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PRACTICAL ASPECTS IN MEASURING VIBRATION DAMPING OF MATERIALS

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ABSTRACT

Already many years material damping data is available from literature, but often there is no consistency in the presented damping values. One reason is the wide availability of testing apparatus for measuring damping, each having its own pros and contras. Next to that, damping data can be analysed in many ways such that obtaining a uniform damping value for a specific material is far from obvious. The work in this paper focuses on finding an accurate and repeatable test method for measuring low damping values. It is found that the suspension has a large influence in test setups for low damping measurements and that material damping varies with sample geometry and stress level.

Keywords: material damping, free-vibration, viscous damping model, non-linear damping

INTRODUCTION TO MATERIAL DAMPING CHARACTERIZATION

The characterization of the damping capacity of metals, alloys and other materials is very sensitive to testing conditions such as temperature, strain amplitude, frequency, humidity, specimen geometry, stress-field state and specimen grip system. Besides this, there is a wide range of evaluation techniques and parameters for reporting damping values. Therefore, comparison of damping capacity data among different materials is made difficult. Already in 1972 report Adams and Fox many inconsistencies for damping values of nominally identical materials (Adams and Fox, 1972). This resulted from an inadequate knowledge of the distribution of energy lost in all parts of the apparatus, including that in the specimen itself. Even if one joint is the only significant source of energy dissipation, it may be sufficient to cause misleading results. In addition to this has been reported (Lazan and Goodman, 1961) that any comparison of data from damping measurements must be considered carefully since data expressed as relative damping units, such as loss factor or logarithmic decrement, have been obtained on a variety of specimen types and stress distributions. This paper now takes a closer look on how damping results are affected by external sources of damping. Three influences which cause unwanted damping are examined in detail: (i) specimen excitation, (ii) specimen vibration response measurement and (iii) boundary conditions, i.e. how the specimen is connected with the environment. Two different test configurations are examined for their applicability in measuring damping for metal specimens. The one suiting best will be used for further experiments which take a closer look on how damping is affected by boundary conditions, sample size and stress level.

