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Polymer damper technology for improved system dynamics

Theo A.M. Ruijl[1], P. Wullms[1]

[1]*MI-Partners, Eindhoven, The Netherlands*

t.ruijl@mi-partners.nl

Abstract

Visco-elastic materials are applied in various applications to achieve damping, however in many of these applications the precise performance is less critical and often the design is realized by trial and error. However, in many high-end applications the complexity and requirements are such that high predictability of the final system performance is indispensable during the development and therefore, design by trial and error is not applicable.

The presented work shows that polymer damping can be applied to these high-end applications to increase the dynamic performance and robustness of the system successfully with sufficiently high predictability. A polymer with high damping and reproducibility is developed and a design methodology has been established, including accurate dynamic material characterization, dynamic system modelling and FE-models of the polymer, resulting in a systematic approach and predictable design.

Polymer Damping, Visco-elastic damping, Hysteretic Damping, Passive Damping of Positioning Stages

1. Introduction

Requirements on the dynamic position stability of high-end mechatronic systems becomes increasingly tighter. This applies to moving systems, like stages, as well as to static systems, like force and metrology frames. A very effective and robust way to improve the system dynamics is by applying damping. This can either be done in an active or in a passive manner. In several applications, passive damping is more preferred than active. This mainly for reasons of robustness, applicability, performance and cost. Visco-elastic damping, a method to achieve passive damping, can be obtained very efficiently by means of polymers. The amount of damping per volume of material is high [1,2], the material is robust and can be shaped efficiently which gives design freedom to tailor damping elements for specific applications [3,4].

Visco-elastic materials are applied in various applications to achieve damping, however in many of these applications the precise performance is less critical and often the design has been realized by trial and error. In our high-end applications, the complexity, requirements and conditions are such that extensive modelling is necessary and predictability of the final performance is essential for an efficient and successful development of the entire system.

This paper describes the steps towards a predictable and efficient implementation of visco-elastic damping to improve the system dynamics in a high-end positioning stage. Starting with the theoretical background on visco-elasticity, followed by material characterization and the design methodology. The actual damper design and the achieved dynamic improvements are shown in the last chapter. The development of a polymer with very high damping and reproducibility in the frequency and temperature range as common in high-end mechatronics systems was part of the entire development as well. The development was done together with an external company. The development of the polymer itself and optimization of the preparation and vulcanization process is out of the scope of this paper.

2. Visco-elastic Behavior

For visco-elastic materials, which exhibit damping, stress and strain are not in phase. The strain is lagging behind the stress by a certain phase angle δ. For dynamic modelling in the frequency domain, it is convenient to describe the linear visco-elastic behavior in complex notation. Considering the complex time dependent strain as

$$\varepsilon^*(t) = \varepsilon_0 e^{j\omega t} \qquad (1)$$

Where ε_0 depicts the amplitude of the sinusoidal variation of the strain and ω the frequency. A linear visco-elastic response will give the corresponding stress

$$\sigma^*(t) = \sigma_0 e^{j(\omega t+\delta)} \qquad (2)$$

It is common in the polymer technology domain, to describe the linear visco-elastic relation between stress and strain by means of the complex modulus as

$$E^* = \frac{\sigma^*}{\varepsilon^*} \qquad (3)$$

Substitution of (1) and (2) onto (3) and applying Euler's exponential relationship we get

$$E^* = \frac{\sigma_0}{\varepsilon_0} e^{j\delta} = \frac{\sigma_0}{\varepsilon_0}(\cos(\delta) + jsin(\delta)) \qquad (4)$$

In general, the ratio between σ_0 and ε_0 as well as the phase angle δ are frequency and temperature dependent [1]. As such we can define two, in general frequency and temperature dependent functions, the storage-modulus E' and the loss-modulus E'' and rewrite (4) into

$$E^* = \frac{\sigma_0}{\varepsilon_0}(\cos(\delta) + jsin(\delta)) = E' + jE'' \qquad (5)$$

