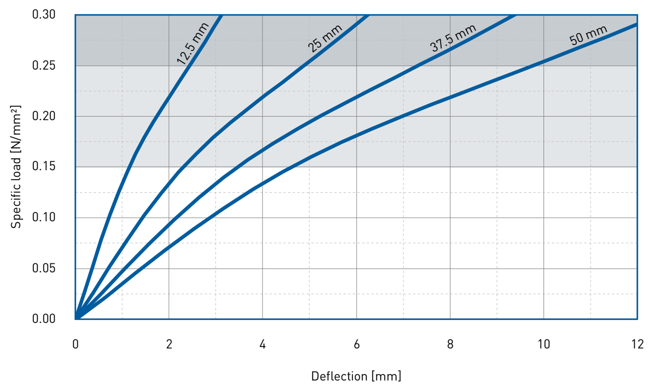



*Fig. 6: Quasi-static load deflection curve of a PURASYS **vibra**dyn material (S 150)*

As with the PURASYS **vibra**foam types, the load deflection curve of PURASYS **vibra**dyn types can be sub-divided into three areas. The linear characteristic curve in the static working area follows a ,degressive', i.e. downward-trending characteristic curve in the dynamic operating range (light grey area). At higher levels of compressive force, the characteristic curve begins to follow a ,progressive', i.e. upward-trending line (dark grey area).

## The dynamic properties

Fig. 7 shows the quasi-static and the dynamic modulus of elasticity (for 10 Hz and 30 Hz) at given load levels.

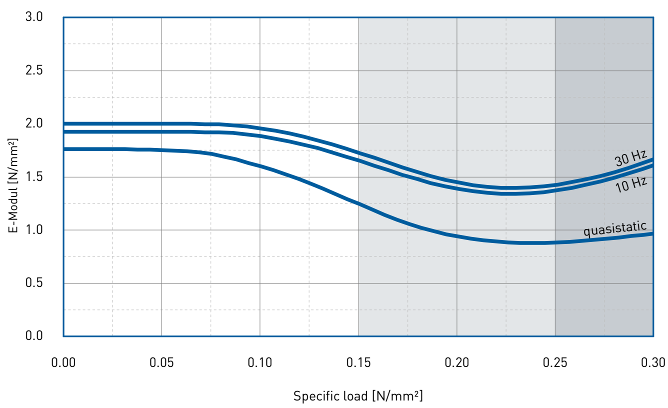



*Fig. 7: Modulus of elasticity of a PURASYS **vibra**dyn material (S 150)*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PURASYS **vibra**dyn materials exhibit very small rigidity-reinforcing factors and are therefore suitable for vibration-isolating applications, even when these involve high dynamic requirements.

Fig. 8 shows the calculated natural frequency of a system consistent of a compact mass and an elastic bearing made of PURASYS **vibra**dyn, dependent upon the loading level (basis: dynamic modulus of elasticity at 10 Hz). With PURASYS **vibra**dyn, the systems involved in vibration damping can be tuned very low. This achieves highly effective vibration isolation.

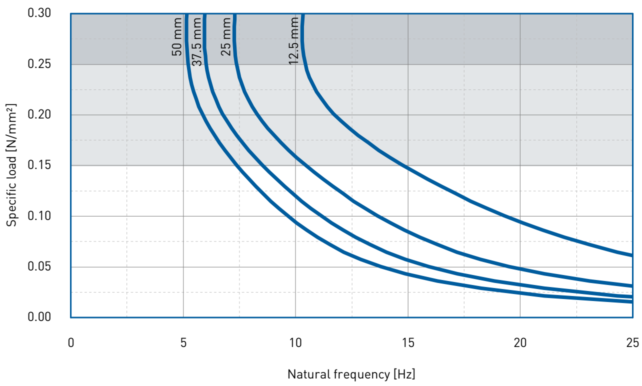


*Fig. 8: Natural frequencies for a PURASYS* ***vibra****dyn material (S 150)*

## The damping characteristics

PURASYS **vibra**dyn materials have very low levels of damping. The mechanical loss factor $\eta$ for all types of PURASYS **vibra**dyn is less than 0.06 (please refer to the product data sheets for more precise details).

## Notes

Project: "Seestraße", Zurich, Switzerland

### The shear modulus

Structural bearings made of PURASYS **vibra**foam/**vibra**dyn materials can be also subjected to shear forces. Always ensure that the shear modulus is less than the corresponding modulus of elasticity. This applies to dynamic as well as to static loadings. You can find information about these shear moduli in the relevant product data sheets. The quasi-static shear characteristic curve describes a relatively linear path.