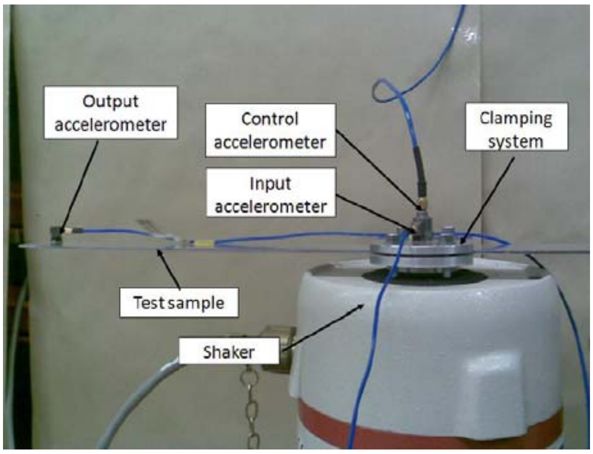
CHOOSING THE APPROPRIATE DAMPING MEASURING METHOD

Before choosing the right damping measuring method among the wide variety of experimental techniques, first some considerations should be taken. First of all one should consider whether or not linear or non-linear damping is expected. The former implies that damping does not change with the excitation level whereas in case of non-linear damping the properties significantly change with excitation level. From a practical point of view, linear damping is chosen because of ease of modelling. If eventually damping is found to be non-linear, damping may be defined within specific limits at which damping shows approximate linear behaviour. The next consideration to be made is how the system under test will be examined, i.e. does a single degree of freedom (SDoF) approach suit or is a multi degree of freedom (MDoF) approach necessary? In case of light damped systems and non-modal overlap (Ewins, 1984), model reduction to a SDoF system is allowed and it is preferred because of ease of damping modelling. Further on, an appropriate damping model has to be chosen. Very often viscous damping, for which the damping force is proportional to velocity, is preferred because it shows advantages in terms of physical and mathematical simplicity. Besides this damping model, the hysteretic or viscoelastic damping model can also be used. Hereby is the energy dissipated per cycle independent of the frequency and proportional to the square of the amplitude of vibration. The latter has found to suit best for modelling damping in structural applications (Clarence, 2005; Rao, 2005; Riberio et. al., 2005; Lazan, 1961). At last, the test configuration of choice determines if measured data will be analysed in the frequency- or time domain. Both can be applied in case of viscous damping because both time- and frequency domain analysis are equivalent whereas hysteretic (or viscoelastic) damping is best modelled through frequency domain analysis because harmonic excitation is assumed with the latter model. Non-harmonic vibrations such as transient oscillations or free vibrations are not adequately described by means of the hysteretic damping model.
Within this paper, the free-vibration test method is proposed and viscous modelling in a SDoF configuration will be used. The customary procedure is to bring the specimen out of its equilibrium position and then to release the excitation. As a consequence, the specimen will dampen out with exponential decaying sinusoidal amplitude at the damped natural frequency ω_d. From the decaying signal follows the logarithmic decrement δ which is the natural logarithm of the amplitude ratio of successive cycles: $\delta = \frac{1}{n} ln\left(\frac{A_1}{A_{n+1}}\right)$, with n the number of cycles and A_1 and A_{n+1} the amplitude at the start respectively the end of these *n* cycles in the decaying response signal. In case of viscous damping, the damping ratio is: $\zeta = \frac{\delta}{2\pi}$. Since the logarithmic decrement is more accurate with increasing number of cycles *n*, the free vibration method ideally suits for low damping materials.

The following sections describe two types of free vibration test-setups from which damping is assessed. Both methods are examined for their sensitivity related to excitation and response measurement and also how the specimen is supported.

Cantilever shaker testing and accelerometer response measurement

The first test setup includes a simple cantilever beam which is mounted onto the outgoing shaft of the vibrator (Vanwalleghem, 2010). The clamping is realized through fixing the sample between two bolted aluminium plates. In Fig. 1 can be seen how the specimen is mounted onto the shaker. The beam response is measured through an accelerometer located at the tip of the sample, control and input accelerometers are necessary for vibration control. The free vibration of the specimen is initiated through a triangular shock at time *t=0s* as depicted in Fig. 2. Once the higher order modes are dampened out, the fundamental mode remains which is then used to assess the damping ratio through the logarithmic decrement analysis.



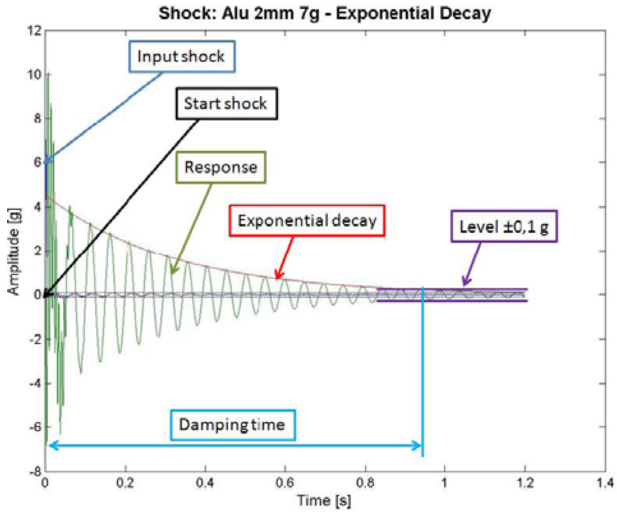


Fig. 1 Cantilever beam test configuration with shaker excitation and accelerometer response measurement

Fig. 2 Free vibration response of cantilever beam due to impulse excitation

From initial analysis it follows that the measured fundamental resonant frequency of the cantilever beam does not match with the theoretically calculated. Two observations have found to be responsible for this. At first, the shaker dynamics play a major role in the frequency shift. Acceleration measurements at the armature of the shaker, near the clamping system, indicate that the armature of the shaker has no rigid foundation because the armature is supporter through a spring, thus no theoretical fixation at the base of the beam is attained. The second aspect relates to the clamping system itself. The upper clamping plate lifts off the test sample while tightening the bolts; consequently the specimen is allowed to vibrate in between the fixation system. Both issues have found to be significant in the frequency shift between the measured and calculated resonant frequency.