The ratio between the loss-modulus and the storage-modulus is called loss-factor η or tan (δ)

$$\eta \equiv \frac{E''}{E'} = \tan(\delta) \qquad (6)$$

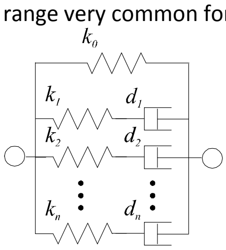
And as such the complex modulus (5) is also given by

$$E^* = E'(1 + j\eta) \tag{7}$$

A polymer typically shows an enormous change in stiffness (e.g. $|E^*|$) at the so-called glass transition temperature T_g. The loss-factor η is also maximum around this glass transition.

A useful and common way to describe the visco-elastic stiffness of polymers, required to model the effect of damping on a system, is by means of combinations of elastic elements (ideal elastic spring) and viscous elements (ideal viscous damper). Figure 1 shows a so-called Multiple Standard Model or Maxwell model.

This model can be used to describe the modulus in a general sense, i.e. Young's modulus E* and shear modulus G*, or the dynamic stiffness k^*. A fit of the multiple standard model of the polymer developed by MI-Partners is shown in figure 2. It shows the high damping (loss-factor of max 1.7) in a frequency range very common for mechanical systems.



$$k^* = k' + jk'' = k_0 + \sum_{i=1}^{n} \frac{jk_i d_i \omega}{jd_i\omega + k_i} = \left(k_0 + \sum_{i=1}^{n} \frac{k_i d_i^2 \omega^2}{d_i^2\omega^2 + k_i^2} \right) + \left(\sum_{i=1}^{n} \frac{k_i^2 d_i \omega}{d_i^2\omega^2 + k_i^2} \right) j$$

Figure 1. Visco-elastic stiffness k* by means of the Multiple Standard Model.

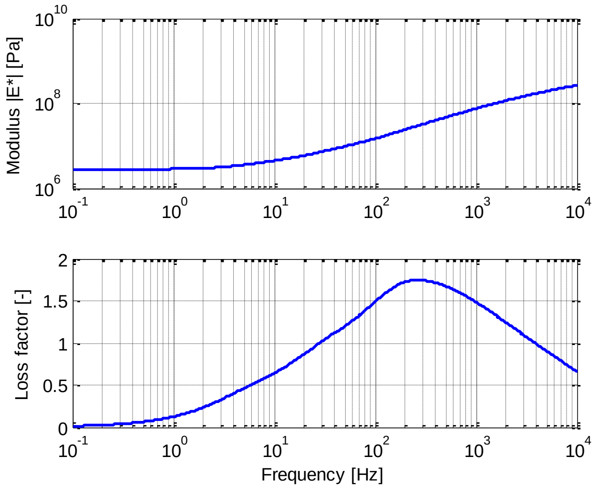


Figure 2. Modulus plot at 25˚C of the polymer (MIP-POL-001) developed by MI-Partners fitted by the Multiple Standard Model with n=10.

3. Material Characterization

In general, for high-end dynamic systems like positioning stages in mechatronics systems, an accurate material description over a large frequency range is necessary to predict the effect of damping properly. This requires an accurate measurements of the complex frequency dependent modulus (Young's and shear modulus) as function of frequency and temperature.

To measure the dynamic modulus of viscoelastic materials, DMTA (Dynamic Mechanical Thermal Analysis) is commonly used [1]. Actually, the dynamic stiffness of a sample is measured over a limited frequency range (typically up to 100 Hz) at various temperatures [0…120 ^{0}C]. For a polymer, qualitatively the effect of frequency is the inverse of the effect of temperature: the higher the frequency, the stiffer the polymer versus the higher the temperature, the weaker the polymer. This temperature to frequency behavior is described by the Williams-Landel-Ferry (WLF) relation.

