# How to select isolators

# How to select isolators

Using linear elastic indentation hardness, a relation between the ASTM D2240 hardness and the Young's modulus for elastomers has been derived by Gent and by Mix and Giacomin. Gent's relation has the form

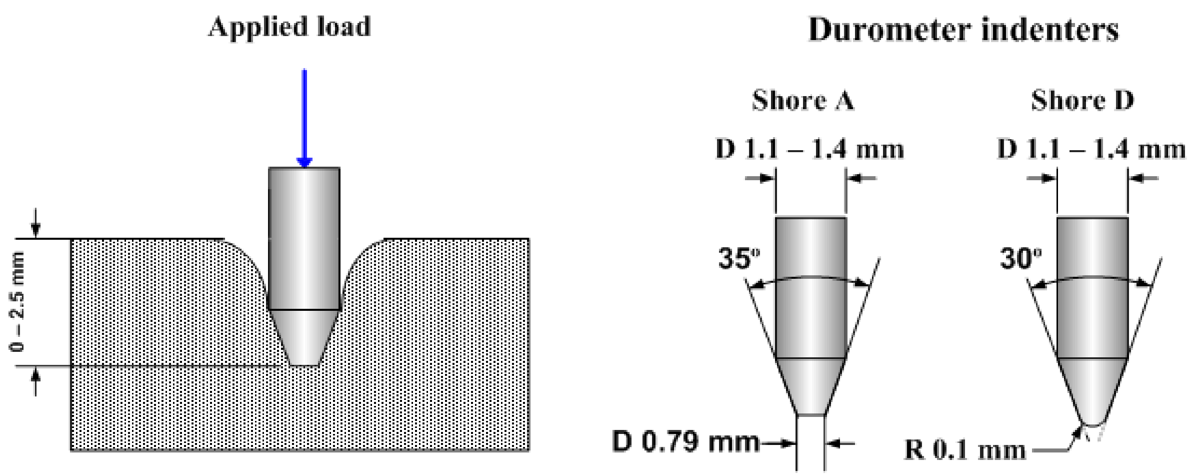
$$E = \frac{0.0981(56 + 7.62336S)}{0.137505(254 - 2.54S)}$$

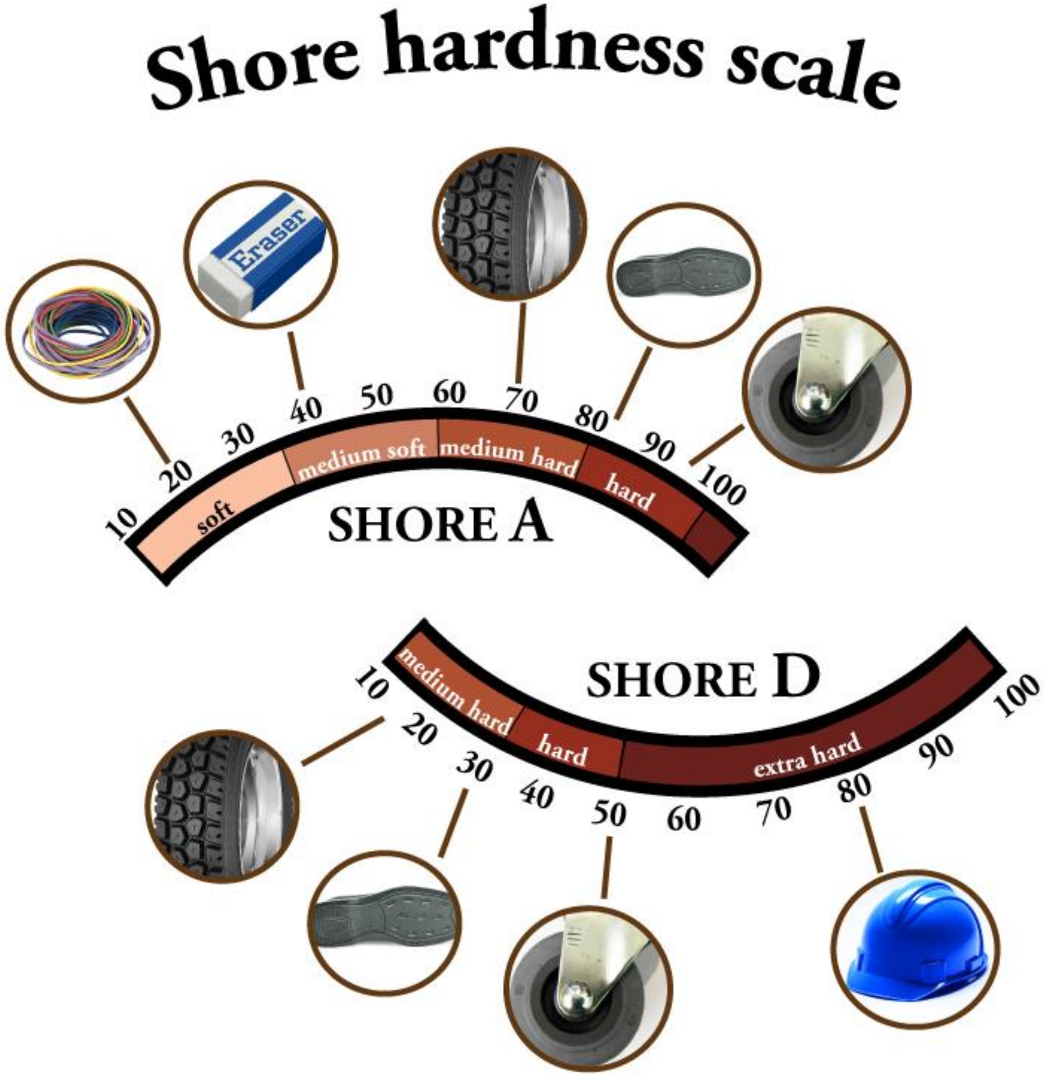
$E$ Young's modulus in MPa

$S$ ASTM D2240 type A hardness

This relation gives an infinite value for $E$ at $S=100$, but departs form experimental data $S < 40$

**Durometer hardness test**



# How to select isolators

The person who is in charge of programming the installation of a plant fitted with vibration isolation should know the various characteristics of current oscillatory systems.

Generally speaking the following materials are available when creating oscillatory systems.

- elastomers
- polyurethane (expanded)
- thermo-plastic materials
- metals (steel)
- gas (air)
- liquids (oil)[1]

[1] only when combined with elastic elements

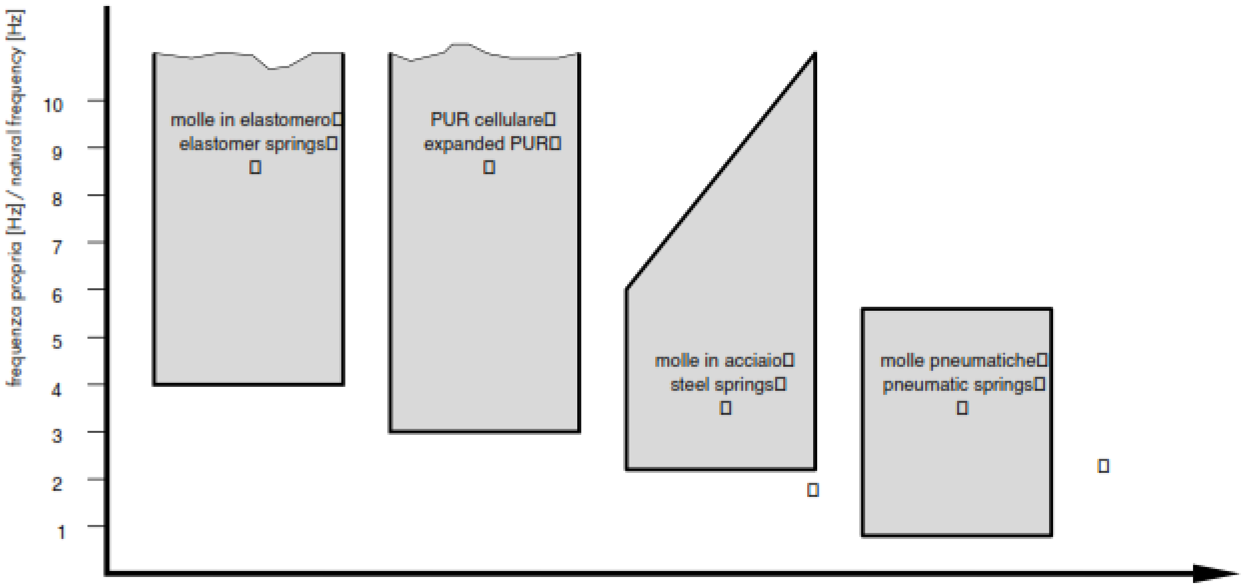
Other oscillatory systems also exist with shaped elements produced in thermo-plastic material and combinations of elastic elements with hydraulic dampers.

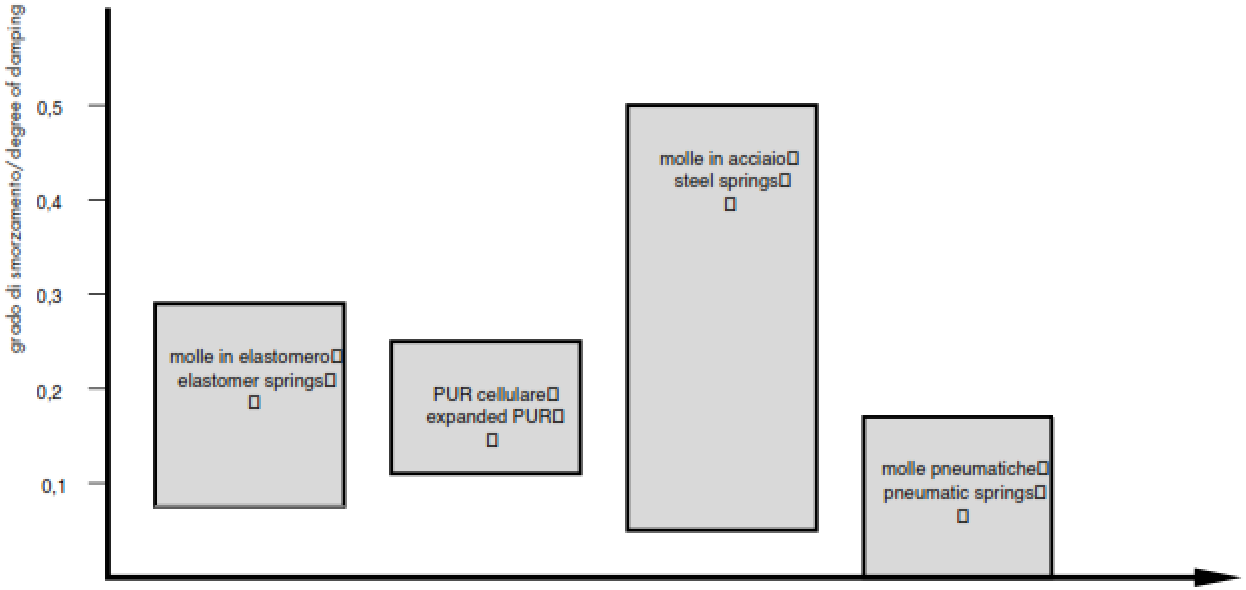
| **Sistemi oscillanti** | **Oscillatory systems** | | | |
|---|---|---|---|---|
| **Elementi/Elements** | **Gruppo/Unit** | | | |
| | **Elastomeri (gomma) Elastomers (rubber)** | **PUR (espanso) PUR (expanded)** | **Acciaio Steel** | **Aria Air** |
| puffer o tamponi/buffers | • | • | | |
| barre/bars | • | | | |
| molle/springs | • | • | | |
| zoccoli/machine mountings | • | | | |
| lastre/plates | • | • | | |
| supporti/supports | • | | | |
| coni/cones | • | | | |
| materassini sotto ballast/mattresses under ballast | | • | | |
| elementi a molle (supporti chiusi)/ elastic elements (closed supports) | | | • | |
| elementi con cavo d'acciaio/elements with steel cables | | | • | |
| elementi a molla, in acciaio/elements with steel springs | | | • | |
| sospensioni pneumatiche/pneumatic suspensions | | | | • |
| molle pneumatiche/pneumatic springs | | | | • |
| stabilizzatori/stabilizer | | | | • |

# How to select isolators

**Natural frequency**
Usually the frequency of the exciter and the mass of the plant are indicated for which only the natural frequency and the system's damping is taken into consideration as variables.

**Level of damping**
The main requirement for creating an oscillatory system is that any unacceptable motion should develop in the resonance area, i.e. the support elements must have an adequate degree of damping.

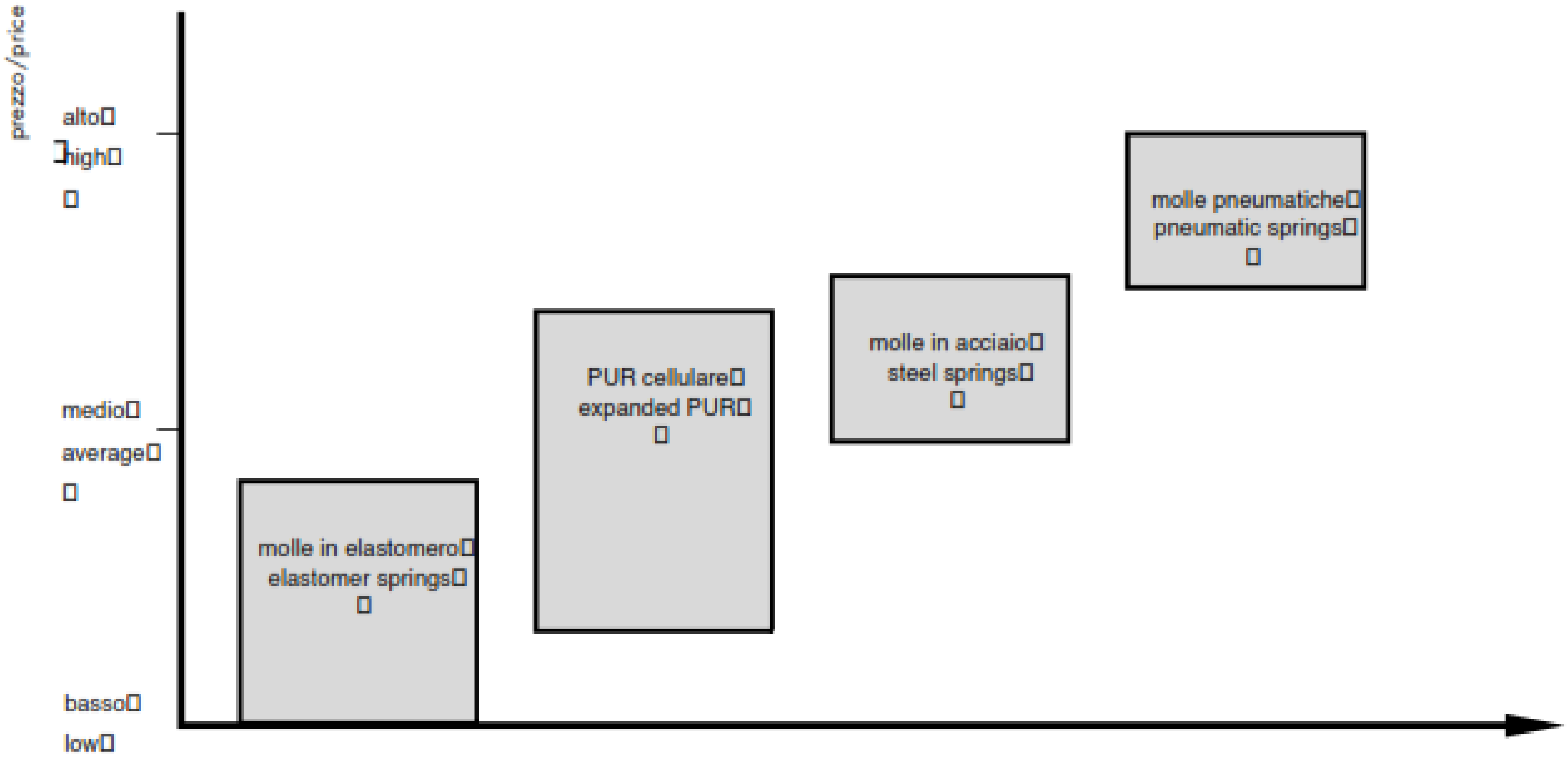



The suspension is effective for r^2 greater than 2

# How to select isolators

**Prezzo/Price**



# How to select isolators

## Elastomer elements

The elements are utilised in the following forms:
- plates
- elastomer springs
- rubber-bonded metal parts
- buffers
- laminated supports, etc.

Elastomer elements are particularly suitable in circumstances where the following principles are important:
- high level of elasticity
- limited mounting heights

# How to select isolators

**PUR elements**

Cross-linked polyurethane

The main characteristics of PUR are:

- high volumetric compressibility
- high dynamic elastic deflection up to 80% of the original height
- resistance to chemical products
- high tearing strength

The main sectors of use are:

- foundations and floating floors
- track supports (mattresses under ballast)
- dampers for cranes
- springs for use in the automobile industry

# How to select isolators

## Steel elements

The main characteristics of steel springs:

- almost every kind of elastic suspension for all types of load can be obtained using steel springs
- the elastic deflection of springs is proportional to the load
- steel springs can be calculated very accurately

Steel springs give limited damping in the material. This is often considered to be a disadvantage. Today, however, steel springs with mute damping (friction damping) or VISCO damping are available which exceed the natural damping of elastomers.

# How to select isolators

## Air elements

The suspension is effective for r^2 greater than 2

Pneumatic elements have the lowest natural frequency of all the various kinds of elements. They can be used in cases where a regular, periodical check can be carried out.