Since this test setup affects the resonant frequencies of the cantilever beam, the measured damping values will certainly be influenced by this test configuration. One source of additional damping is due to the inherent damping of the shaker armature which moves in the magnetic coil of the shaker. This motion is already observed with the acceleration measurements at the armature of the shaker. The second source of additional damping is frictional damping because of the vibrating specimen in between the two clamping plates. Another source of additional damping is due to the accelerometer mounted at the tip of the specimen under test. The major error is not the accelerometer itself but the oscillating wire of the accelerometer during specimen vibration. The inertial force of the wire and accelerometer working on the sample changes the specimen dynamics and will certainly influence the damping results. This effect will even increase with higher resonant frequencies and especially light weight structures are susceptible to damping measurement errors if the response is measured with an accelerometer.

From previous observations can be concluded that many external sources may influence the dynamics of the specimen under test. Because of this, another test configuration is necessary which is less (or not) subject to external sources of damping.

Free vibration damping through contactless excitation and response measurement

To eliminate this unwanted damping from external devices, specimen excitation is done through acoustic wave excitation using a loudspeaker and specimen response is measured with a laser Doppler vibrometer (LDV). Both methods assure there will be no effect of devices attached to the specimen introducing additional damping. To eliminate the effect of the boundary conditions, the sample is mounted vertically through very thin nylon wires, therefore the suspension is orthogonal to the deflection of the vibrational mode shape.

Fig. 3 shows how the specimen is positioned in between the loudspeaker and the laser beam of the LDV. The customary procedure is to bring the specimen out of its equilibrium state through harmonic wave excitation at the resonant frequency of interest. Once steady-state vibration is established, the acoustic wave excitation is stopped such that the specimen is in free-vibration condition.

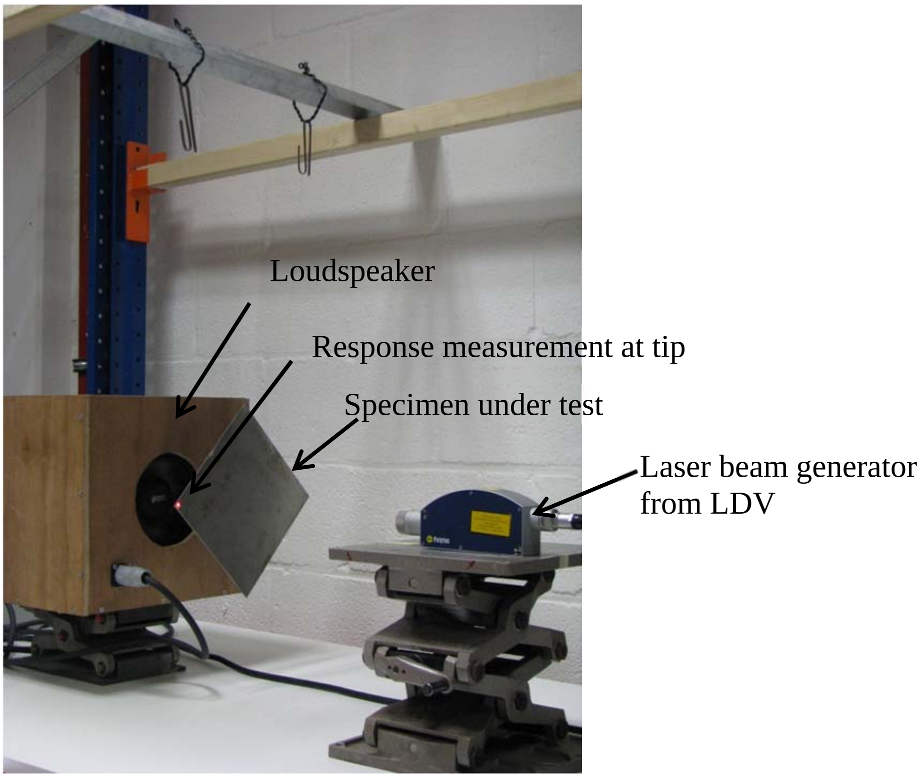


Fig. 3 Free vibration test setup with contactless excitation and response measurement