In the DMTA, the WLF relation is used to interchange frequency and temperature to simulate the materials response over a large frequency range (typically 6…10 orders) by measuring over a limited frequency range, but at different temperatures.

For our applications, the results of such a DMTA is found to be insufficiently accurate, caused by e.g. errors due to the limited validity of the WLF relation over a larger frequency range. Furthermore, for the sake of general use, the fixation of a sample in such DMTA is simply done by preloading. Such fixation becomes doubtful at a higher stiffness of the sample (e.g. higher frequencies or lower temperatures) resulting in considerable measurement errors. Also, the applied strain levels are relatively high, typically > 0.5 %, whereas in our applications stain levels are typically < 0.01 %. Furthermore, such an apparatus is only feasible to handle predefined samples and is not able to characterize arbitrary entire damper modules.

To overcome the limitations of the DMTA apparatus, two dedicated dynamic measurement setups have been built at MI-Partners. They are placed in a temperature controlled enclosure which can also be used to change the temperature during evaluation (typically 15…40˚C).

The Damper Test Rig (DTR) has a generic interface which allows different samples up to 300x300x100 mm to be measured up to a frequency of 400 Hz. The measurement principle is shown in figure 3. The test apparatus generates a force by means of a direct drive linear motor. The generated force is measured by a load cell, positioned at the static end of the sample, and the deformation of the sample is measured by 2 capacitive sensors (symmetrically along the force line). The dynamic Young's modulus can be determined from the measured stiffness $k^*(\omega, T)$ by taking the sample geometry into account.

$$E^*(\omega, T) = \frac{k^*(\omega,T)}{\psi} \tag{8}$$

Where the constant ψ can be determined by a FEM analysis of the sample geometry.

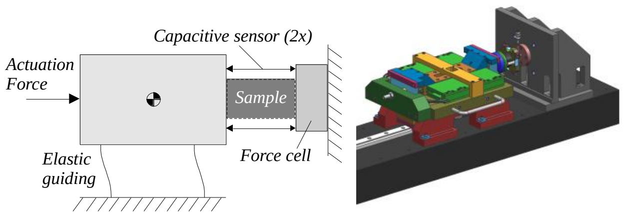


Figure 3. Principle (l) and overview (r) of the Damper Test Rig (DTR), for material and damper characterization up to 400 Hz.

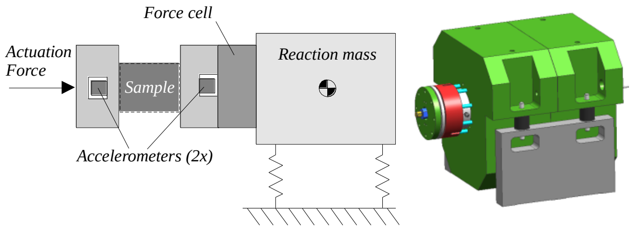


Figure 4. Principle (l) and overview (r) of the High Frequency Damper Test Rig (HF-DTR) for material and damper characterization in the range of 50-5000 Hz.

The High Frequency Damper Test Rig (HF-DTR) is depicted in figure 4. To enable characterization in the range of 50 Hz to 5 kHz, this test rig is equipped with accelerometers, to determine the deformation of the sample, and a piezo force cell. The force is generated by a modal exciter. To minimize errors due to surrounding dynamics (e.g. supporting table) a reaction mass is used. Both the modal exciter and the reaction mass are supported at about 5 Hz to create a low pass filter to and from the surrounding. To minimize rocking of the reaction mass and as such transversal acceleration and via crosstalk inside the sensors (accelerometers and force cell) measuring errors, the modal exciter and sample under test are properly aligned with the COG of the reaction mass.

The extreme frequency range up to 5 kHz sets tight requirements on the sample and sample fixation. Hence the HF-DTR is limited in accepting sample geometry and size. To obtain a measuring range up to 5 kHz, the first internal mode in the measurement direction are designed above 9 kHz. Furthermore, correction (post processing of k^*) is necessary to compensate for sensor dynamics (1) and the test setup dynamics (2) to reach an uncertainty below 1 % and 1 °.