The main characteristics of pneumatic springs:

- low natural frequencies range from 0.4 to 4 Hz
- natural, almost constant frequencies which do not depend on the load
- possibility of obtaining damping with air choke
- possibility of adjusting the level by varying the air pressure

# How to select isolators – Angst+Pfister

Isolating elements are long lasting products. This, however, will only be true if the elements are selected correctly.

In the case of rubber elements it is necessary to consider the fact that, with an effect of equal force, strain will differ according to the type of stress. Most elements can be subjected to pressure, shear and torsion. Short term traction loads deriving from shock effects are acceptable. Continuous traction loads are not acceptable.

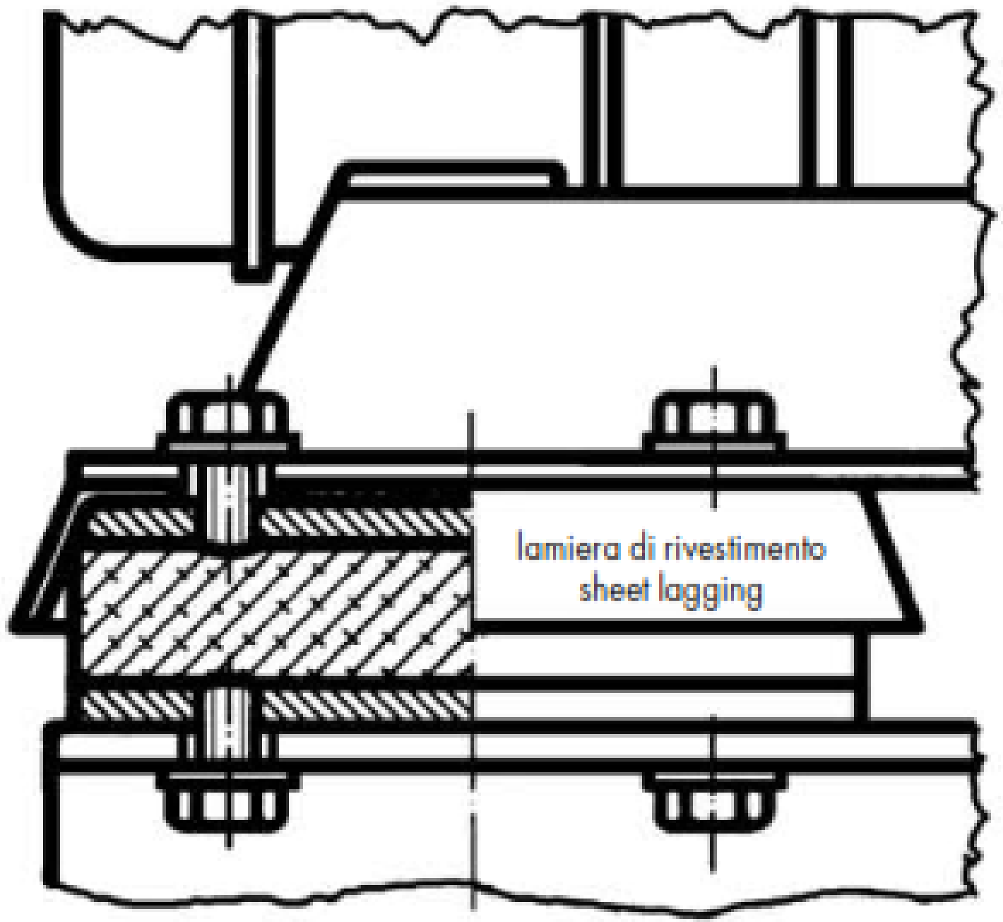
Example table containing typical characteristics of isolators.

**Valori indicativi di carico** **Approximate load values**

| Tipo di carico/Type of load | Carico ammissibile/Acceptable static | | Urto/Shock |
|---|---|---|---|
| | statico/load | dinamico/dynamic | |
| | $N/mm^2$ | $N/mm^2$ | $N/mm^2$ |
| compressione/pressure | 0,5 | ± 0,125 | 2,0 |
| taglio/shear | 0,2 | ± 0,05 | 0,6 |
| trazione/traction | – | – | 1,5 |
| torsione/torsion | 0,3 | ± 0,075 | 0,9 |
| compressione/taglio (45°) pressure/shear (45°) | 0,5 | ± 0,125 | 2,0 |

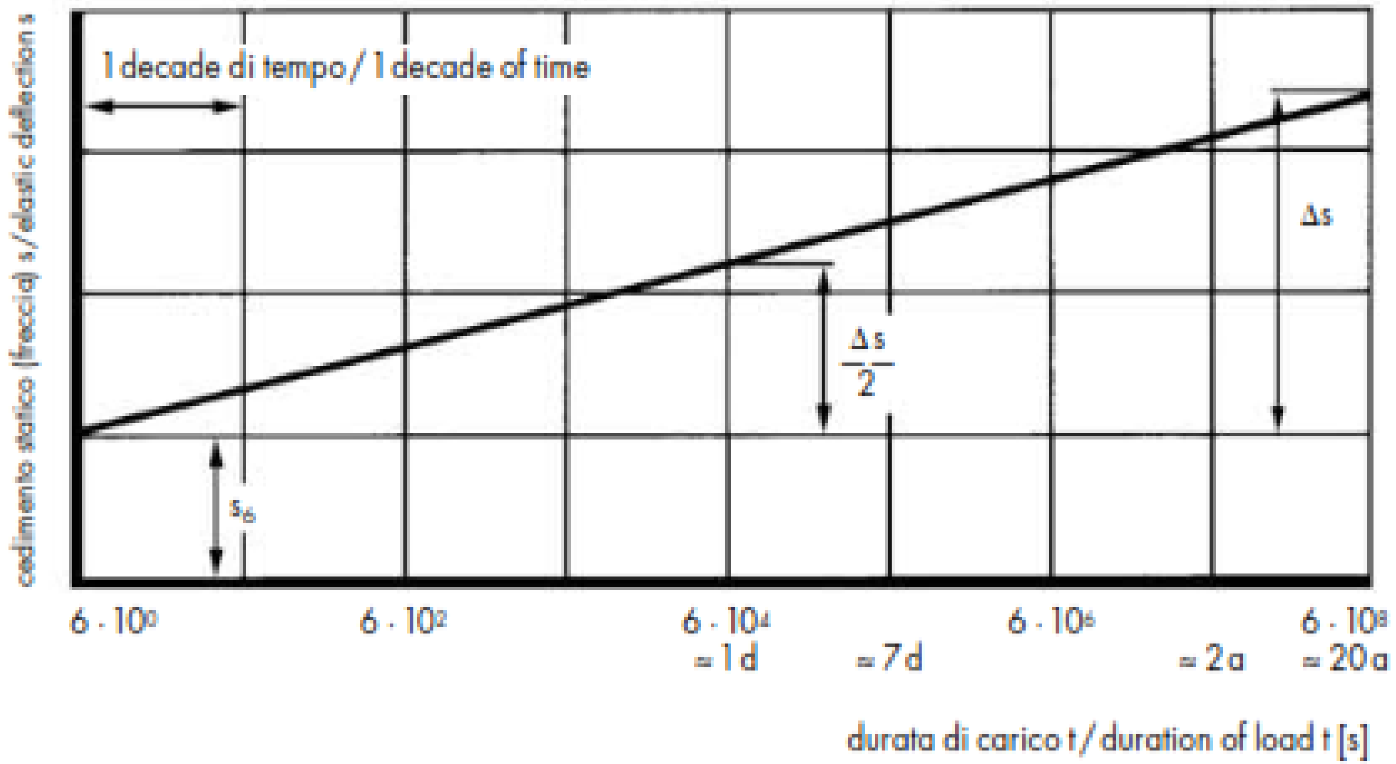
# How to select isolators – Angst+Pfister

NR based damping elements are not resistant to the actions of oil, grease, fuel or other chemical products. These elements should be protected by sheathing.



# How to select isolators – Angst+Pfister

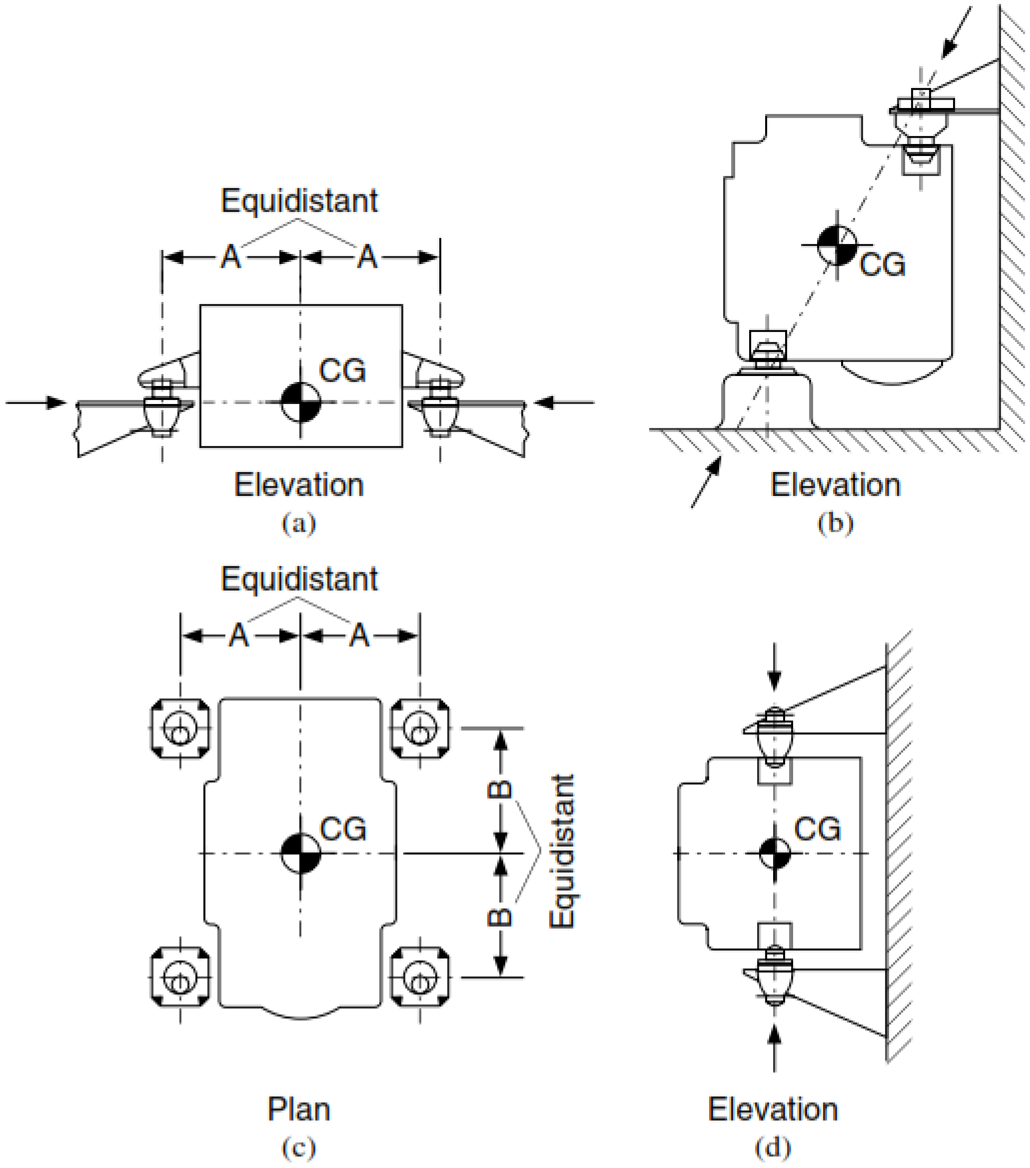
"Creep" is a characteristic common to elastomer elements. This can be attributed to strain caused by the effect of a load which does not return completely to its original position. In practice, the increase in elastic deflection caused by creep in isolating elements is, in most cases, negligible.



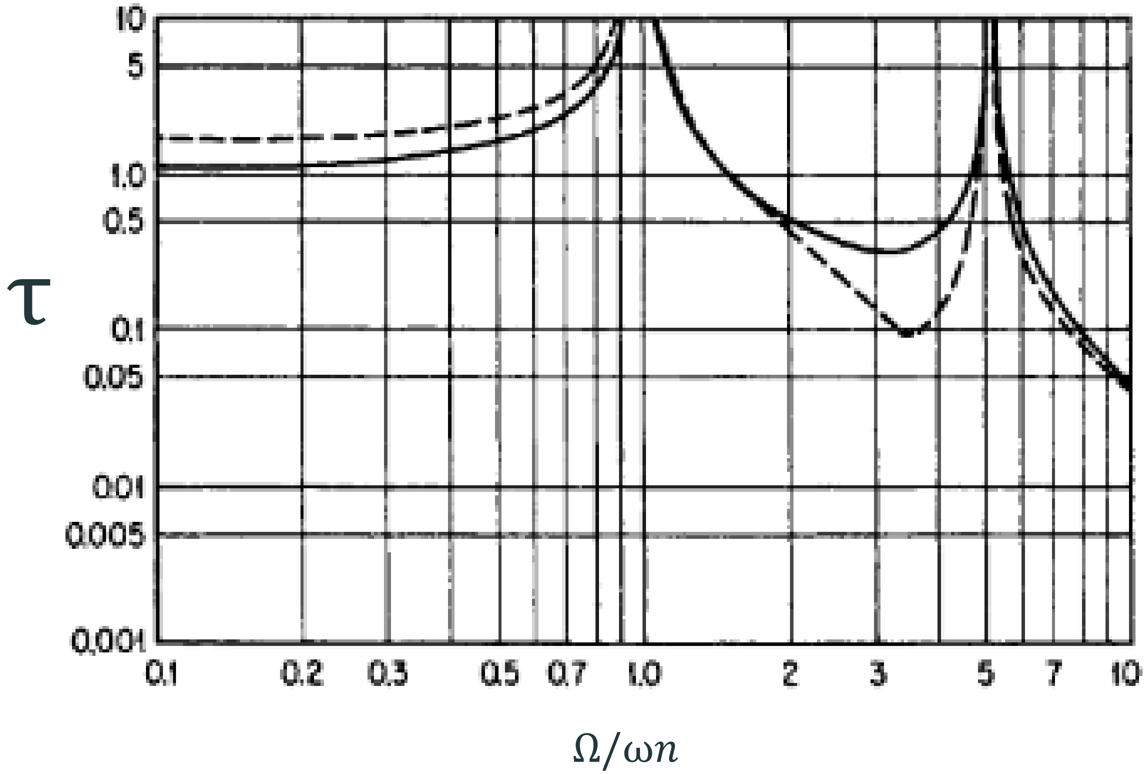

$s_6$ = cedimento statico dopo 6 secondi
$t$ = durata di carico [s] o (decadi)

$s_6$ = elastic deflection after 6 seconds
$t$ = duration of load [s] or (decades)

# How to select isolators




When the equipment and the isolator system have several degrees-of-freedom and the isolators are located in such a manner that several natural modes of vibration are coupled, it becomes necessary to consider the contribution of the several modes in determining the motion transmitted from the support to the mounted equipment or the force transmitted from the equipment to the foundation.



# How to select isolators

Step 1. Required isolation efficiency. First, indicate the percentage of isolation efficiency that is desired. In general, an efficiency of 70 to 90 percent is desirable and is usually possible to attain.

Step 2. Transmissibility. Determine the maximum transmissibility $T$ of the system at which the required vibration isolation efficiency of Step 1 will be provided (from chart or formula)

Step 3. Forcing frequency. Determine the value of the lowest forcing frequency, i.e., (the frequency of vibration excitation). The lowest forcing frequency is used because this is the worst condition. If a satisfactory value of isolation efficiency is attained at this frequency, the vibration reduction at higher frequencies will be even greater

Step 4. Natural frequency. Find the natural frequency of the isolated system (from chart or formula) required to provide a transmissibility determined in Step 2 for a forcing frequency determined in Step 3

Step 5. Static deflection. Determine the static deflection (form chart or formula) required to provide a natural frequency of Step 4

Step 6. Stiffness of isolation system. Calculate the stiffness $k$ required to provide a natural frequency determined in Step 4

Step 7. Stiffness of the individual vibration isolators. Determine the stiffness of each of the $n$ isolator depending on whether the vibration isolators are in parallel or in series

Step 8. Load on individual vibration isolators. Now calculate the load on each individual isolator

Step 9. Isolator selection. From a manufacturer’s catalogue select a vibration isolator which meets the stiffness requirement determined in Step 7 and which has a load-carrying capacity equal to the value obtained in Step 8

# Design formula consideration

The level of isolation is defined as *1-T*. It was noted that in the isolation region *($r^2>2$)* the transmissibility decreases (hence, the level of isolation increases) as the damping ratio ζ decreases.

$$T = \sqrt{\frac{1+\left(2\zeta\,\omega/\omega_n\right)^2}{\left(1-\omega^2/\omega_n^2\right)^2+\left(2\zeta\,\omega/\omega_n\right)^2}} \xrightarrow{\text{For small } \zeta} \boxed{T = \left|\frac{1}{1-\omega^2/\omega_n^2}\right|}$$

Isolator stiffness

$$\omega^2/\omega_n^2 = \omega^2 m/k \xrightarrow{\text{For small } \zeta} \boxed{k = \frac{\omega^2 mT}{\left(1+T\right)}}$$

Static deflection of the isolator

$$\delta_s = \frac{mg}{k} \xrightarrow{\text{For small } \zeta} \boxed{\delta_s = \left(1+T\right)\frac{g}{\omega^2 T}}$$

It is possible to easily compute k and the static deflection of the isolator using these simplified equations. As a matter of fact both quantities are function of T (design data I want to obtain), omega (excitation frequency) and of the mass of the equipment we want to suspend.

# How to select isolators – Angst+Pfister

**Isolating elements**

In practice, it is very difficult to calculate isolating elements because, in most cases, the necessary basic information regarding forces of equilibrium, the position of the centre of gravity, the rigidity of the machine and the acceptable oscillation speeds are not available. Moreover, these influences can vary considerably even in machines of the same structure.