Damping behaviour characterization

Before the test-setup is investigated for its sensitivity to external damping sources, the damping behaviour of the sample is evaluated. The specimen under test is a 320x320x2mm cold rolled construction steel plate which is suspended at the nodal positions of the first (torsion) mode shape. The suspension wires have a length of 400mm. The excitation level is increased up to a response amplitude of 80mm/s, measured at an anti-nodal position of the first mode shape. Fig. 4 shows the response velocity as function of time from the specimen at free vibration. From initial analysis it is clear that not one single exponential function fits the decaying amplitude which makes that multiple exponential fits are necessary to assess the damping capacity of the specimen. Within five amplitude ranges, from 80mm/s up to

2.5mm/s, an exponential is fitted onto the measured response signal. The damping ratio varies from $13 \cdot 10^{-5}$ to $9.2 \cdot 10^{-5}$ at the amplitude range of 80-40mm/s to 5-2.5mm/s respectively. Fig. 5 shows a detailed view of the best exponential fit in the range 80-40mm/s, other exponential curves do not fit with vibrational decay. Analogue observations are found for other amplitude ranges, Fig. 6 shows that within the range 20-10mm/s again one single exponential fits the best whereas other do not. These findings, namely that the damping ratio is amplitude dependent, indicate that non-linear damping is present.

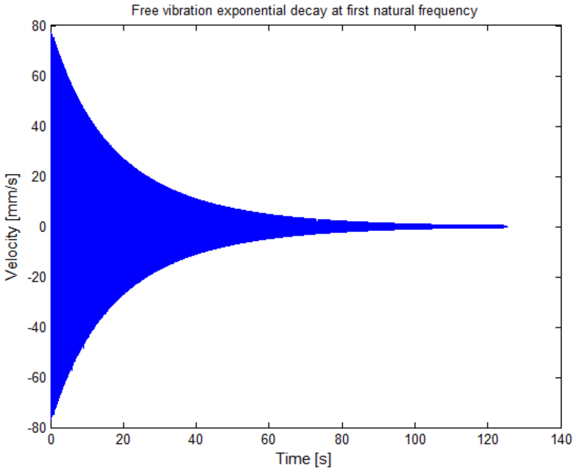


Fig. 4 Exponential decay of steel plate sample at first resonant frequency: response velocity as function of time