### The form factor

The rigidity and/or the load deflection curve of the cellular elastomer is dependent in part on the volumetric compressibility level of the PURASYS **vibra**foam/**vibra**dyn material. The more compact the types of PURASYS **vibra**foam/**vibra**dyn are, the lower are their respective levels of volumetric compressibility. The parameter of form factor q (= surface subjected to load/curved surface area) makes it possible to determine the values for suspension action, dynamic modulus of elasticity and natural frequency for the prevailing geometry of the bearing. The dependent relationships between these properties and the form factor are itemised on page 3 of the product data sheets for each type of PURASYS **vibra**foam/**vibra**dyn. These figures serve as correction values to the graphs on page 2 of the data sheets.

### Static and dynamic properties when subject to continuous load

Elastic vibration bearings tend to exhibit load-dependent creepage characteristics. A continuous high level of load can alter the static and dynamic properties of an elastomer. However, the limit val-ues stipulated for PURASYS **vibra**foam/**vibra**dyn are selected for the permitted levels of load in such a way that no significant change in the dynamic modulus of elasticity does occur, even over very long periods of time.

### Influence of temperature

The operational temperature range of PURASYS **vibra**foam/**vibra**dyn materials should lie between -30°C and +70°C. The details provided in the product data sheets apply to normal climates (room temperature). Temperature-dependent changes in the dynamic modulus of elasticity at different temperature are itemised in the detailed data sheet, and must be considered in the design.

### Dependency on amplitude

The dynamic properties of PURASYS **vibra**foam/**vibra**dyn materials are only slightly dependent on amplitude (see detailed data sheet) so this factor can be treated as insignificant.

### Fire characteristics

The classification of PURASYS **vibra**foam/**vibra**dyn materials is defined in DIN EN ISO 11925-1 as Class E (EN 13501-1). There is no risk of corrosive gas fumes being created in the event of fires. The composition of these materials is similar to that of organic materials such as wood or wool.

### Resistance to environmental influences and to chemicals

PURASYS **vibra**foam/**vibra**dyn materials are resistant to water, concrete, oils, and to diluted acids and lyes. More precise information about their resistance to environmental conditions and to chemicals can be found in the data sheet 'Stability against chemical influences'


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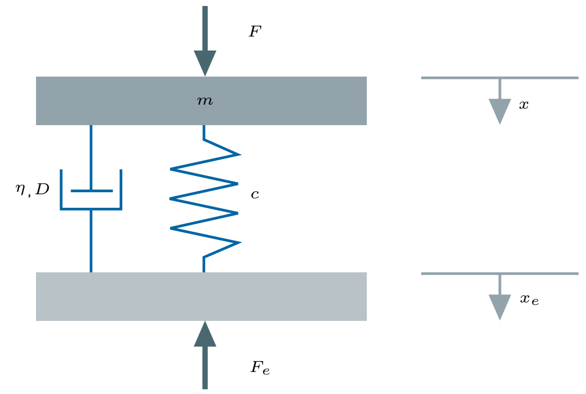
### Vibration isolation

The transmission of undesirable mechanical vibrations to the structure requiring protection can be reduced by the right choice of vibration isolation material. With the help of a damped spring, depending on the type of insulation, the source can be isolated from the receiver, or vice versa. Since PURASYS **vibra**foam/**vibra**dyn materials are 'visco-elastic' construction elements, they perform the role of a damped/slightly damped spring.

### The simple computational model

The simply physical model of a one-dimensional mass-spring system (Fig. 9) can be used to analyse many vibration problems.




| | | |
|---|---|---|
| $F$ | acting dynamic force | [N] |
| $m$ | oscillating mass | [kg] |
| $c$ | dynamic spring constant | [N/mm] |
| $F_e$ | dynamic contact force | [N] |
| $x$ | deflection of the mass | [mm] |
| $x_e$ | dynamic deflection of the abutment | [mm] |
| $\eta$ | mechanical loss factor | [ ] |
| $D$ | Lehr's damping factor | [ ] |

*Fig. 9: One-dimensional mass-spring system*

A free linear-damped oscillation is described by the following equation of motion:

**Formula 1**

$$\ddot{x} + 2 \cdot D\omega_0\dot{x} + \omega_0^2 x = 0$$

| | | |
|---|---|---|
| $\dot{x}, \ddot{x}$ | first or second derivative of deflection with respect to time | [mm/s], [mm/s²] |
| $\omega_0$ | natural angular frequency of an undamped oscillation | [1/s] |