In order to understand better the relationships, the following calculations refer to a single mass oscillator with excitation of harmonic force and an extremely rigid installation position (something which never actually happens). This simplification is, however, acceptable for many plants.

The exciter frequency $f_{err}$ of the supported object is always a decisive factor in establishing the efficiency of an isolating support where vibrations exist. The necessary natural frequency $f_0$ of isolating elements can be calculated by the degree of efficiency of isolation i indicated or recorded.

When calculating the degree of efficiency of isolation it is necessary to bear in mind that one hundred per cent isolation is not possible. The economic limit falls between 80% and 95%. Better levels of isolation are possible using specific applications for supplementary foundations, pneumatic supports, etc.

Theory

$$T = \frac{1}{1-\omega^2/\omega_n^2} \qquad i = 100\left(1-T\right)$$

# Example of isolator

**Zoccolo A + P** — **A + P machine mounting**

| Cod. art. / Art. no. | Modello / Model | Durezza / Hardness (Sh A) | Portata / Load capacity $F_z$ (kg) | Forza di taglio / Shear force $F_{x,y}$ (N) | Cedimento statico / Elastic deflection $s_z$ (mm) | Cedimento statico / Elastic deflection $s_{x,y}$ (mm) |
|---|---|---|---|---|---|---|
| 85221101 | 110 | 45 ±5 | 760 | 8 500 | 3,5 | 3,0 |
| 85221102 | 110 | 60 ±5 | 1 480 | 14 450 | 3,5 | 3,0 |
| 85221103 | 110 | 70 ±5 | 2 100 | 20 250 | 3,5 | 3,0 |

**Materiali:**
– elemento in elastomero: NR, nero
– corpo metallico: acciaio fosfatato (leggermente lubrificato)
– parti metalliche di collegamento: acciaio fosfatato (leggermente lubrificato)
**Esecuzione:** con regolazione d'altezza

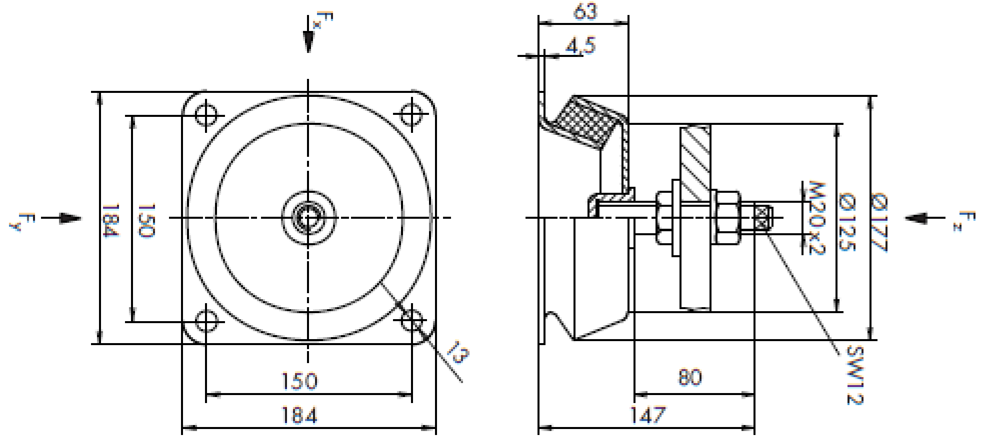
**Applicazioni:**
Elementi di impiego universale per il supporto antivibrante di macchine ed apparecchi di tutti i tipi.

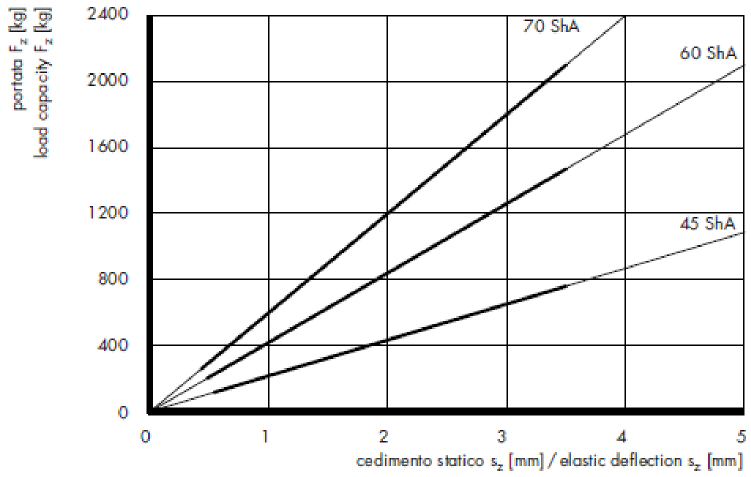
**Attenzione:**
La rigidità al taglio degli elementi è maggiore della rigidità verticale.

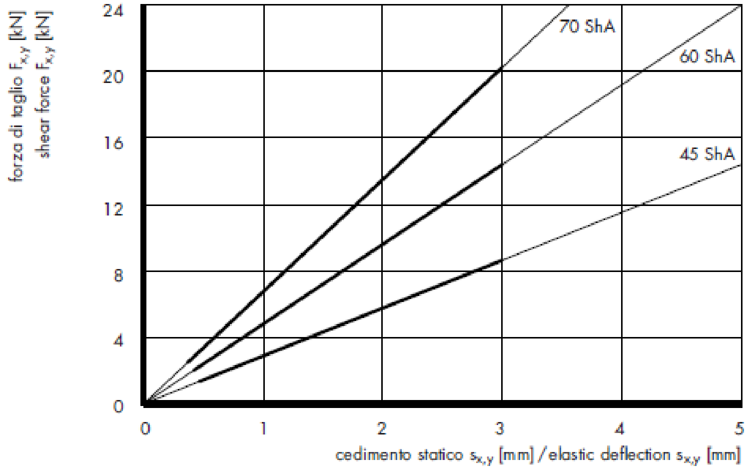
**Materials:**
– elastomer element: NR, black
– metal body: galvanised steel (slightly lubricated)
– metal connecting parts: galvanised steel (slightly lubricated)
**Execution:** with height adjustment

**Applications:**
Universally used elements for the vibration-proof support of machines and apparata of all types.

**Attention:**
The shearing strength of the elements is higher than the vertical strength.



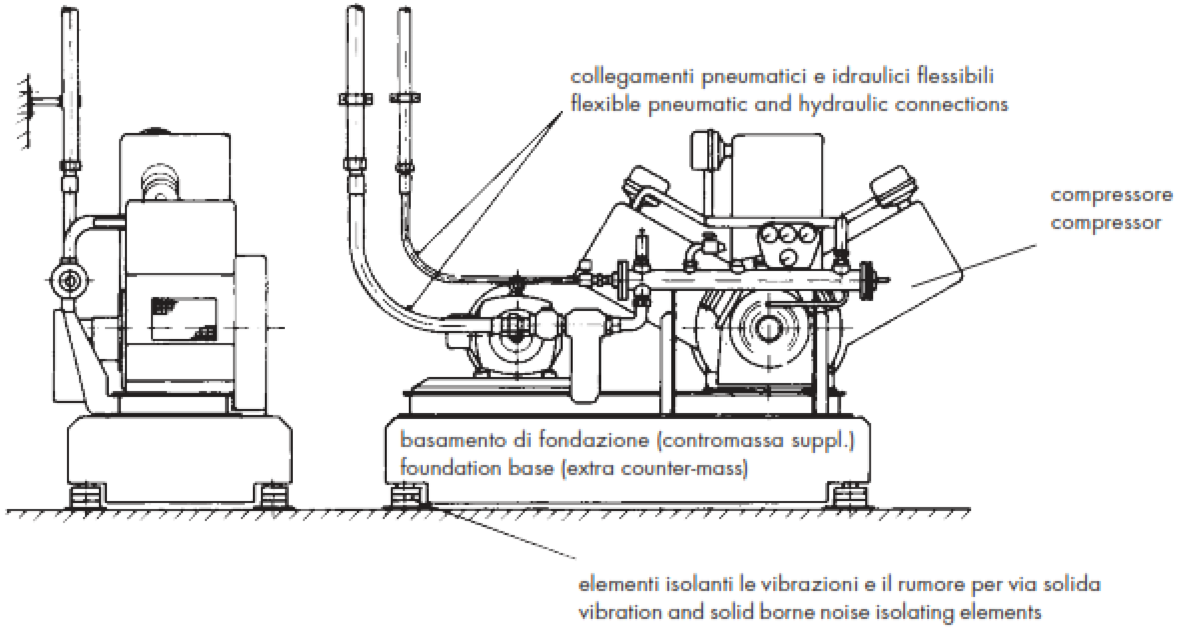



# How to select isolators – Angst+Pfister

**Support element**

A 3-cylinder compressor must rest on support elements as shown in the diagram below:




Data:

- mass of the electric motor with accessoires $m_1 = 20\,kg$
- mass of the compressor $m_2 = 400\,kg$
- number of revs ot the compressor $n_2 = 900\,min^{-1}$
- number of revs of the motor $n_1 = 1450\,min^{-1}$
- number of damping elements 6
- isolation effect 90 %

To be obtained:

- counter-mass requested a foundation base
- centre of gravity of the plant
- layout of the damping elements
- type of element
- measures to be taken to prevent the transmission of vibrations via the cooling piping or forced air piping.

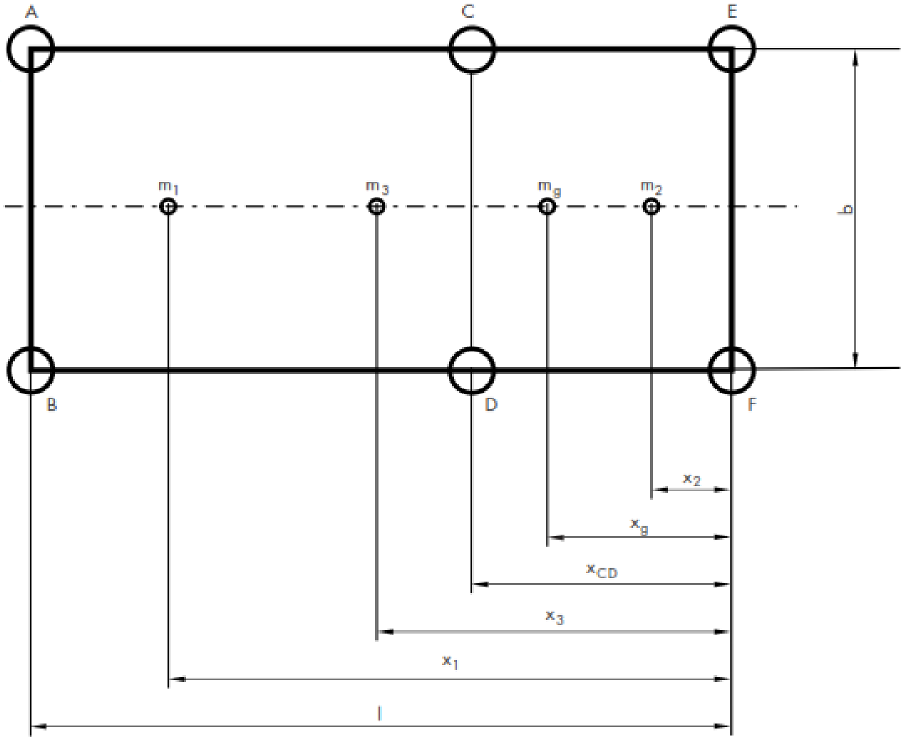
# How to select isolators – Angst+Pfister

**Definition of extra counter-mass**

The amplitude of oscillation of a supported plant is reduced thanks to extra counter-mass. In the case of slow-rotating compressors, in particular, a foundation base equal to three to five times the weight of the compressor will be required.

Pre-calculated data:
concrete foundation with the following measurements:
length $l = 2200$ mm
width $b = 1100$ mm
thickness $h = 300$ mm
density of the reinforced concrete $\rho = 2.3$

the foundation will have a mass of $m_3$
$m_3 = l \cdot b \cdot h \cdot \rho$
$m_3 = 22 \cdot 11 \cdot 3 \cdot 2.3 = 1669.8$ kg




**Definition of the layout of the support elements**

Calculation of the position of the general centre of gravity:
$x_1$ = distance from the centre of gravity per $m_1$ = 1600 mm
$x_2$ = distance from the centre of gravity per $m_2$ = 200 mm
$x_3$ = distance from the centre of gravity per $m_3$ = 1100 mm

Weight of the plant:
$m_g = m_1 + m_2 + m_3$
$m_g = 20 + 400 + 1669.8 = 2089.8$ kg

Position of the centre of gravity:

$$x_g = \frac{(x_1 \cdot m_1) + (x_2 \cdot m_2) + (x_3 \cdot m_3)}{m_g}$$

$$x_g = \frac{(1600 \cdot 20) + (400 \cdot 200) + (1100 \cdot 1669.8)}{2089.8} = 932.5 \text{ mm}$$

Forces balance with respect to plane EF

# How to select isolators – Angst+Pfister

**Layout of oscillating elements**

According to its function the plant must be supported by 6 elements. The layout of the elements must permit an even distribution of the load on all the elements.

Load for each element, in the presence of 6 elements:

$$m_E = \frac{m_g}{6} = \frac{2089.8\ \text{kg}}{6} = 348.3\ \text{kg}$$

$$\sum M_{FE} = 0 = 2 \cdot m_E \cdot l - m_g \cdot x_g + 2 \cdot m_E \cdot x_{CD}$$

Moment balance with respect to the line FE (mA and mB are at distance l , 2 masses at unknown distance CD, 2 masses in correspondence of FE which do not make contribution and the barycentric mass)

$$x_{CD} = \frac{-2 \cdot m_E \cdot l + m_g \cdot x_g}{2 \cdot m_E} = \frac{-2 \cdot 348{,}3 \cdot 2200 + 2089{,}8 \cdot 932{,}5}{2 \cdot 348{,}3} = 597{,}5\ \text{mm}$$

Considering i=90%, the obtained transmissibility is of 0,1 $\Longrightarrow$ $i = 100\left(1 - T\right)$

Considering a transmissibility of 0,1 and an excitation frequency equal to the one of the compressor (that is the lower of the two and so it is the most critical), we obtain that the natural frequency of the 348 kg mass placed on the isolator has to be of 5Hz $\Longrightarrow$ $T = \dfrac{1}{1 - \omega^2 / \omega_n^2}$

# How to select isolators – Angst+Pfister

**Elemento a molle GERB® S3Q, con Sordino**

**Springs element GERB® S3Q, with Sordino**

| Cod. art. / Art. no. | **Modello / Model** | **Altezza libera / Height without load $H_u$** mm | **Altezza sotto $F_z$ / Height below $F_z$ $H_z$** mm | **Portata / Load capacity $F_z$** kg | **Costante elastica verticale / Vertical elastic constant $c_v$** N/mm | **Costante elastica orizzontale / Horizontal elastic constant $c_h$** N/mm | **Pre-compressione / Pre-compression** mm | **Peso / Weight** kg |
|---|---|---|---|---|---|---|---|---|
| 12.2155.0321 | S3Q-241 S | 66 | 51 | 280 | 140 | 210 | 5 | 2,8 |
| .0322 | S3Q-242 S | 66 | 51 | 310 | 160 | 250 | 4 | 2,9 |
| .0323 | S3Q-243 S | 66 | 51 | 340 | 190 | 280 | 2 | 2,9 |
| .0324 | S3Q-244 S | 66 | 51 | 400 | 240 | 340 | 2 | 2,8 |
| .0325 | S3Q-245 S | 66 | 53 | 420 | 280 | 420 | 2 | 3,1 |
| .0326 | S3Q-246 S | 66 | 54 | 500 | 350 | 530 | 2 | 3,1 |
| .0327 | S3Q-247 S | 66 | 54 | 580 | 410 | 580 | 2 | 3,1 |
| .0328 | S3Q-248 S | 66 | 53 | 690 | 490 | 720 | 1 | 3,1 |

The following element was chosen:
GERB® spring element,
model S3Q-244S,
Art. no. 12.2155.0324

Using the SDOF systems equation for the estimation of the natural frequency ($\omega n = \sqrt{\left(\frac{k}{m}\right)}$ ), it is possible to obtain the isolator stiffness necessary to sustain the 348 kg mass with natural frequency of 5 Hz. The selected isolator is the .0324. Using the SDOF equation it results 343 N/mm.

# How to select isolators – Angst+Pfister

**Isolating elements**

Electrical control apparatus must be mounted onto a transport system. Isolation must be provided in order to prevent problems of electronic nature.

Known data:
- weight of the control apparatus m = 60 kg
- number of fix points: 4
- exciter frequency of the transport system $f_{err}$ = 1450 min-1
- desired isolating effect, very good (> 80%)

To be found:
an isolating element to be fixed to the wall

**Solution**

The solution required in this case is of the passive isolation type, which must be capable of protecting the electronic apparatus from potential external disturbance.
Use of a wall fixture means that the elements to be isolated must be chosen according to shear.