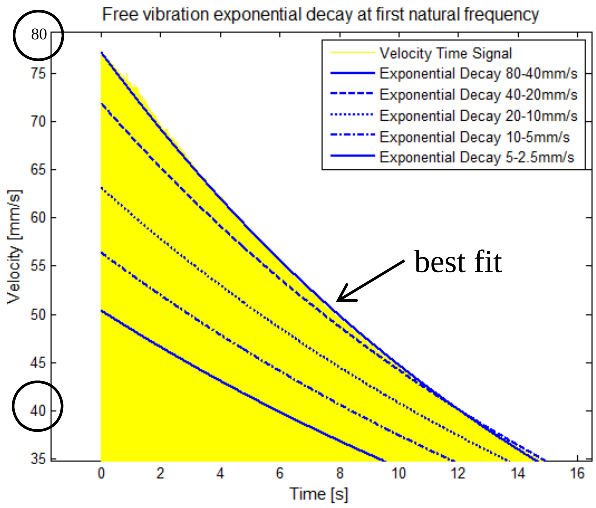


Fig. 5 Non-linear damping - detailed view at high response level

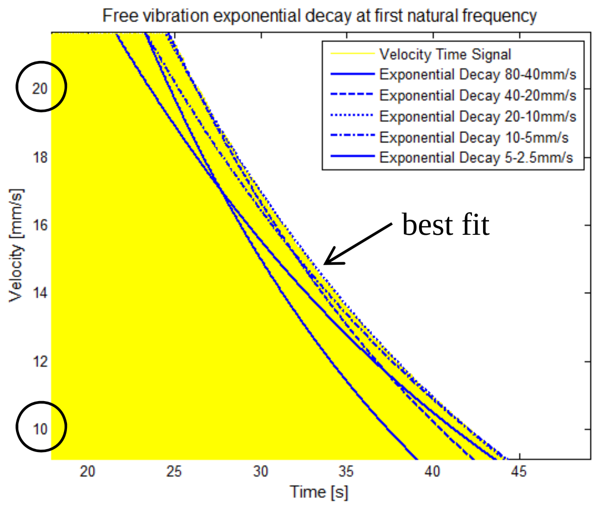


Fig. 6 Non-linear damping - detailed view at intermediate response level

Table 1 Repeatability of damping measurements gives an indication of the accuracy realized with this test method. From five consecutive measurements follows an average viscous damping ratio value and its corresponding spread. The very low spread allows for accurate comparison among different damping values, if any differences are observed, it will not be due to measurement errors.

Table 1 Repeatability of damping measurements

Response amplitude [mm/s]	Average damping ratio ζ	Spread
80 → 40	13.0E-05	6.4E-07
40 → 20	11.3E-05	2.3E-07
20 → 10	10.3E-05	3.2E-07
10 → 5	9.68E-05	4.4E-07
5 → 2.5	9.20E-05	1.2E-07

Sensitivity analysis of the free vibration damping test setup

To find out the influence of the suspension type on the measured damping, the exponential decaying amplitude is measured with the 400mm length suspension wires at the nodal and anti-nodal position of the corresponding resonant mode shape. The damping ratio shows a small increase if the suspension cords are placed at the anti-nodal positions of the first mode shape. This additional damping is because of the suspension wires preventing the sample to vibrate in free condition. If the same experiment is repeated for damping measurements in the second (saddle) mode, it is found that damping hardly increases if the suspension point is changed from nodal to anti-nodal position. This may be due to smaller displacement deflections at the anti-nodal position of the second mode shape due to the higher resonant frequency. A graphical view of the previous is depicted in Fig. 7. Shorter suspension wires make this effect even worse. Decreasing the suspension length from 400mm to 200mm drastically increases the damping ratio of the first mode shape if suspended at the anti-nodal positions whereas the damping ratio remains the same if suspended at the nodal positions. Measured damping values from the second mode shape hardly change with the position of the nylon cords. Table 2 and Fig. 8 show the damping ratio for the first and second resonant mode shape, twice for specimen suspension at the nodal and anti-nodal position with a suspension length of 200mm. The relative difference between damping ratios from suspension at nodal and anti-nodal positions increases up to more than 50% for the first mode shape whereas results for the second mode shape remain the same.