The sensor dynamics (1) includes mainly the low-pass filtering in the amplifier of the accelerometer (given by the sensor supplier). The correction of the setup dynamics (2) is more comprehensive, this involves compliancy in the interfaces, inertia effects and the influence of the internal mode at 9 kHz toward lower frequencies. Based on a dynamic model, 1 DoF lump-mass model, the influences on k^* are predicted.

The test setup is verified by the measurement of a dedicated reference sample, assumed to behave purely elastic and so without damping over the full frequency range. This sample was statically calibrated in an external setup. Figure 5 shows the error of the dynamic measurement of this elastic reference sample on the HF-DTR setup with and without compensations.

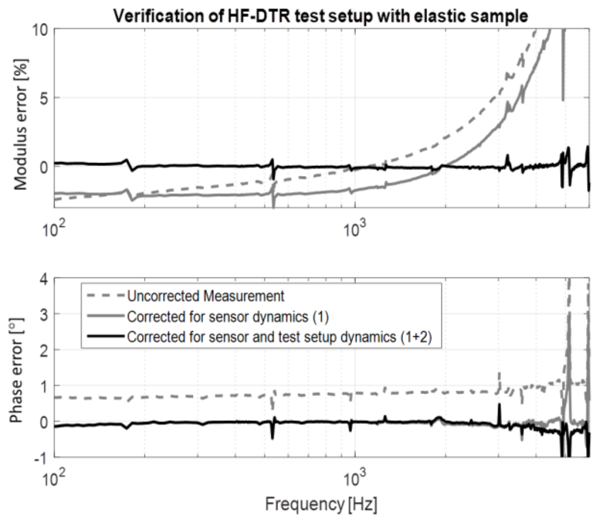


Figure 5. Verification of HF-DTR setup with an elastic reference sample and the effect of the sensor and test setup dynamics compensation.

3. Damper Design Methodology

From the characterization of the polymers it appears that the damping in shear (G-modulus) and tension loading (Young's modulus E) are the same (both result in the same loss-factor). From the characterization it also appears that the applied material behaves linear, even up to considerable larger strain levels than applicable in our application. Note that this does not apply in general for every polymer. Hence, the material can be treated as homogenous isotropic linear visco-elastic. As such, the dynamic stiffness of an arbitrary specimen is given by

$$k^*(\omega,T) = \psi E^*(\omega,T) \tag{9}$$

Where the constant ψ is determined by the actual damper geometry (geometric stiffness constant [m^{-1}]).

The diagram in figure 6 shows the entire design sequence, including verification. The system dynamics is modelled in Matlab (e.g. State Space model). As such, complex systems can be evaluated containing all kind of dynamic characteristics, including feedback loops. Also the frequency dependent properties (stiffness and damping) of the visco-elastic material is added in Matlab (Maxwell model shown in figure 1). Next, the damped dynamic system is evaluated and the properties (e.g. stiffness, position etc.) of the visco-elastic part can be optimized. This process results in an optimal design target stiffness $k^*_{optimal}$ for the damper.

Note that as the stiffness is frequency and temperature dependent, it is convenient to define a reference frequency and temperature (e.g.$(\omega_0, T_0) = (100\,Hz, 22\,℃)$) and work with the properties at this reference, e.g. $|k^*_{optimal}(\omega_0, T_0)|$.

Accordingly the damper can be designed to the corresponding stiffness by means of FE-analyses using a homogenous isotropic linear elastic model by applying $(|E^*(\omega_0,T_0)|, |G^*(\omega_0,T_0)|)$. Finally, the iteratively obtained actual geometry with k^*_{actual} is evaluated again in the dynamic system model in Matlab (Design Iteration Loop). Next, after finishing the complete design including interfacing, the damper is produced in close cooperation with an external partner. The last step is to verify the damper properties, e.g. the geometry and dynamic stiffness, which can be measured in our test setups (DTR or HF-DTR).