**Definition of the load each fixing element**

$F = m \cdot g = 60 \cdot 9.81 \approx 600\,N$

$F_{1\cdot 4} = \frac{F}{4} = \frac{600}{4} = 150\,N$

The formula used for the degree of isolating efficiency, broken down on the basis of $f_0$, will give the value of natural oscillation required for a degree of isolation of 80%:

$$i = 100(1-T) \qquad T = \frac{1}{1-\omega^2/\omega_n^2}$$

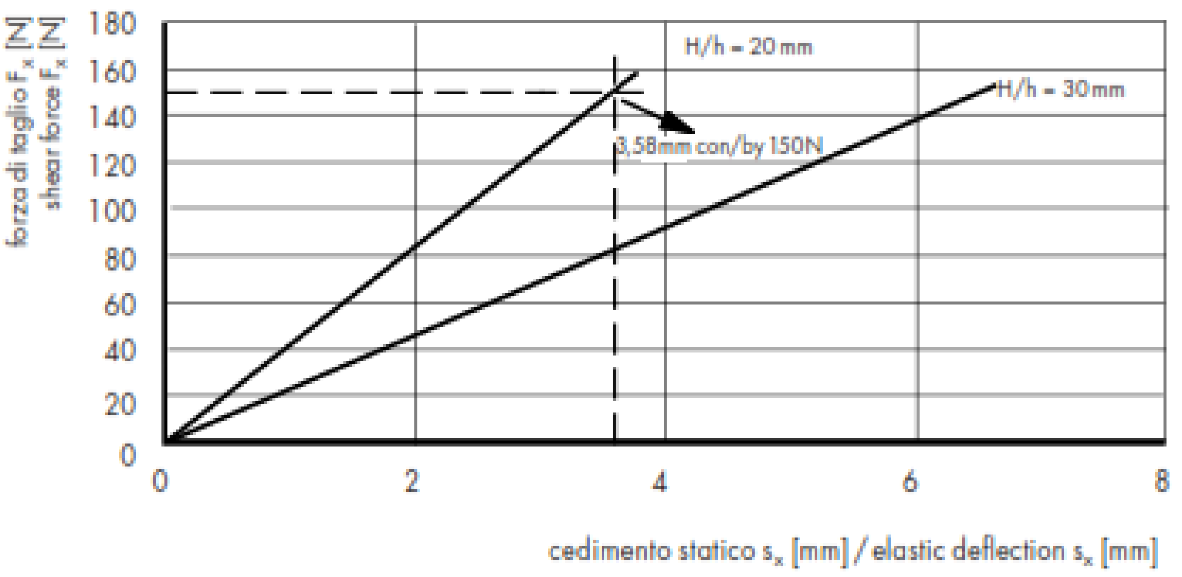
From which follow that the natural frequency of the system has to be of about 12Hz.

The static deflection results of about 1,72mm ➡ $\omega_n = \sqrt{g/\delta_{st}}$

# How to select isolators – Angst+Pfister

The following has been chosen:
cylindrical buffer model A
Art. no. 12.2001.6903
hardness: 57 Shore A
dimensions: Ø 30 x 20 mm
threaded pins: M8 x 20mm

According to their elastic shear characteristic, with a load of 150 N, the selected cylindrical buffers will produce an elastic deflection $s_S$ of 3.58 mm.




A new natural frequency arises from the choice of the isolators:

$$\omega_n = \sqrt{g / \delta_{st}}$$

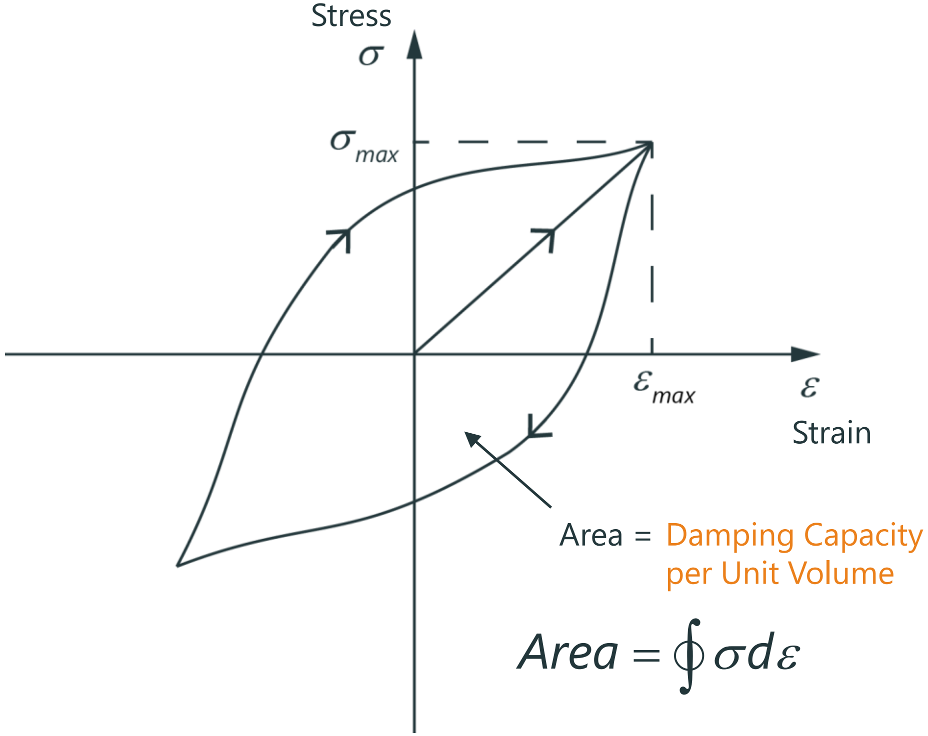
Moreover from the choice of the isolators arises a new transmissibility and a new isolation degree of about 87%, that is greater than the one requested.

$$T = \frac{1}{1 - \omega^2 / \omega_n^2}$$

$$i = 100(1 - T)$$

# Vibration isolators – experimental characterization

# Damping in viscoelatic materials




$$Area = \oint \sigma d\varepsilon$$

Internal damping of materials originates from the energy dissipation associated with microstructure defects, such as grain boundaries and impurities; thermoelastic effect caused by local temperature gradiensts resulting from nonuniform stresses. No single model can satisfactorily represent the internal damping characteristics of all materials. Nevertheless, two geneal types of internal damping can be identified: viscoelatic damping and hysteretic damping.

For a linear viscoelastic material, the stress-strain relationship is given by a linear differential equation with respect to time, having constant coefficients. The Kelvin-Voigt model is often used for modelling viscoelastic material:

$$\sigma = E'\varepsilon + E''\frac{d\varepsilon}{dt}$$

$E'$ is the Young's modulus

$E''$ is the complex modulus representing the viscoelatic material damping properties that is assumed to be time independent

The frequency-dependent complex modulus model is an approach which allows the complex modulus to vary as a function of the excitation frequency

# Experimental characterization of viscoelastic material

The complex dynamic modulus of the material is defined as:

$$E^* = E' + jE''$$

$$E' = \frac{\sigma}{\varepsilon} = k\frac{h}{A}$$

$$E'' = c\omega\frac{h}{A}$$

Loss factor which represents the damping capity of the material

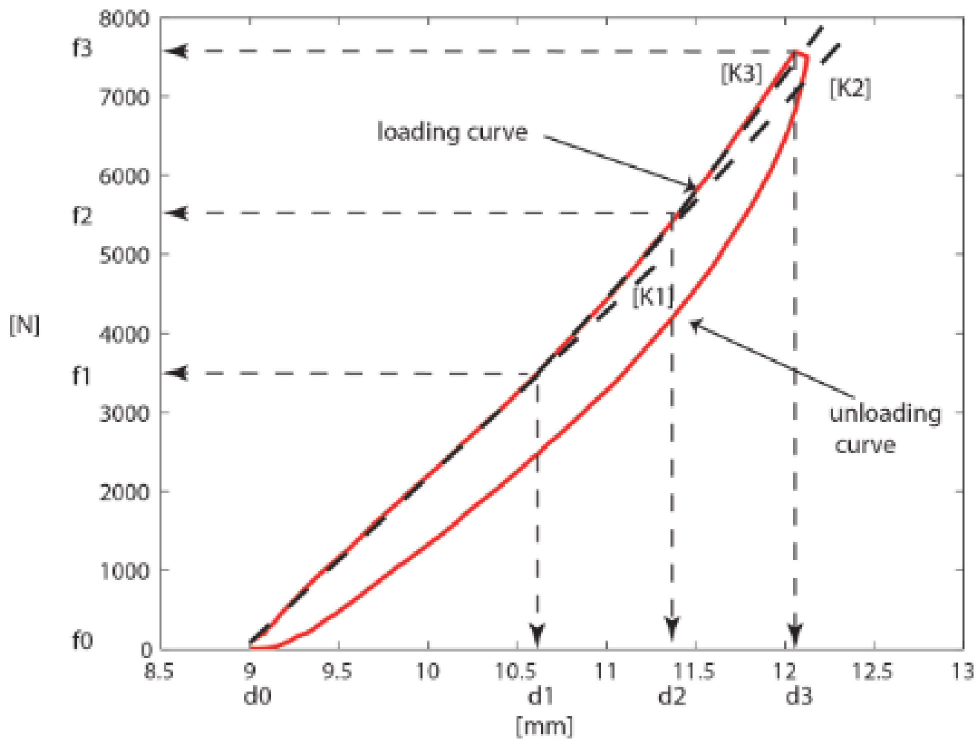
$$\eta(\omega) = \frac{E''(\omega)}{E'(\omega)}$$

Generally, the experimental accelerometer signal is used as the response. As a consequence the complex stiffness and the complex modulus are obtained through the frequency response function as

$$E'(\omega) + jE''(\omega) = \frac{h\omega^2}{A}\left(\frac{1}{\frac{\ddot{x}(\omega)}{F(\omega)}}\right)$$

# Experimental characterization of viscoelastic material – Example

Test on polyurethane material for heavy-duty wheel




Cylindric sample 9mm height and 20mm diameter.

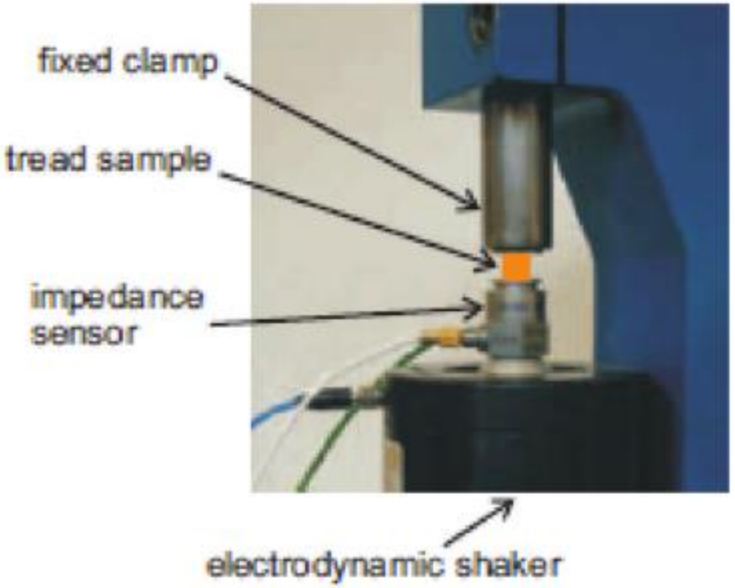
Stage 1. Static analysis.

| $n$ | $\delta n$ (mm) | $\delta f$ (N) | $K$ (N/mm) | $E$ (MPa) |
|---|---|---|---|---|
| 1 | 1.6 | 3500 | 2187 | 8.57 |
| 2 | 0.75 | 2000 | 2631 | 10.31 |
| 3 | 0.69 | 2000 | 2898 | 11.36 |

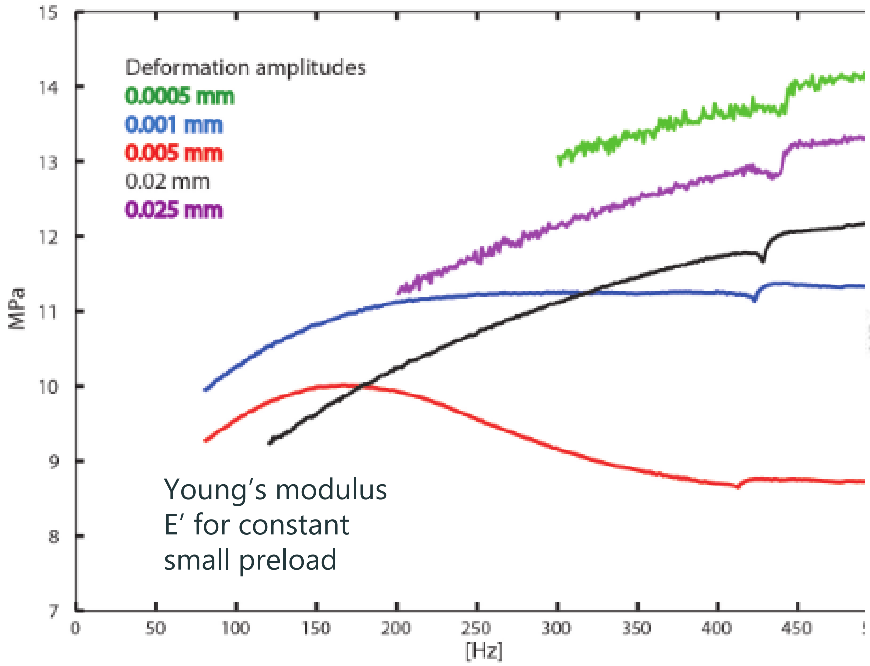
Different static stiffness values are obtained as a function of the applied load.

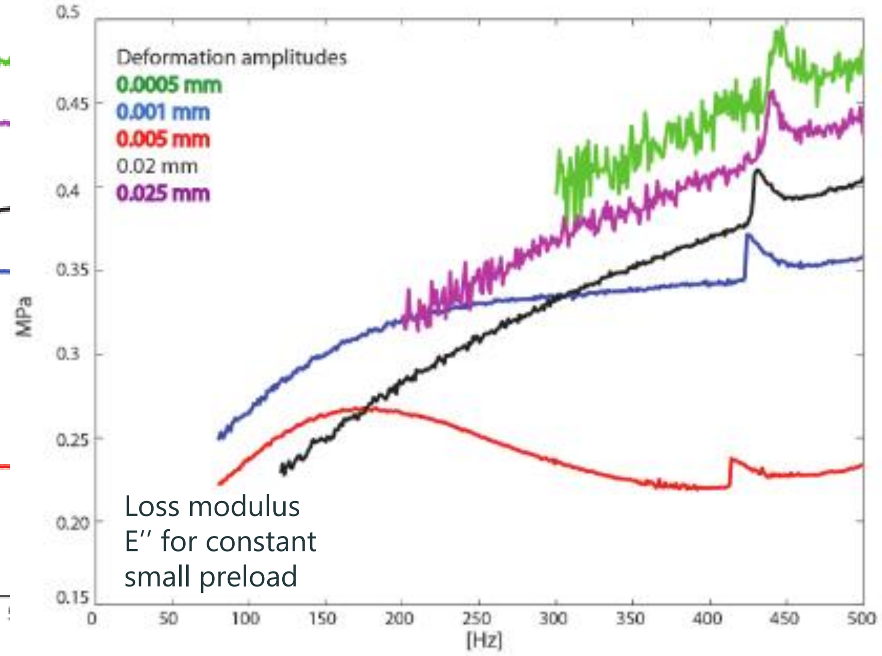
# Experimental characterization of viscoelastic material – Example

## Test on polyurethane material for heavy-duty wheel




The measure curves do not cover the same frequency range. The upper frequency for each excitation level is constrained due to the limited power of the shaker. The higher the vibration amplitude, the more power is needed at the same frequency. The lower frequency limit is related to the sensitivity of the impedance sensor. There is a lower limit for the acceleration and force measuurement below which the signal to noise ratio becomes too low.



- The catalogues propose a static procedure for the choice of the isolator, based on the “static deflection”, “static stiffness”, shore.
- This is almost never sufficient because:
  - The real system could almost never be represented as a 1 DOF system (the transmissibility function presents many resonances after the first one)
  - Stiffness depends on the frequency (dynamic stiffness)
  - Stiffness depends on the pre-load (static deflection)
  - Stiffness depends on the dynamic load (peak-peak)
- The isolator choice using the static procedure is a necessary but not a sufficient condition.
- Experimental dynamic tests and related checks are needed.