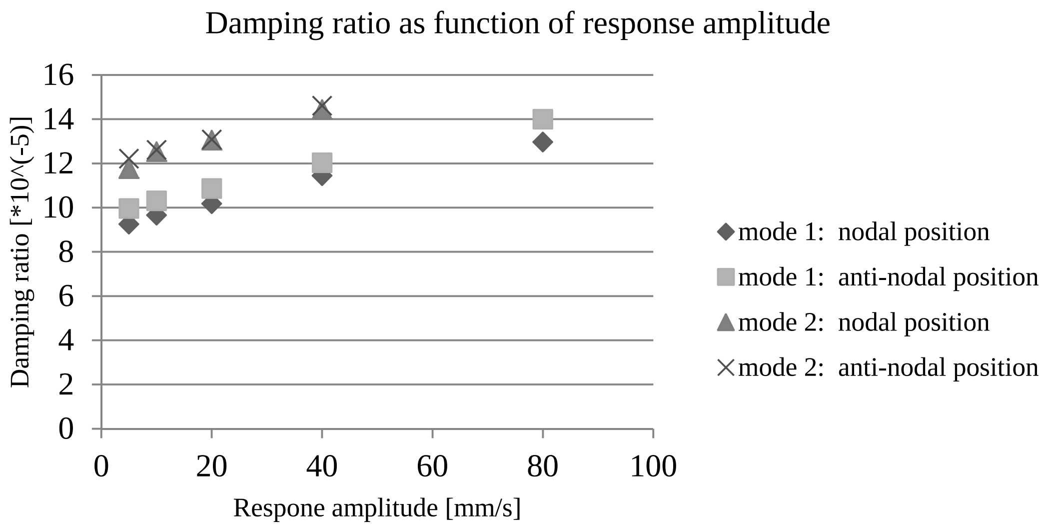


Fig. 7 Variation of damping ratio as function of suspension position (suspension length = 400mm)

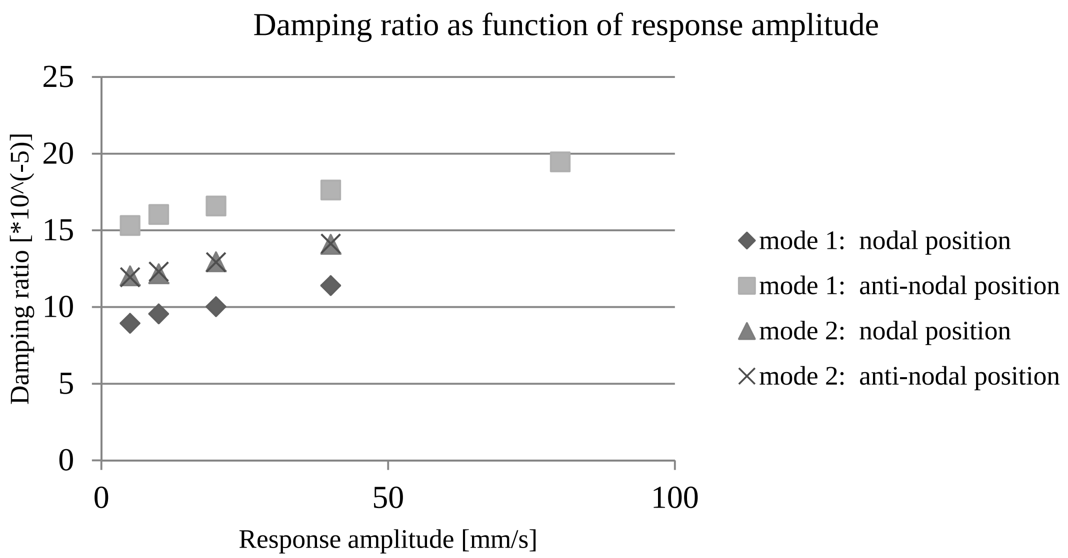


Fig. 8 Variation of damping ratio as function of suspension position (suspension length = 200mm)

Table 2 Sensitivity of damping ratio to different test cases (suspension length = 200mm)

Viscous damping ratio ζ [*10^(-5)]

	Suspension position at specimen	Resonant frequency [Hz]	Response amplitude [mm/s]				
			80→40	40→20	20→10	10→5	5→2.5
mode 1	nodal position	67.82	-	11.40	10.03	9.56	8.94
mode 1	anti-nodal position	67.95	19.47	17.63	16.60	16.04	15.32
	Relative error			55%	66%	68%	71%
mode 2	nodal position	96.05	-	14.09	12.95	12.17	12.04
mode 2	anti-nodal position	96.1	-	14.14	12.93	12.30	11.95
	Relative error			0%	0%	1%	-1%

From these experiments it follows that even small external modifications, such as changing the suspension location, affect the damping ratio significantly. Based on these results it could be thought that using wired sensors, such as accelerometers or strain gauges, for measuring the vibrational response of light damped specimens is even worse in influencing damping values.