The following relationship exists between the mechanical loss factor $\eta$ and damping factor:

**Formula 2**

$$\eta = 2 \cdot D$$

If the mass is moved out of its rest position by an external force applied for a short time, this causes free, damped oscillations to occur at natural frequency $f$ (Fig. 10). In a first approximation, the natural frequency of the damped system $f'$ is essentially equal to the natural frequency of the undamped system $f_0$ ($\eta^2/4 \ll 1$):

**Formula 3**

$$f_0 = \frac{\omega_0}{2 \cdot \pi} = \frac{1}{2 \cdot \pi}\sqrt{\frac{c}{m}} = \frac{1}{T}$$

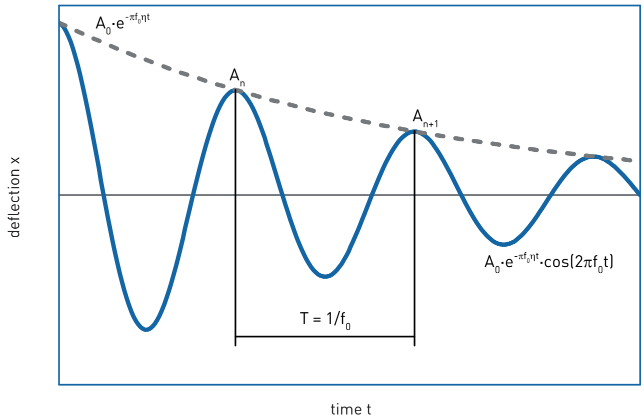



*Fig. 10: Free damped vibration*

| | | |
|---|---|---|
| $f$ | excitation frequency | [Hz] |
| $f'$ | natural frequency of a damped oscillation | [Hz] |
| $f_0$ | natural frequency of an undamped oscillation | [Hz] |
| $T$ | period duration | [s] |
| $t$ | time | [s] |

Due to the damping action, amplitude declines over time. The speed at which the amplitude diminishes depends on the damping or the mechanical loss factor. The relationship between damping and the ratio of two consecutive amplitude maximums is provided by:

**Formula 4**

$$\frac{A_{n+1}}{A_n} = e^{-2 \cdot D\pi} = e^{-\eta\pi}$$

| | | |
|---|---|---|
| $A_n$ | amplitude of the n-th oscillation | [mm] |

### Transfer function

If the mass is excited into oscillation by a periodic force $F$ with an amplitude of $\hat{F}$ and an excitation frequency $f$ this gives rise to oscillations with an amplitude of $\hat{x}$:

**Formula 5**

$$\hat{x} = \frac{\hat{F}}{c} \frac{1}{\sqrt{\left[1-\left(\frac{f}{f_0}\right)^2\right]^2 + \eta^2\left(\frac{f}{f_0}\right)^2}}$$

| | | |
|---|---|---|
| $\hat{x}$ | deflection amplitude of a driven oscillation | [mm] |
| $\hat{F}$ | amplitude of the acting dynamic force | [N] |

In its attenuated condition, the mass oscillates at excitation frequency $f$. The excessive increase in amplitude at the resonance frequency of the system depends upon mechanical damping. Due to the damping action available in PURASYS **vibra**foam/**vibra**dyn materials, this peak of amplitude is however only small in magnitude.

Vibration isolation is described by transmission function $V$. With force excitation (source insulation) the ratio of dynamic mounting force $\hat{F}_e$ and the reciprocal force excitation level $\hat{F}$ are indicated. In contrast, with travel excitation (receiver isolation), the amplitude ratio of mass $\hat{x}$ and of the substrate $\hat{x}_e$ is considered. The transfer function therefore yields the mathematical relationship between the system response and the action exerted thereon, and is dependent on frequency ratio $f/f_0$ and on the damping.

**Formula 6:**

$$V = \sqrt{\frac{1+\eta^2\left(\frac{f}{f_0}\right)^2}{\left[1-\left(\frac{f}{f_0}\right)^2\right]^2 + \eta^2\left(\frac{f}{f_0}\right)^2}}$$

| | | |
|---|---|---|
| $V$ | transfer function | [ ] |

The effectiveness of an elastic bearing is frequently quoted as an insulation efficiency rating $I$ in percent or as transmission factor $L$ in dB.