# MOUNT characterization (1)

- Exp
  - In-situ
    - In firing - measure of the upstream and downstream accelerations in steady state and run-up operational conditions
      - Notes: ratio between autopowers and mount efficiency in a particular condition (engine rpm, cabin weight ). Comparison between mount type.
    - FRF measurements
      - By hammer and accelerometers
        - Notes: Dynamic stiffness; real static mount deflection; no variable displacement; problem with possible non linearities; verify reciprocity
      - By shaker (free), force sensor and accelerometers
        - Notes: Dynamic stiffness; real static mount deflection; excitation with variable acceleration ; small variable displacement; problem with possible non linearities
  - Lab Test
    - By shaker + force sensor and accelerometer in close loop (sweep control, random control)
      - On a specimen (e.g. cylindrical)
        - Notes: Dynamic stiffness and E (freq, Xst, Xdynam) (by a model). Effect of different static deflection, dynamic displacement and acceleration
      - On the real mount
        - Notes: Dynamic stiffness; application of the real static mount deflection; excitation with the real variable acceleration and real variable displacement; analysis of the possible non linearities (by test with different variable displacements or accelerations or static deflections). Test with the real acceleration and displacement profile are obtained by accelerometer and laser measurement in firing

# MOUNT characterization (2)

Numerical → Development of an FE model of the mount with BCs and applied load equal to the Lab test (on the basis of the Lab test, selection of the best material model, E(f)) → Model Validation : static validation, dynamical validation (by loading the model with the same acceleration/displacement profile as in the Lab test). Model validation should be verified in a certain range of Xst, Xdynam, frequency. Attention should be paid for modal analysis (the stiffness matrix is taken as constant)

↓

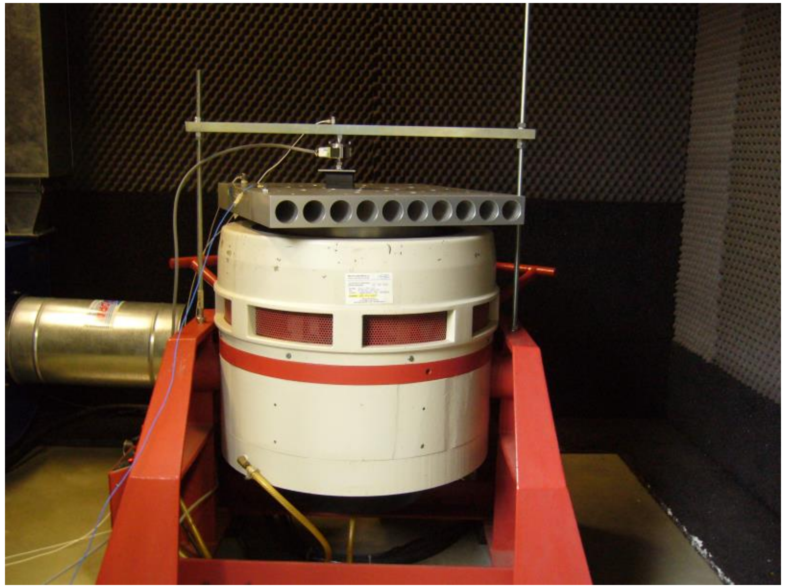
Geometry or stiffness modification in order to improve performance

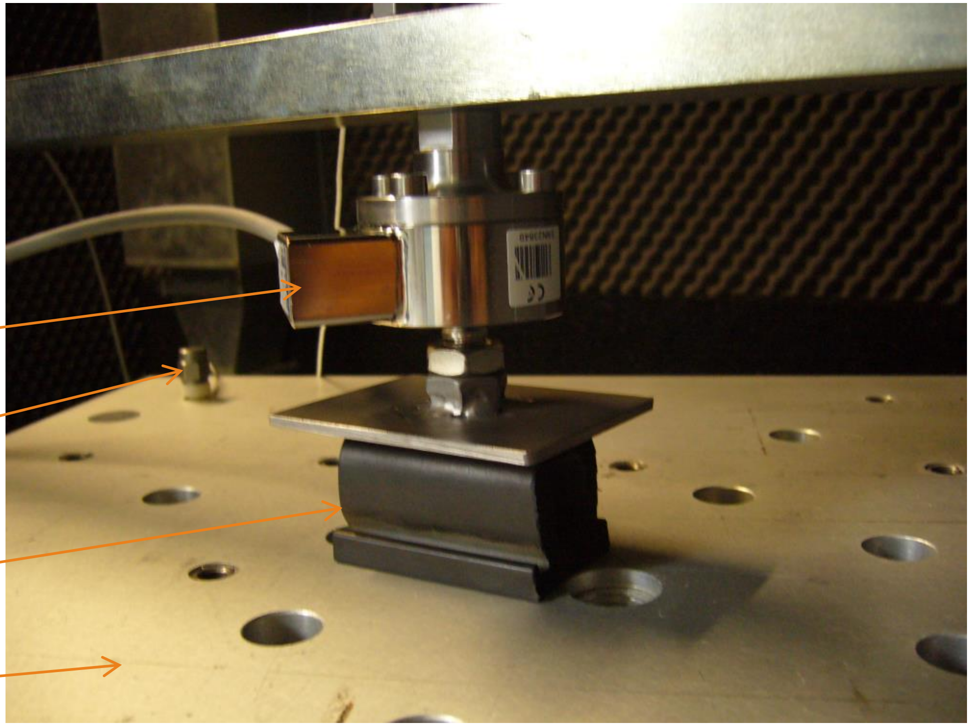
# Case study 1 - MOUNT characterization – Direct Method

Model for E determination (correction for BCs)

$$E'(\omega) + jE''(\omega) = \frac{L\omega^2}{A}\left(\frac{1}{\ddot{x}/F}\right)$$

Setup for DS measurements in Lab test (Direct Method, i.e. direct measurement of acceleration and Force)





Force sensor

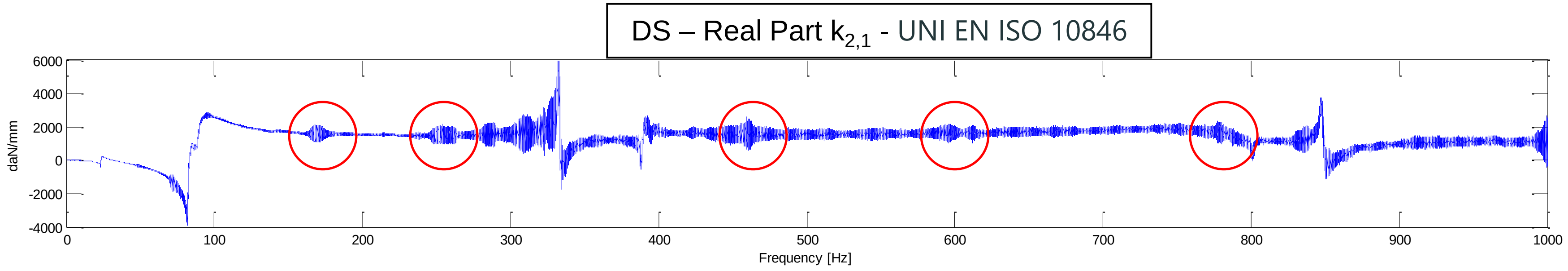
Control accelerometer

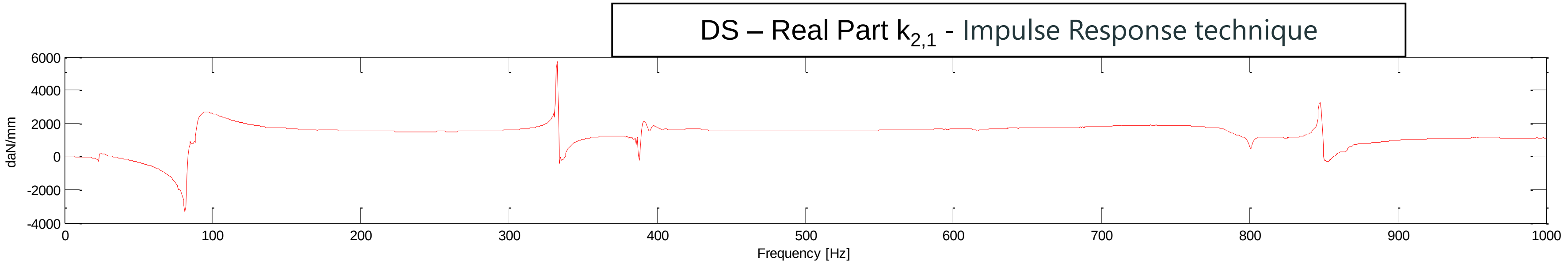
specimen

Shaker fixture

# Case study 1 – Dynamic Stiffness Lab measurements

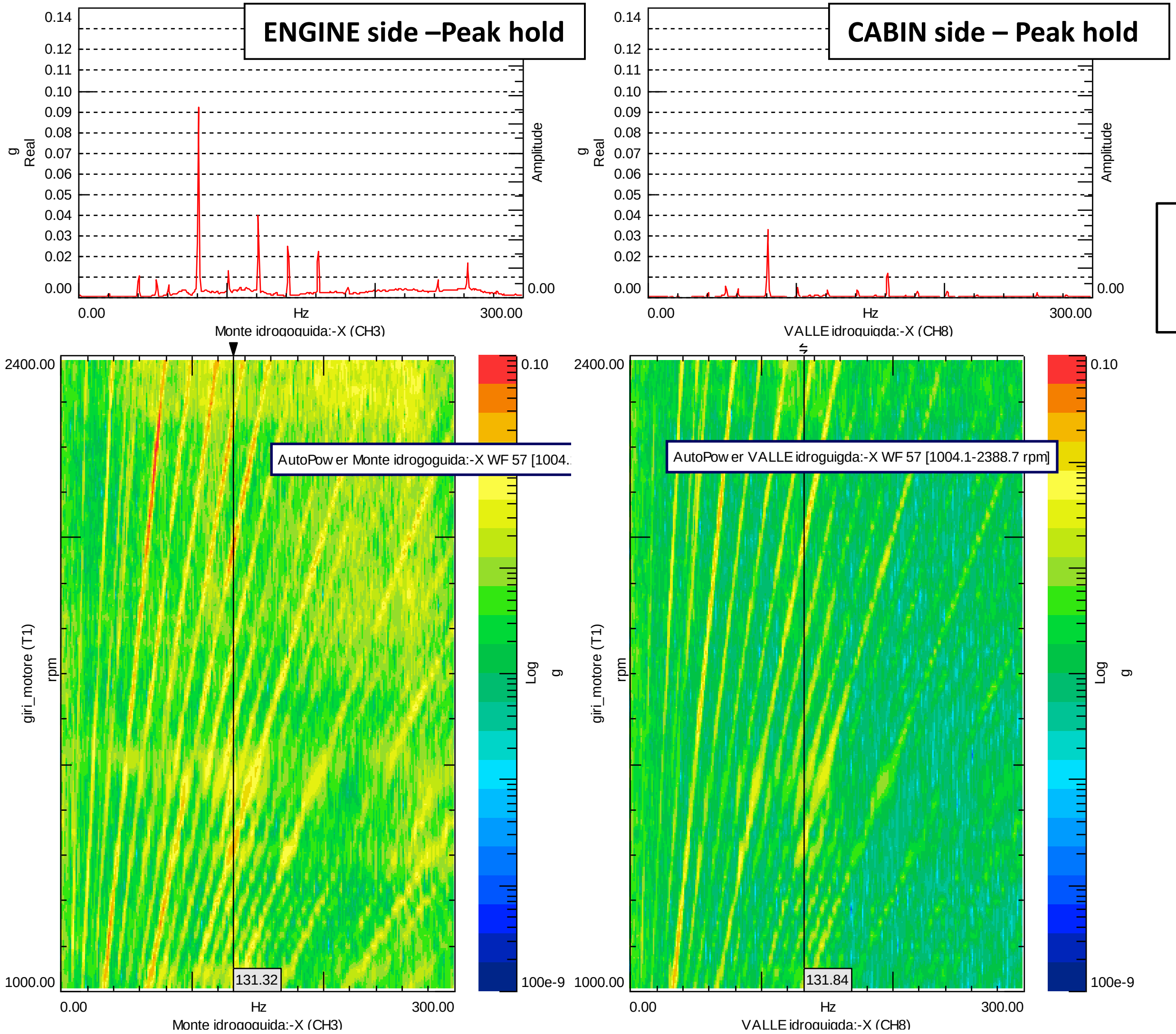
- Difference between 2 type of algorithm for DS calculation
- Attention to system resonances (due to the supporting structure)



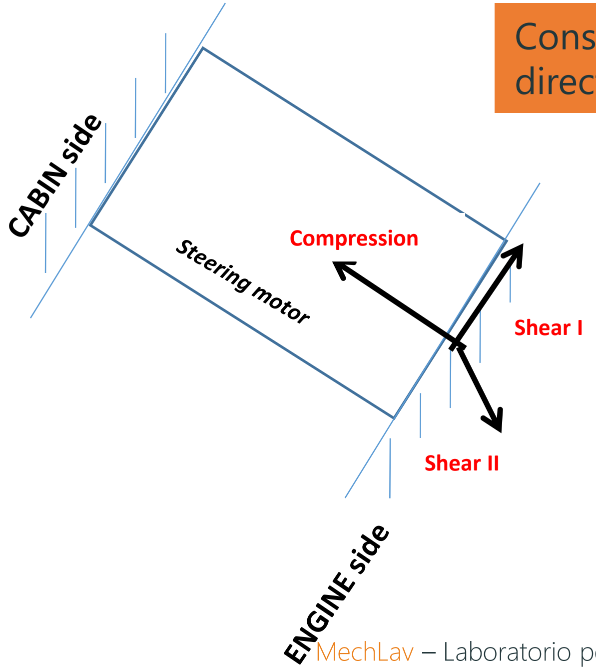
# Case study 2 –MOUNT characterization

**Transmissibility – COMPRESSION direction**



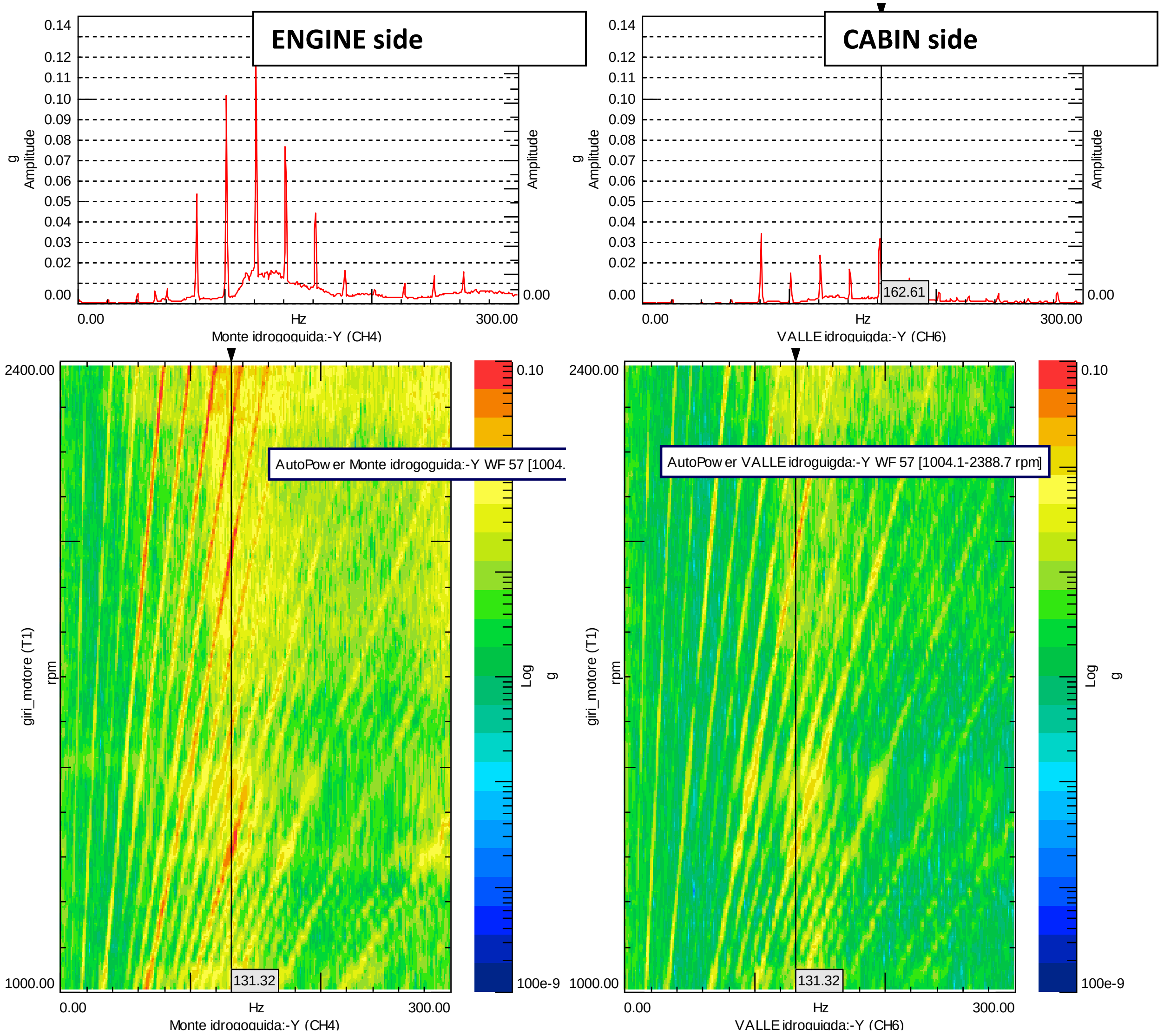

- ↓ 10 dB from engine to cabin side.

Consider all the directions



# Case study 2 – MOUNT characterization

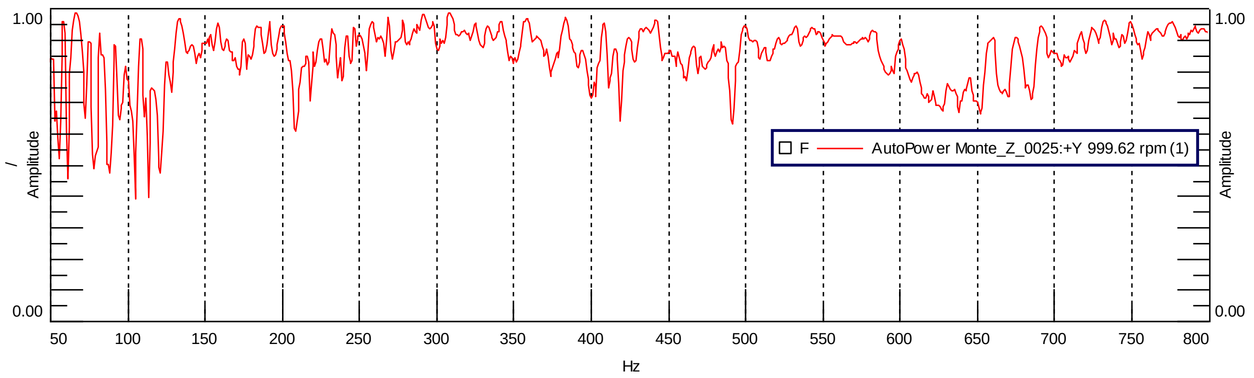
**Transmissibility – SHEAR direction I**



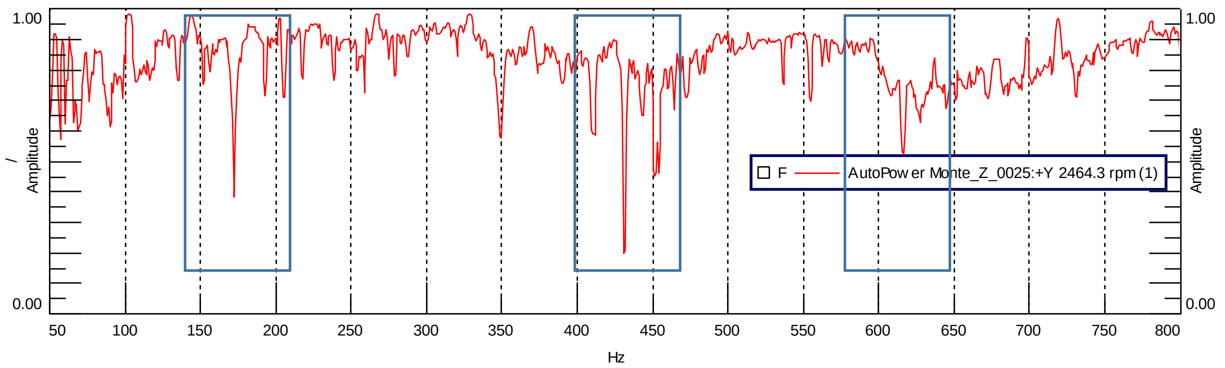

- ↓ 15 dB from engine to cabin side;
- RESONANCE firewall/cabin at 131 Hz.