INFLUENCE OF SPECIMEN SIZE ON DAMPING RATIO

As the previous method is found successful in measuring damping with high accuracy, elaborate work on how damping is affected by specimen size is possible. Six steel plate samples are selected; the size varies from 100x100x2mm up to 350x350x2mm with increasing steps of 50mm. The first vibration mode shape (torsion mode) is for all samples the same; this eliminates the influence of the specimen stress-field state on the damping ratio.

Fig. 9 plots the damping ratio as function of the maximum tip displacement, it is seen that the damping ratio increases linear with the deflection amplitude and the slope is steeper with increasing resonant frequency (or smaller sample size). Furthermore if the damping ratio is plotted against the sample size (represented as the first resonant frequency) at different strain rates, the damping value tends to shows a minimum value, see Fig. 10. First a steep decrease of the damping ratio with increasing resonant frequency is observed up to a minimum value and further increases slowly as resonant frequency increases. This curvature is found for all strain rate intervals except for the smallest samples because there is lack of excitation power to have a response of more than 40mm/s. From the same graph it also follows that higher strain rates correspond with higher damping ratios, which is true for all sample sizes.

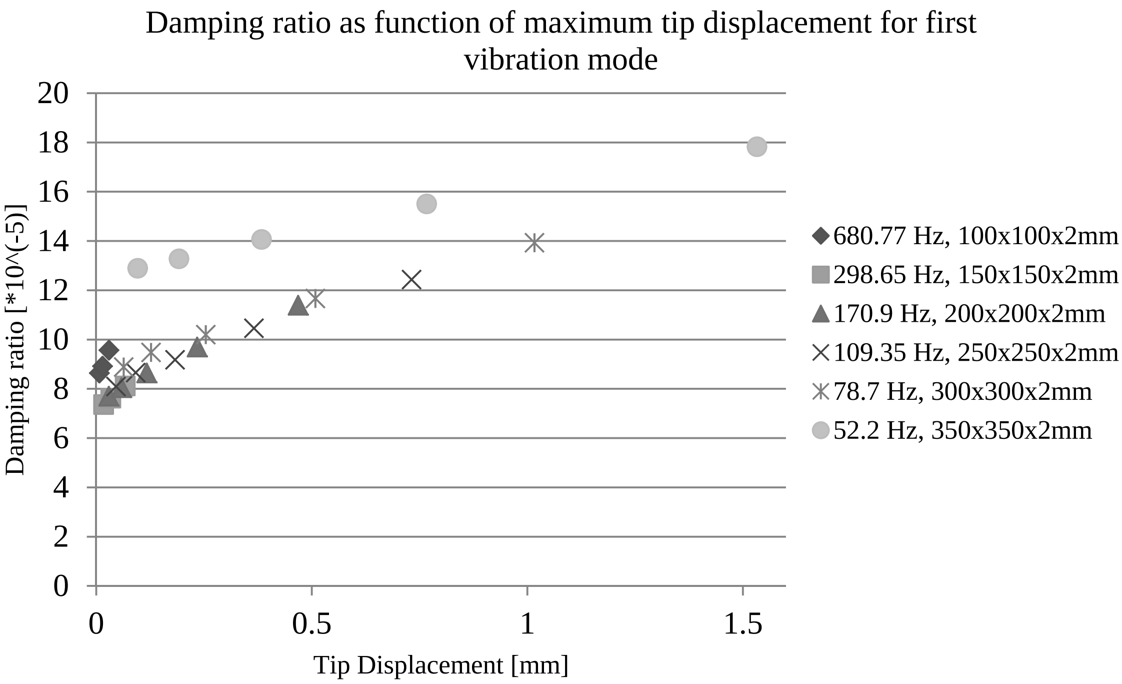


Fig. 9 Stress level dependent damping ratio for various sample sizes

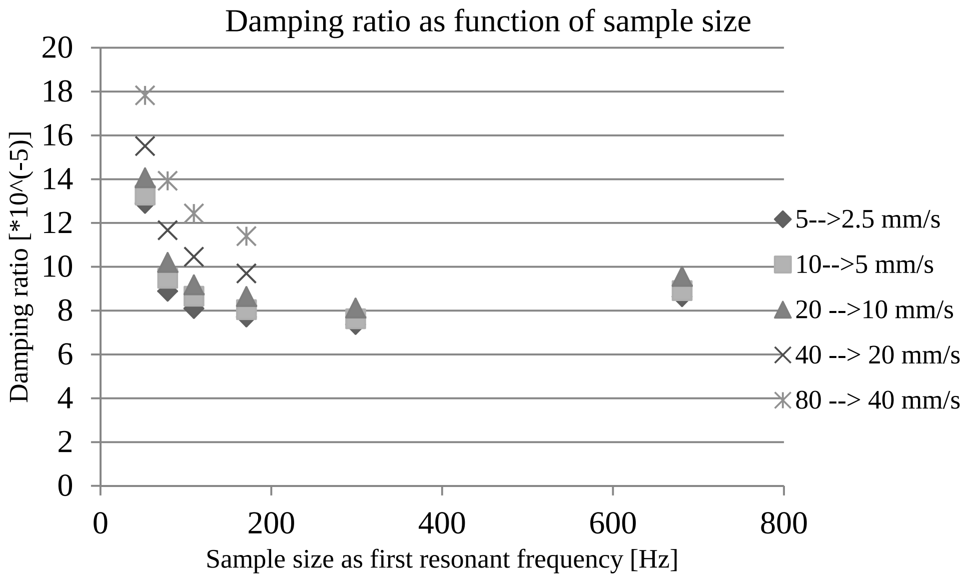


Fig. 10 Damping ratio as function of sample size for different strain rates

CONCLUSION

Two different test configurations to measure low damping values of metal specimens in free vibration condition have been proposed. Any contact making device, necessary for response measurement or specimen excitation, has shown to impair resonant frequencies and corresponding damping ratios significantly. The ideal test configuration exists of contactless excitation and response measurement, in combination with nodal specimen suspension. This setup shows high accuracy and reproducibility in free vibration testing of low damping specimens.