This complete design and realization sequence, from material characterization to design verification, can be realized within uncertainty of about 15 % regarding stiffness and damping. The variations include batch variations of the polymer, modeling and measurement inaccuracies. This small uncertainty allows for predictable and successful application of damping in high-end applications.

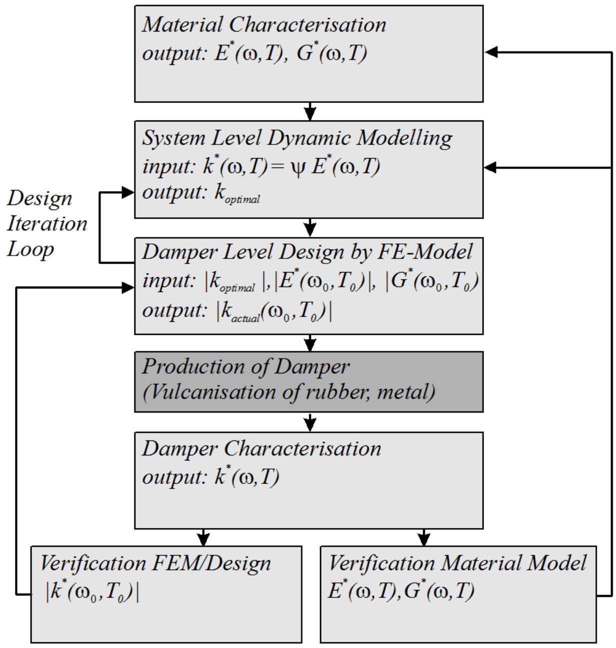


Figure 6. Damper design methodology.

4. Polymer Damper in a Positioning Stage

For highly dynamic stages, as for example used in fast tool servos for mass production of optical components, the servo bandwidth is typically limited by internal mechanical resonance frequencies. In our case, in order to increase the controller bandwidth to improve the dynamic performance of the system, a dedicated damper is designed to damp these internal resonances.

To test and verify the principle, a dummy stage was made with corresponding dynamic modes. The three dominant modes of the dummy stage are shown in figure 7. The polymer damper (see figure 8) is positioned in the suspension (left and right) of the mass (dynamic mass). The damper assembly (polymer damper with dynamic mass) acts as an additional dynamic mass which becomes part of the entire dynamic systems at several critical modes. Depending on the mode shape and the contribution of the dynamic mass (level of deformation of the polymer suspension), much or little damping will be introduced in this particular mode.

This principle is different from a so-called elastic tuned mass damper (TMD). The latter is actually no damper (no energy is dissipated), but it interacts with existing modes such that the mode shapes are changing and becoming more beneficial to the system. The design of such a tuned mass damper is very critical to achieve the proper interaction with the existing modes, as the shapes have to interfere properly. And it appears that in more complex systems, like positioning stages, a proper implementation of such tuned-mass damper is often not feasible.

The polymer based damper however, dissipates energy in every mode where the dynamic mass is interacting and the suspension is deforming, not depending on the precise motion (mode shape) of the dynamic mass. The result is a broad (frequency band) and robust damping principle.

In the actual system, no additional mass was added, but the weight of an existing sub-module was used to act as the dynamic mass. The corresponding damped and undamped bode plots from force input to displacement output (see figure 7) are shown in figure 9 and 10.

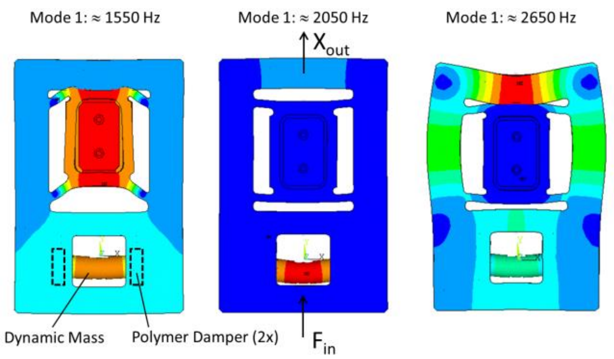


Figure 7. Dummy stage with comparable dynamics (modes 1…3) and input force F_{in} and output displacement X_{out}.