# Case study 2 –MOUNT characterization

**Transmissibility – in firing - Different speeds**

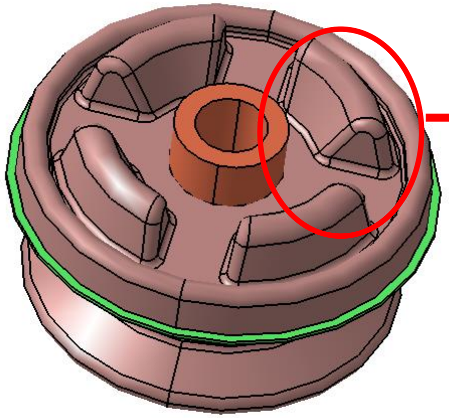


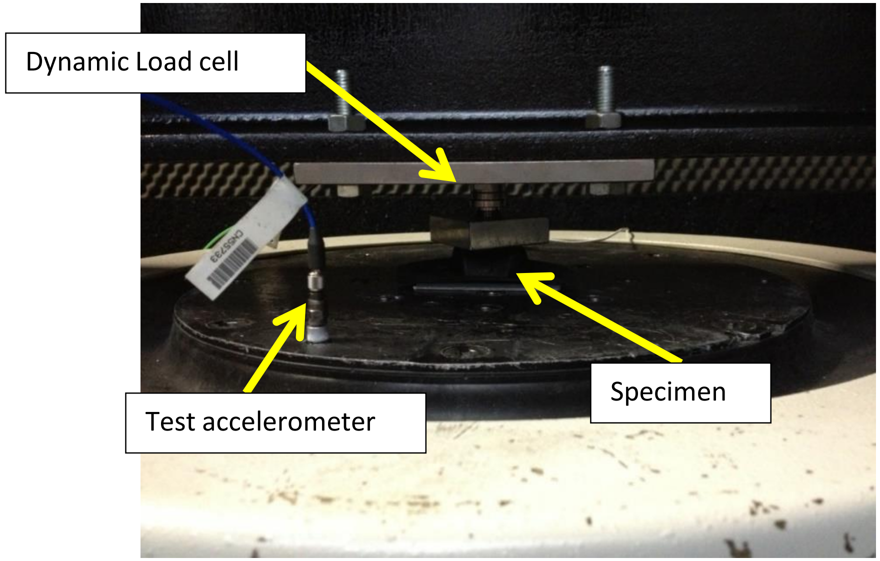

1000 rpm



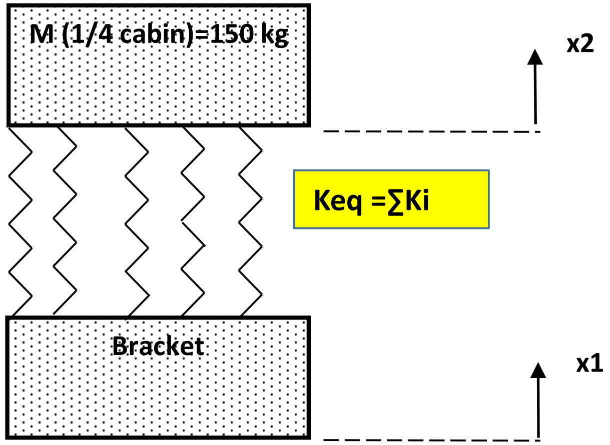

2400 rpm – Best behaviour at high speed

# Case study 3 – silent block

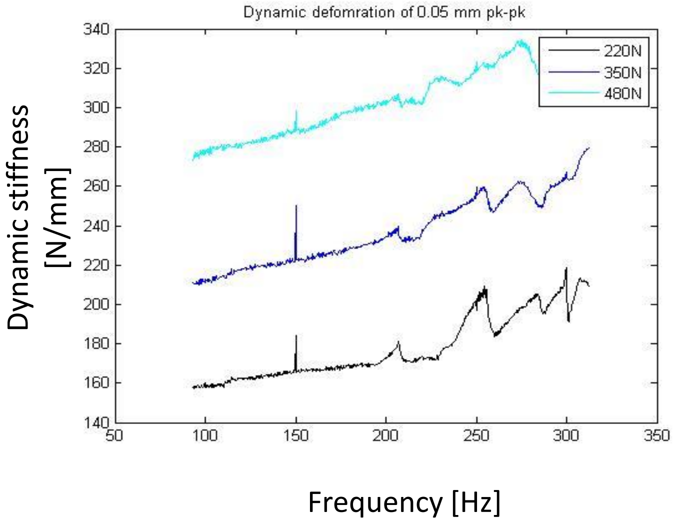


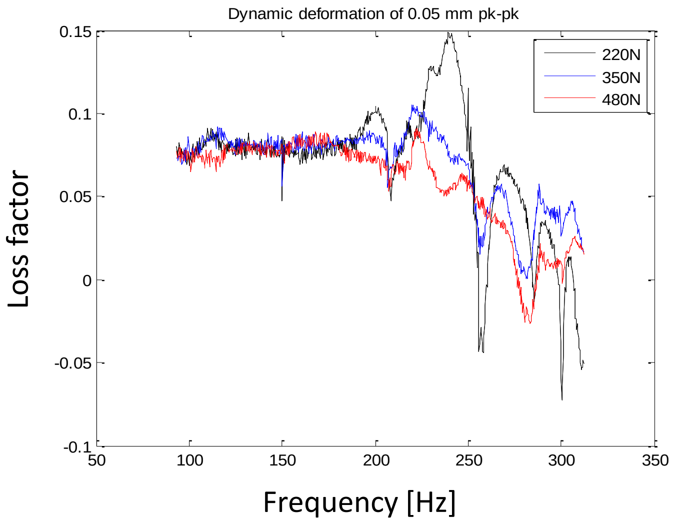


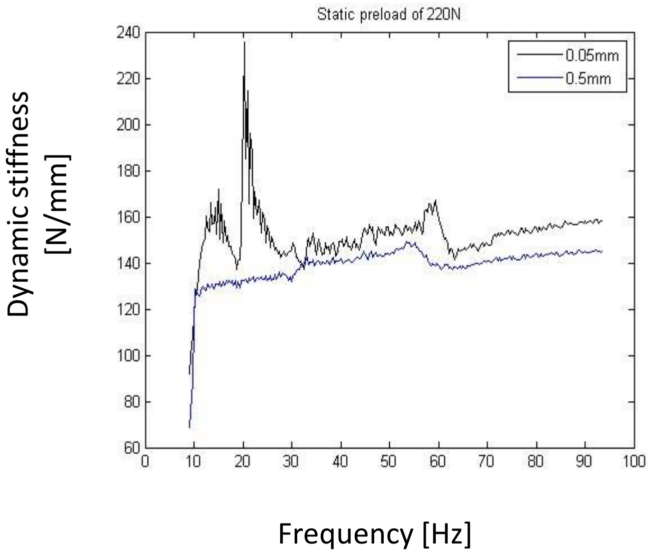

-Tested preloads: 220N, 350N, 480N;
- Tested dynamic displacements: 0.5 mm, 0.05 mm;
- Operational preload:
6000N / 4(mounts)= 1500N (load for each mount)
1500 N /5 =300N (load for each equivalent spring)

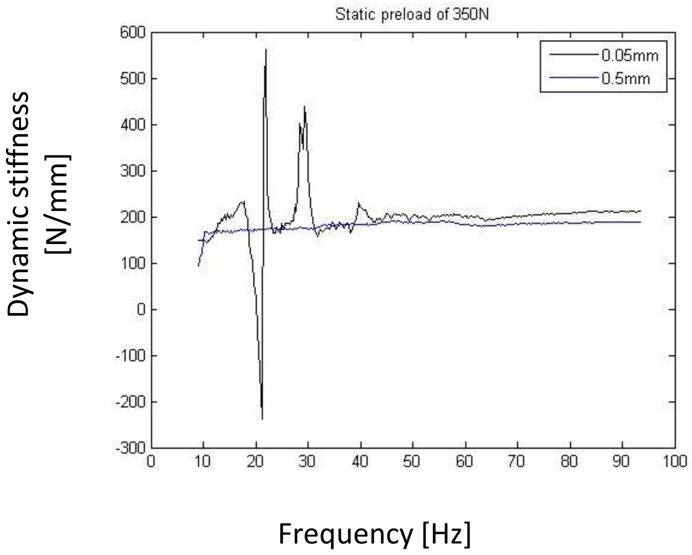


# Case study 3 – Vibration absorber

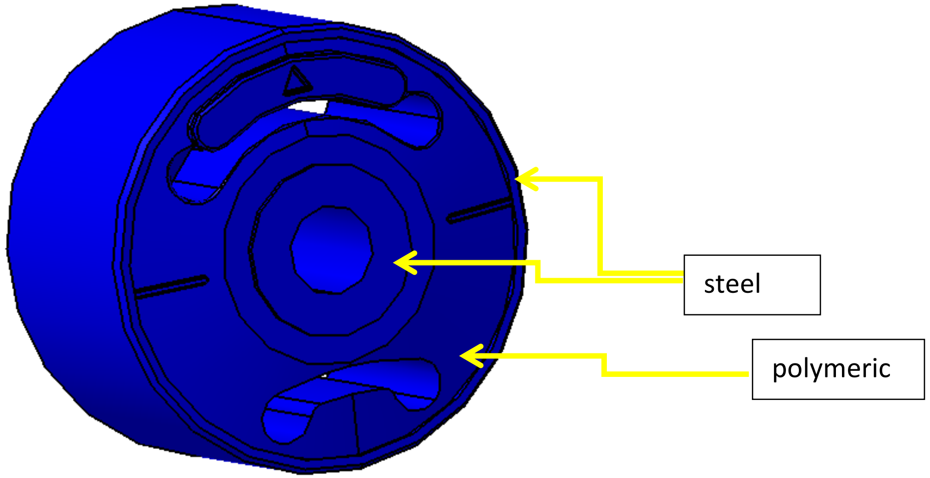


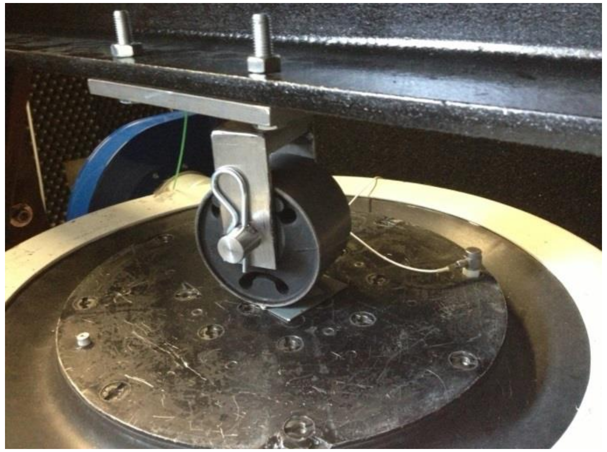



# Case study 4 – silent block

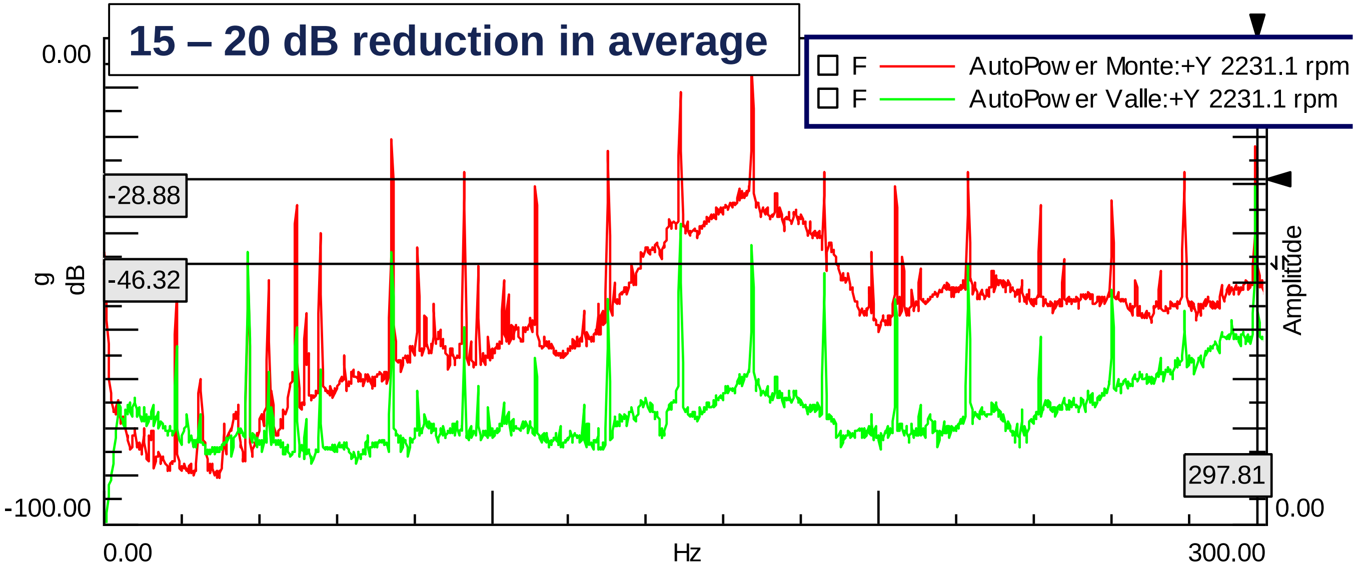


- Tested preloads: 250N, 500N;
- Tested dynamic displacements: 0.5 mm, 0.05 mm, 1mm;
- Operational preload: 1800N.

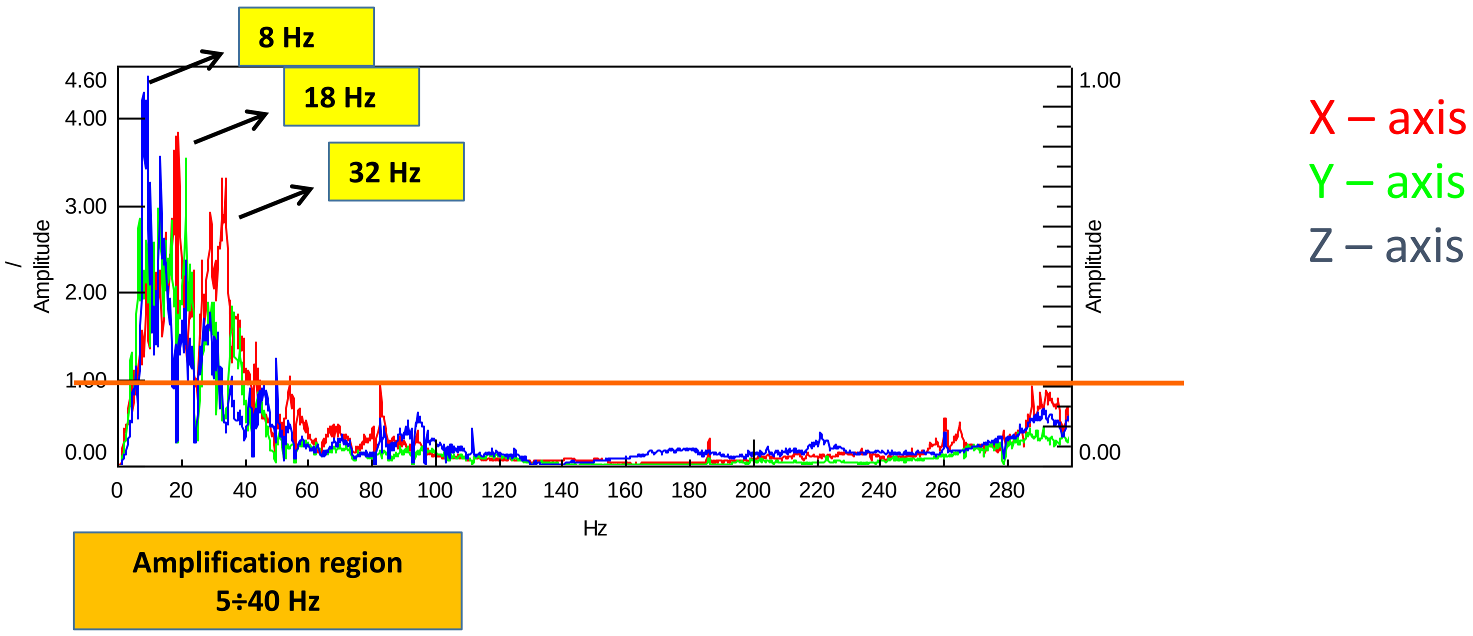
# Case study 4 – silent block for tractor cabin