Measurements for a 320x320x2mm metal plate show non-linear damping behaviour and the damping is very susceptible to how the specimen is suspended. Anti-nodal sample suspension gives rise to significant larger damping values compared to nodal sample suspension. Clearly external damping losses in free vibration testing are minimized through contactless excitation and response measurement and nodal specimen suspension.

From damping measurements at different sample sizes it results that the damping ratio is definitely not a material constant, but it varies widely with frequency and stress level within the same specimen stress-field state (cf. resonant mode shape). Moreover all sample sizes show increasing damping with increasing stress level and the rate of increase is steeper for smaller samples (equivalent with higher resonant frequencies). The damping ratio as function of sample size is less clear as the damping ratio shows a minimum value within this range of sample sizes.

REFERENCES

Adams, R.D., Fox. M.A.O. Measurement of the damping capacity and dynamic modulus of high-damping metals under direct cyclic stresses. Journal of Physics D: Applied Physics, 1972, 5, p. 1274-1283.

Clarence, W. Vibration damping, chapter from Vibration and shock handbook/ed. by Clarence, W., 2005

Ewins, D. J. Modal Testing – Theory and practice, Research Studies Press, 1984

Lazan, B.J., Goodman, L.E. Material and interface damping, chapter from shock and vibration handbook/ed. by Cyril M. Harris, 1961

Rao, S.S. Mechanical vibrations, Pearson Higher Education, 2005

Ribeiro, A.M.R., Maia, N.M.M., Silva, J.M.M., Reis, L. Free vibration response using the constant hysteretic damping model. The 11th International conference on vibration engineering, Timisoara, Romania, September 27-30, 2005

Vanwalleghem J. Study of the damping and vibration behaviour of flax-caron composite bicycle racing frames. Ghent University, Master’s thesis 2009-2010