**Formula 7 and 8**

$$I = 100 \cdot \left[1 - \sqrt{\frac{1+\eta^2\left(\frac{f}{f_0}\right)^2}{\left[1-\left(\frac{f}{f_0}\right)^2\right]^2 + \eta^2\left(\frac{f}{f_0}\right)^2}}\right]$$

$$L = 20 \cdot log \left[\sqrt{\frac{1+\eta^2\left(\frac{f}{f_0}\right)^2}{\left[1-\left(\frac{f}{f_0}\right)^2\right]^2 + \eta^2\left(\frac{f}{f_0}\right)^2}}\right]$$

| | | |
|---|---|---|
| $I$ | isolation efficiency rate | [%] |
| $L$ | transmission factor | [dB] |

Fig. 11 illustrates the transmission factor for three different mechanical loss factors. An insulation effect is only provided for frequency range $f/f_0 > \sqrt{2}$.
Below the √2 multiple of resonance frequency, mechanical vibration levels are amplified by physically induced amplitude peaks.

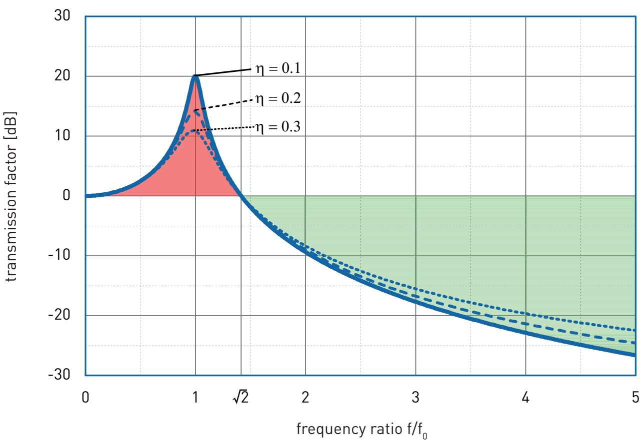



*Fig. 11: Transmission factor for various mechanical loss factors*

## Natural frequency and damping action of vibration systems with PURASYS **vibra**foam/**vibra**dyn

For the simplest design scenario, involving a vibration bearing with a type of PURASYS **vibra**foam/**vibra**dyn in accordance with the static design rating for compressive force, the computed natural frequency can be obtained by consulting page 2 of the product data sheets.

The calculation of natural frequency involves formula 3. Here, the dynamic spring constant of the bearing is determined as follows:

**Formula 9**

$$c = \frac{EA}{d}$$

| | | |
|---|---|---|
| $E$ | dynamic modulus of elasticity | [N/mm²] |
| $A$ | contact surface area | [mm²] |
| $d$ | material thickness | [mm] |

As an alternative to formula 3, the following formula can be used:

**Formula 10**

$$f_0 = 15.76 \cdot \sqrt{\frac{E}{d\sigma}}$$

| | | |
|---|---|---|
| $\sigma$ | surface compression caused by the weight of the oscillating mass | [N/mm²] |

The modulus of elasticity $E$ to be used for the corresponding surface pressure can be found on page 2 of the product data sheets. When calculating the dynamic spring constant using formula 9, and natural frequency using formula 10, ensure that the material thickness for PURASYS **vibra**foam/**vibra**dyn should be applied in unloaded condition. For sequential switching and/or for a combination of elastomer springs, the natural frequency obtained using formula 3 must be computed from the level of total rigidity.

This computational model is also valid for shear loads. However, in this case the dynamic shear modulus should be used.

The isolation level and isolation value of the elastic bearing can be calculated using formula 7 and formula 8 for the corresponding frequency ratio as a function of the prevailing mechanical loss factor.

These two parameters, dependent upon natural and interference frequency, are illustrated for the simplified case ($\eta$=0) in the detailed data sheet.

The calculation of natural frequency, assisted by static suspension action as applied to the design of forms of undamped vibration isolation (e.g. steel springs) is not suitable for calculating the natural frequency of a PURASYS **vibra**foam/**vibra**dyn bearing.

### Modelling

The modelling of a vibration system with one degree of freedom is usually enough to create a mechanical one-dimensional analogous model of the mass-spring system. This presupposes theoretically dynamic infinitely rigid and compact masses and a dynamically rigid foundation. This case generally applies to excitation masses that are very small compared to the mass of the foundation, as a first approximation. Here it is usually sufficient to know the lowest resonant frequency of the system.

When linked to structures with many other discrete individual masses and springs, additional natural frequencies can be observed. It can be advisable to extend the model in a suitable manner for this case. Particularly high levels of isolation efficiency can for example be achieved by using a dual-mass vibration source.

### Notes

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