## Operational measurements – In-firing tests
## Upstream/Downstream @2400 rpm



# Case study 4 – silent block for tractor cabin

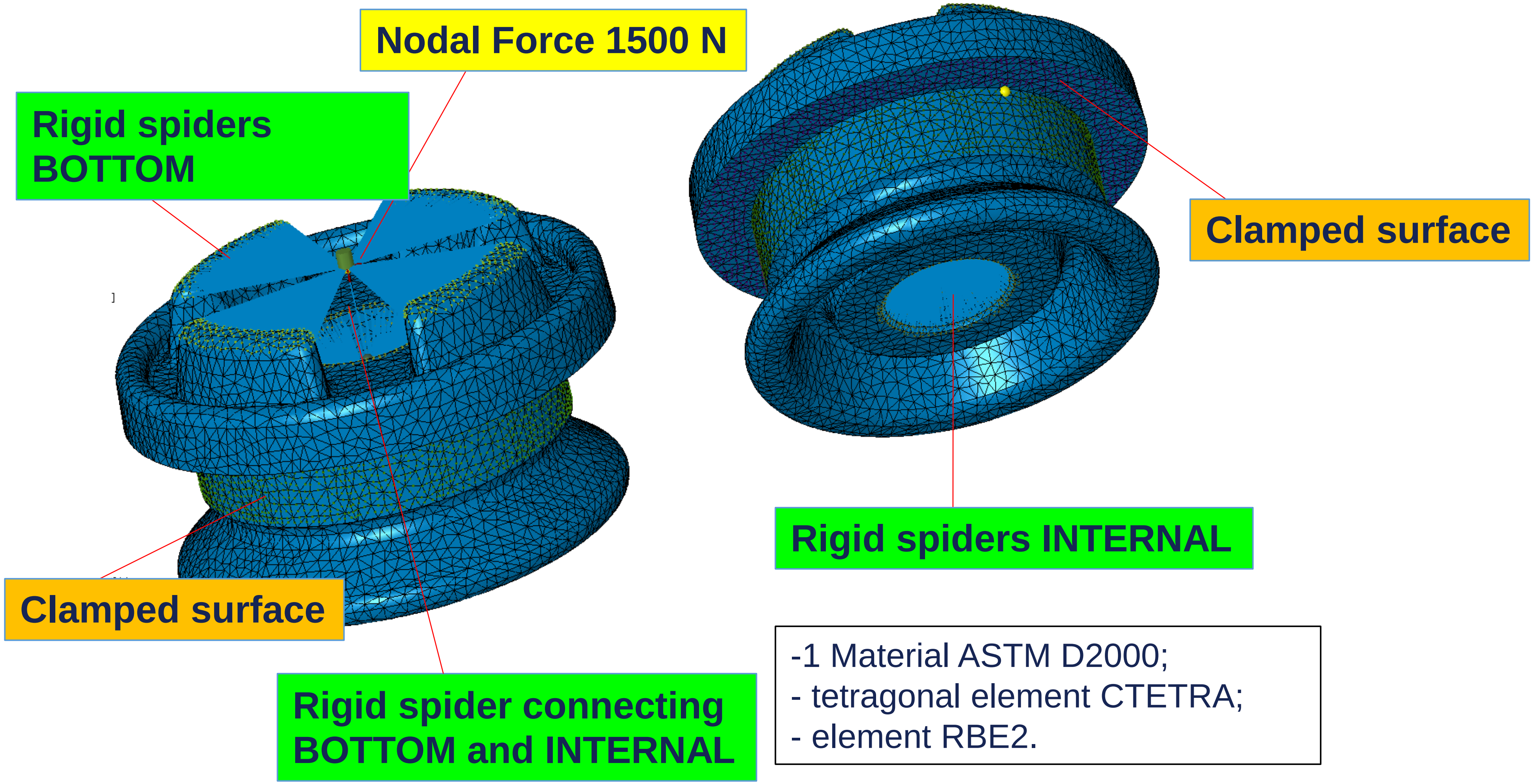
Trasmissibility




The amplification region is due to the low frequency modes in the range 0-30Hz.
In particular, it depends on the mounts stiffness (X-Y-Z direction) and mass and inertia moments.
The amplification region is NOT due to an unsatisfactory behaviour of the mounts, but to the low frequency modes which occur in this region.

# Case study 5 – silent block

FE model  Nastran STATIC Case




- 1 Material ASTM D2000;
- tetragonal element CTETRA;
- element RBE2.

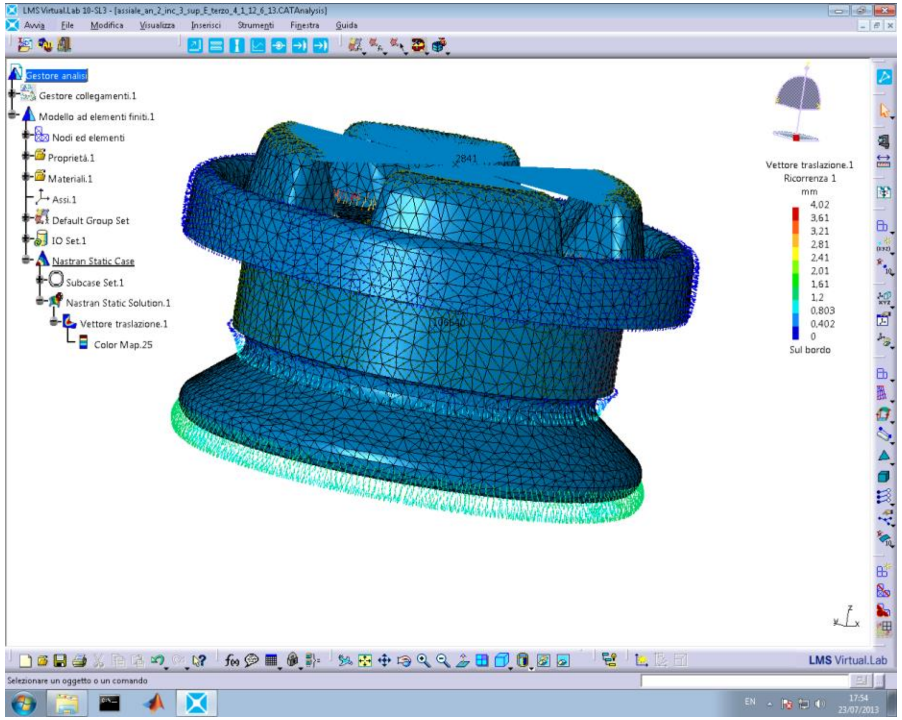
# Case study 5 – silent block

Starting from the E modulus of the literature (E=2,8N/mm2), I try to obtain a static stiffness of 375N/mm (experimentally obtained) performing static analysis on the real component

FE model Nastran STATIC

Case – finding E Modulus

-1st trial: E1=2,8N/mm2 → static analysis →deformation of 3,44 mm → $K_0$=436N/mm
-2nd trial: E2=2,5N/mm2 → static analysis → deformation of 3,86 mm → $K_0$=388N/mm
**-3rd trial: E3=2,4N/mm2 → static analysis →deformation of 4,02 mm → $K_0$=373N/mm**




This methodology is simpler and more precise, since in this case I have the static stiffness of the component.

# Case study 7 – Engine test-rig isolators

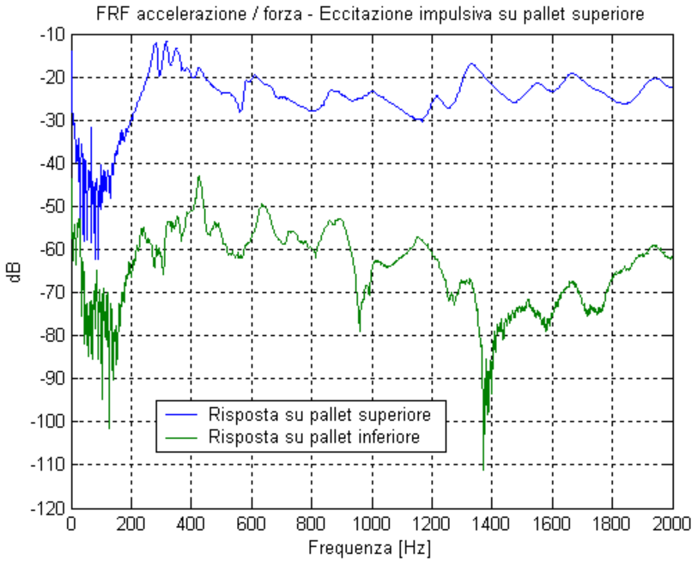
Vibrational experimental measurements on diesel engine test-rig

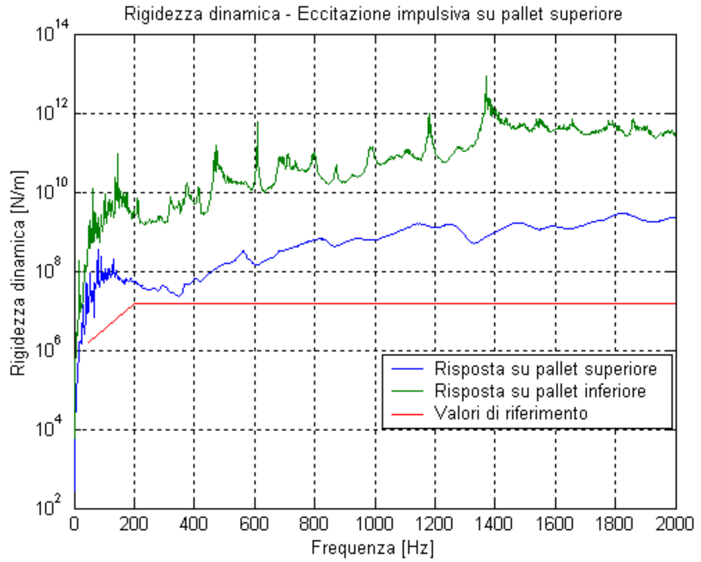
1. Determination of the dynamic stiffness (force-displacement transfer function) of the engine anchor points on the test-rig pallet, through measurements using instrumented impact hummer impulsive excitation  and evaluation of the acceleration response.

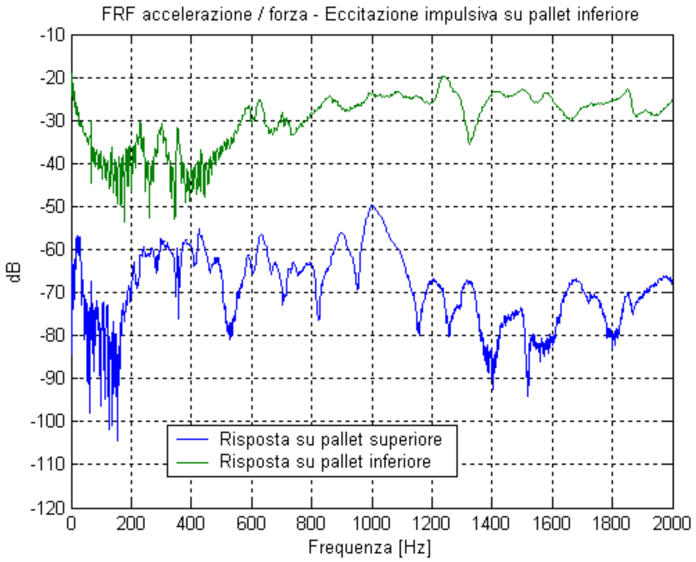
2. Determination of the filtering levels induced by the isolators (ratio of the acceleration auto spectrum measured on the upper pallet and lower pallet), in five different steady state engine operating conditions.

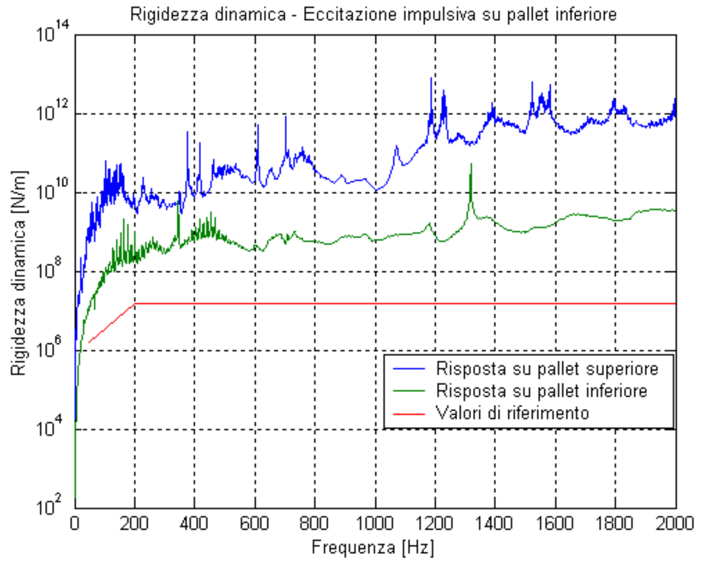
3. Determination of the filtering levels induced by the isolators (ratio of the acceleration auto spectrum measured on the upper pallet and lower pallet), in run-up engine operating conditions.

# Case study 7 – Engine test-rig isolators

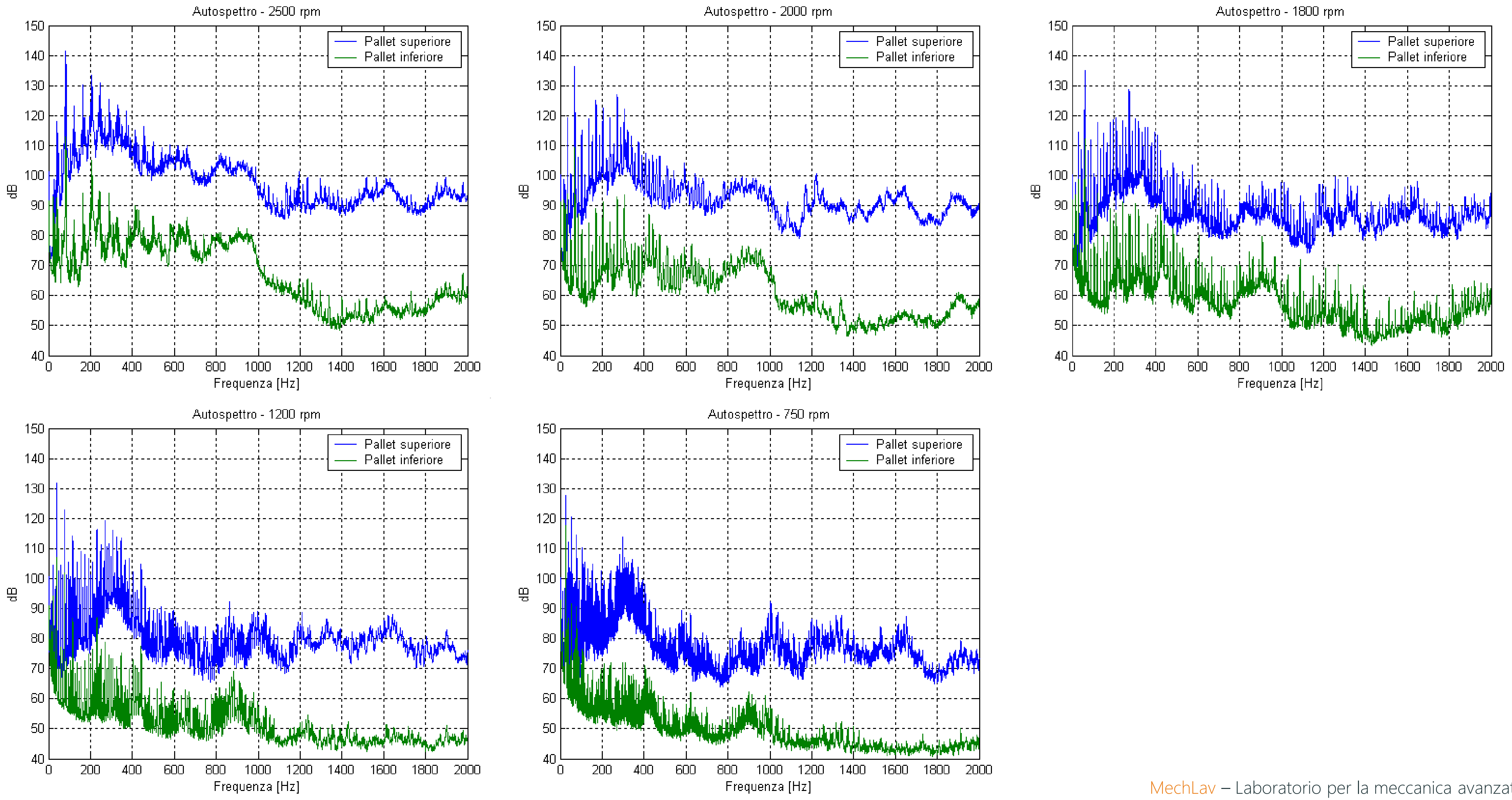


# Case study 7 – Engine test-rig isolators



# Case study 7 – Engine test-rig isolators

In order to obtain a global characterization of the equipment, the auto-spectra related to the points of the upper pallet ($G_1$, $G_3$) and of the lower pallet ($G_2$, $G_4$) have been respectively averaged. $L_s$ are the levels related to the upper pallet, while $L_i$ are the levels related to the lower pallet.