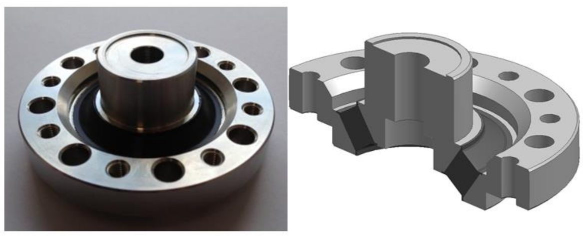


Figure 8. Polymer damper (vulcanized rubber-metal parts) suspending the dynamic mass.

The first dominant internal modes are in the range of 1.6…2.7 kHz and appear as highly undamped resonance peaks in the transfer function, thereby limiting the controller bandwidth.

The bode plot shows the measured and the simulated response, which are in good agreement. The deviation at the first 3 damped resonances are within 2 dB, which shows the high predictability of this methodology.

The polymer damper is able to suppress these dominant modes by 30 dB and more. It appears that the controller bandwidth could be increased by roughly 35 %. Further, as the transfer function becomes more "smooth", an enormous increase in system robustness is achieved.

This case shows the high potency of applying visco-elastic damping in a high-end mechatronic system.

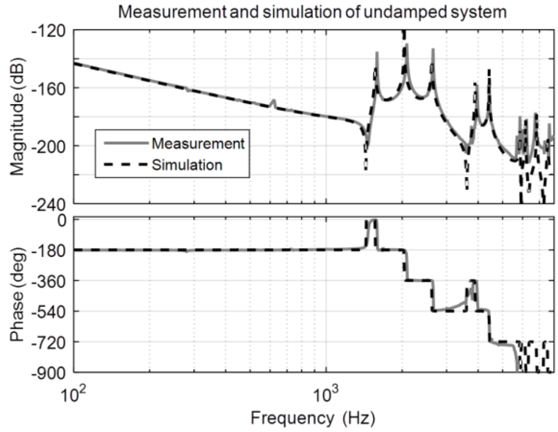


Figure 9. Measurement and simulation of the dummy system without damping.

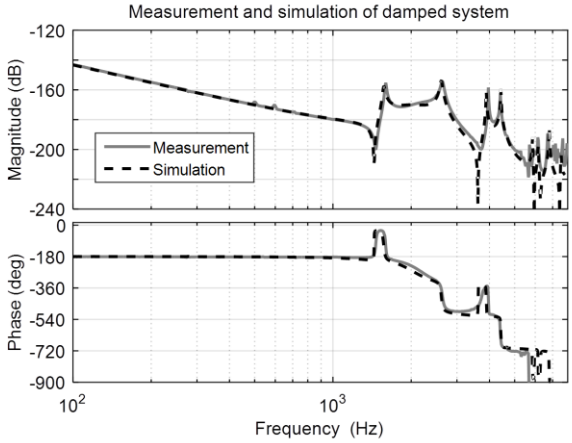


Figure 10. Measurement and simulation of the dummy system with damping.

5. Summary and Conclusion

It is shown that polymer damping technology can be applied in high-end dynamic positioning systems to improve servo bandwidth and system robustness. A polymer material with high damping and reproducibility has been developed and the dynamic properties (storage and loss moduli) are accurately characterized with two dedicated dynamic measurement rigs. Finally, a design methodology has been developed, including dynamic system modelling and FE-models of the polymer, resulting in a systematic design approach. It is shown that the application on system level has such high predictability that this technology can be used in the development of high-end positioning systems.

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