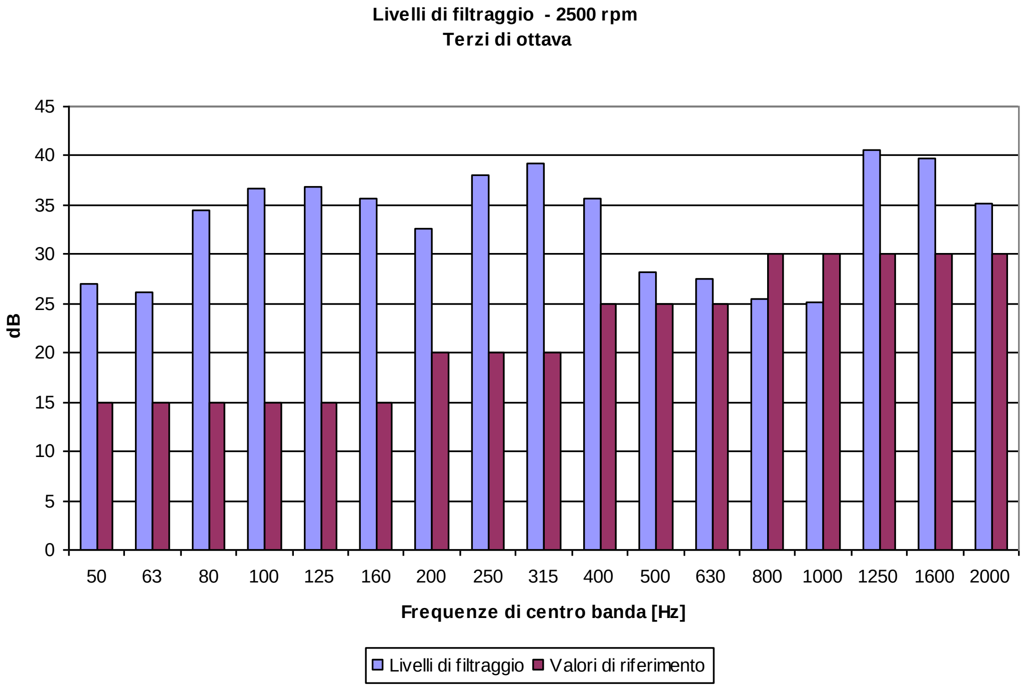
$$L_S = 10 \cdot \log_{10} \frac{\frac{1}{2}\left(\sum_{banda} G_1 + \sum_{banda} G_3\right)}{10^{-12}\left[\frac{m^2}{s^4}\right]}$$

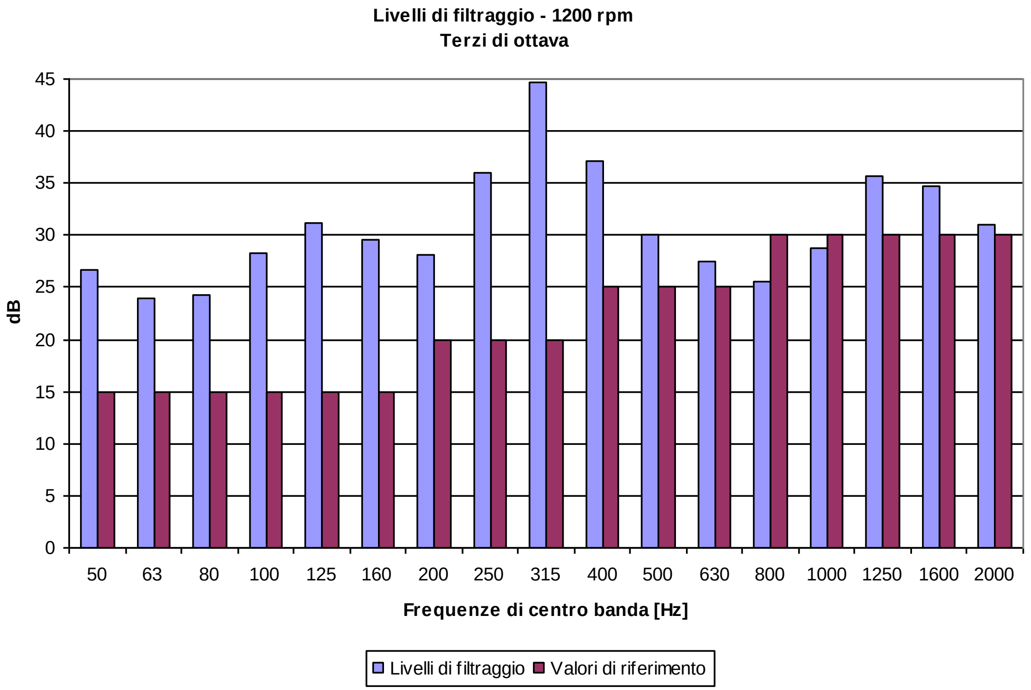
$$L_I = 10 \cdot \log_{10} \frac{\frac{1}{2}\left(\sum_{banda} G_2 + \sum_{banda} G_4\right)}{10^{-12}\left[\frac{m^2}{s^4}\right]}$$

One-third octave filtering levels:

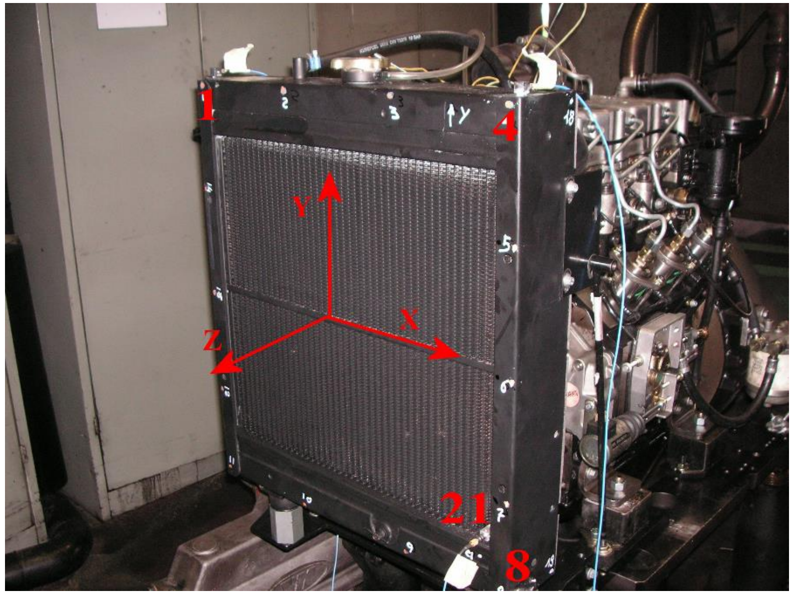
$$L_F = 10 \times \log_{10}\left[\frac{1}{2}\left(\frac{1}{N_{CB}}\sum_{banda}\frac{G_1}{G_2} + \frac{1}{N_{CB}}\sum_{banda}\frac{G_3}{G_4}\right)\right]$$

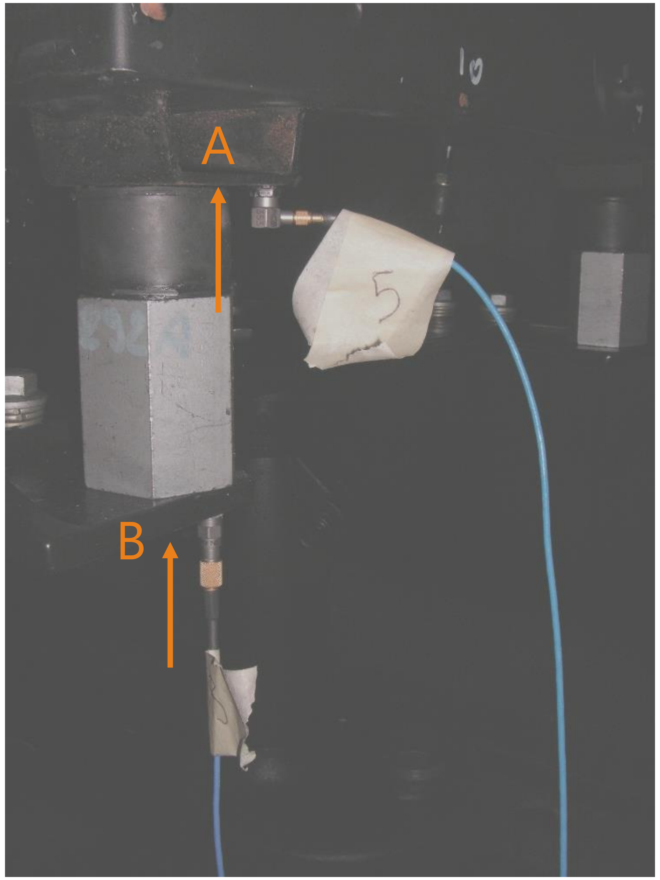
Where, the summations are performed on the components contained in each one-third octave band and $N_{CB}$ is the number of such components. The filtering levels show the vibration energy ratio (in terms of acceleration) on the upper and lower pallet, in dB.



# Case study 8 – Radiator support isolators



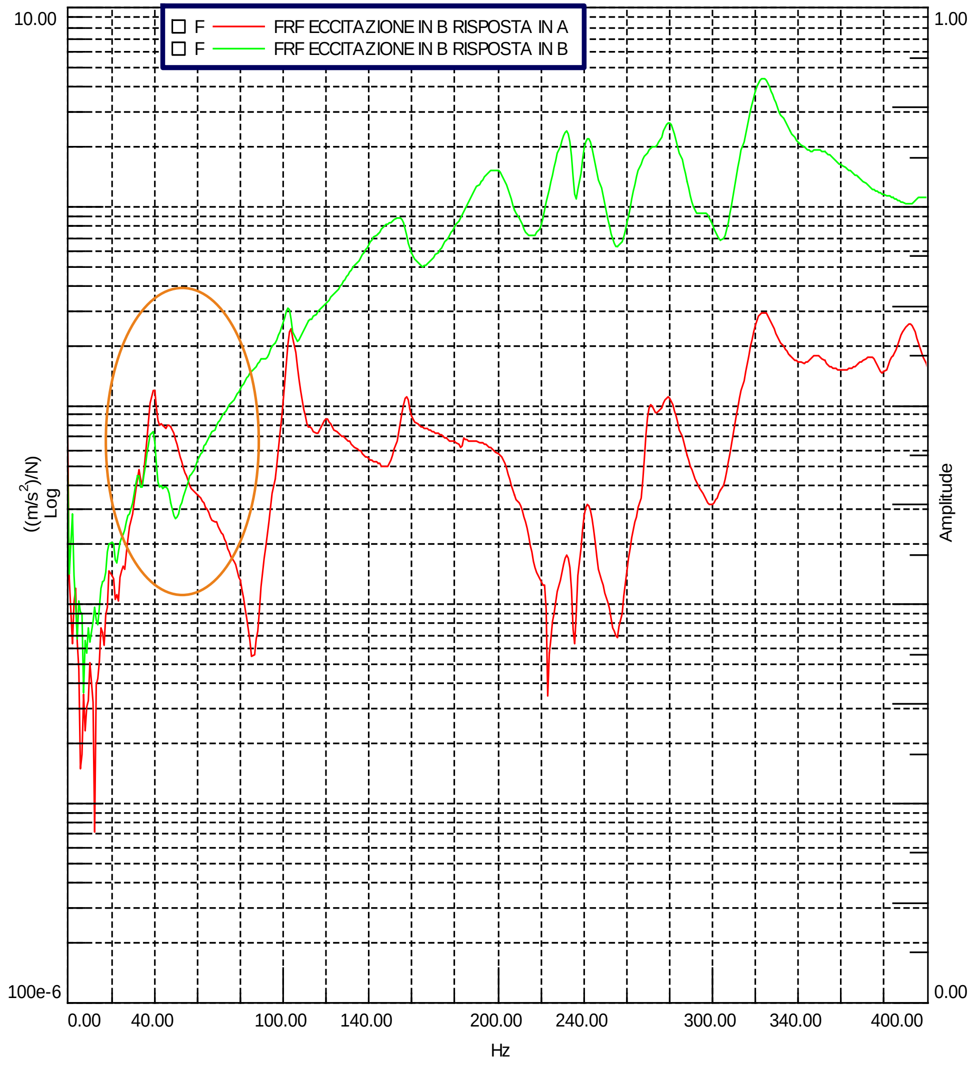

1. Determination of the dynamic stiffness (force-displacement transfer function), through measurements using instrumented impact hummer impulsive excitation  and evaluation of the acceleration response.

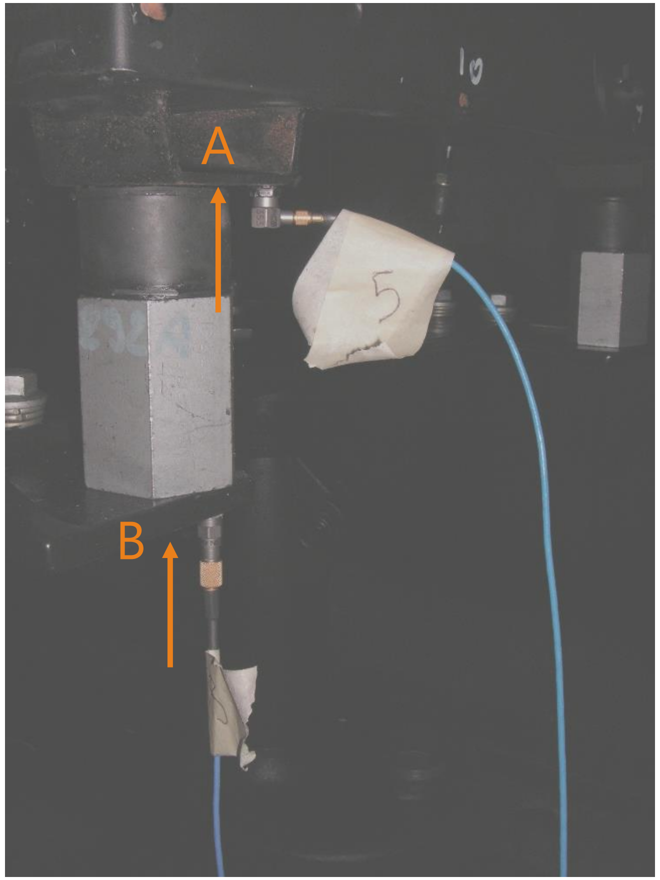
2. Determination of the filtering levels induced by the isolators in operating conditions.

A radiator side
B engine side

# Case study 8 – Radiator support isolators

Excitation in B (engine side)



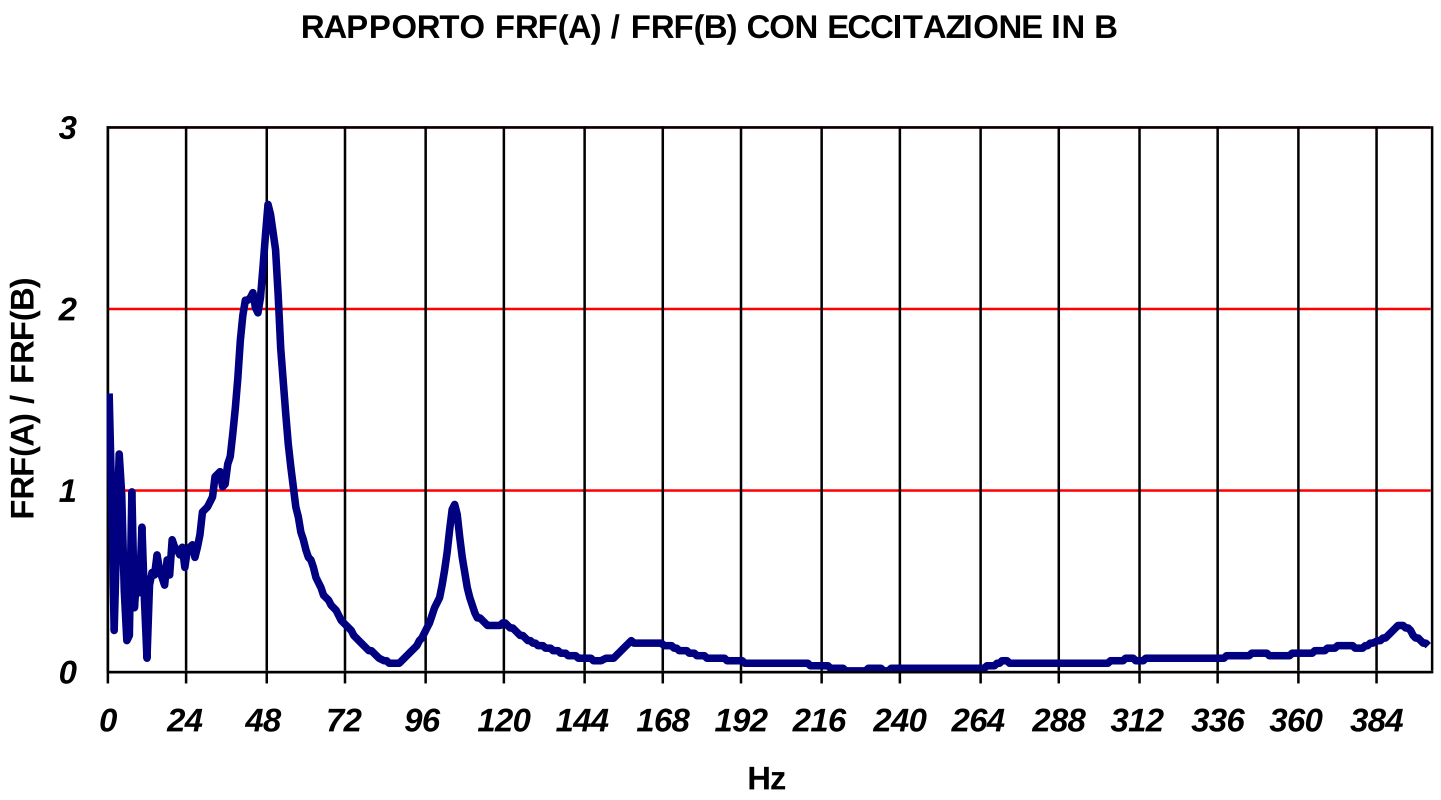




The filtering level resulting from an impulsive excitation in the point B is poor for frequency values lower than 120 Hz. Particularly, it can be observed that, in the 30-60 Hz range, the FRF measured in the point A is higher than the FRF measured in the point B. This means that in this particular case the vibration is not damped but it is amplified.

The system composed by the nut and the isolator has a resonance at 40 Hz and at around 100 Hz, where both the FRFs show pretty sharp peaks.

# Case study 8 – Radiator support isolators

Excitation in B (engine side)




From the FRFs ratio in the 0-400 Hz range, it is possible to notice that the isolator is not effective for in the range between 32 Hz and 56 Hz, because the ratio is higher than 1. The initial part of the plot, from 0 Hz to 20 Hz, shows that the **efficiency index** has an irregular trend. This is due to the fact that the consistency between response and excitation is lower than 0.75 until 20 Hz, which means that the results are not reliable in this frequency range.