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## Active cancellation of unwanted excitation when measuring dynamic stiffness of resilient elements

Ruud Vermeulen, René Lemmen, Arthur Berkhoff and Jan Verheij  
TNO TPD, P.O. Box 155, 2600 AD Delft, The Netherlands

### Abstract

Laboratory measurements of the vibro-acoustic transfer properties of resilient elements are dealt with in a series of five International Standards (ISO-10846-1 through 10486-5, partly still forthcoming). The measured quantity that is standardised, is dynamic stiffness. Because of the multidirectional nature of the vibration transmission, the standards cover separate measurements of stiffness for three perpendicular excitation directions. For each of the three orthogonal translational excitations, the input for the unwanted directions have to be at least 15 dB lower. To meet this requirement over a large frequency band may be quite difficult, especially in case of ‘transverse’ excitation. Then special measures have to be taken to suppress rotation inputs. In this paper active vibration control on the input side of a test element is shown to be useful to suppress unwanted input rotations. The applicable frequency range depends on the controller available. The results presented show the feasibility of the method and good suppression results up to 250 Hz. New tests with a state-of-the-art controller are planned to cover a wider frequency range and to enable the use of a chirp as a driving signal.

### 1. Introduction

Consider the system in Figure 1 for the measurement of dynamic transfer stiffness, using the so-called indirect method, see [1]. The resilient element (3, usually a rubber mount or a flexible coupling) is placed between two blocks (2, upper block with mass  $m_u$  and 4, lower block with mass  $m_l$ ). Relatively soft rubber mounts (1 and 5) are placed at the top and the bottom, in order to decouple the system dynamically from the environment. This whole system is placed in a very stiff test rig of which only the lower part (6) is drawn. A static preload  $F_{\text{preload}}$  is applied by a hydraulic system. The upper block is excited in the desired direction by the primary dynamic force  $F_p(t)$  (here horizontally).

The frequency dependent dynamic transfer stiffness is defined as

$$k = \left. \frac{F_l}{x_u} \right|_{x_i=0} \quad (1)$$

where  $F_l$  is the blocking force at lower side of the resilient element in the desired direction and  $x_u$  is the translational displacement at the upper side of the resilient element in the desired direction.  $x_i = 0$  denotes that other input displacements and rotations must equal zero. For harmonic signals  $x_u$  can be written as  $x_u = -a_u/\omega^2$ , where  $a_u$  is the translational acceleration of the upper block and  $\omega$  the radian frequency.

The blocking force is estimated from  $F_l \approx m_l a_l$  ( $a_l$  is the translational acceleration of the lower block). This approximation is valid for frequencies well above the natural radian frequency  $\omega_0$  of the mass-spring-system consisting of the lower block and the stiffness of the test element (3) and the lower decoupling mounts (5). Eq. 1 now can be rewritten as

$$k = -\frac{m_l a_l \omega^2}{a_u}, \quad \omega \gg \omega_0. \quad (2)$$

A detailed description of this measurement method according to the ISO international standards can be found in [1].

In practice it often appears to be hard to make the upper block move in only one direction. Especially when the upper block is excited in the X-direction, in practice often unwanted rotation about the Y-axis is introduced. This can be caused by for example small alignment errors or a different impedance that the upper block ‘experiences’ at its lower and upper side, which may result in erroneous measurements of  $k$ . Creating a more symmetrical test set-up - by applying an identical mount as the resilient mount to be tested upside down as the upper decoupling mount (1) - can reduce this. However, this regularly does not yield sufficiently improvement. In case the resilient element to be tested is a flexible coupling, it is mostly even not applicable in practice.

In this paper the results are presented of the use of active control in order to suppress rotation about the Y-axis. A secondary dynamic force  $F_s(t)$  is applied on the upper block (see Figure 1). Only two degrees of freedom are regarded in this paper: translation in the X-direction and rotation about the Y-axis.

At two positions the accelerations  $a_1$  and  $a_2$  are measured. They can be decomposed into a component  $a_{t+r}$  due to both translation and rotation and a component  $a_r$  purely due to rotation. The following definitions are used:

$$\begin{aligned} a_{t+r} &= \frac{a_1 + a_2}{2} \\ a_r &= \frac{a_1 - a_2}{2}. \end{aligned} \quad (3)$$

If  $a_r \rightarrow 0$  then  $a_{t+r} \rightarrow a_t$ : pure translation remains.

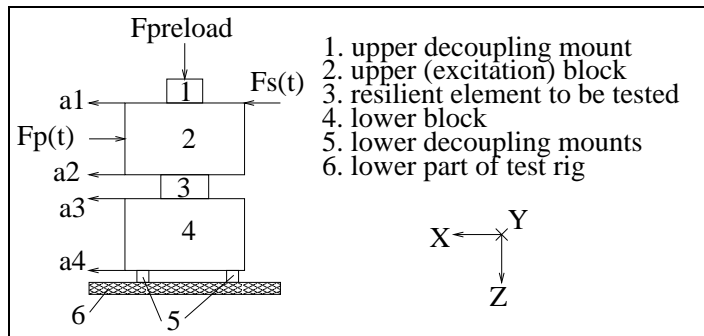


Figure 1: Test set-up for X-direction.

## 2. Requirements

According to [1] the following expression must be valid in order to sufficiently suppress unwanted vibrations:

$$L_a(\text{excitation}) - L_a(\text{unwanted}) \geq 15 \text{ dB}. \quad (4)$$

The standards are solely concerned with dynamic transfer stiffnesses of translational inputs. By requiring that translational accelerations perpendicular to the excitation direction (at the lower part of the upper block) are at least 15 dB lower than the translational accelerations in the excitation direction, it is assumed in [1] that unwanted input rotational accelerations are also sufficiently suppressed.

The upper block of Figure 1 has a width of 440 mm and a height of 240 mm. If it is assumed that it is only translating in the X-direction and rotating about the Y-axis, then  $a_X = a_{t+r} \rightarrow La_X = La_{t+r} = L_a(\text{excitation})$  and  $a_Z = \frac{440}{240}a_r \rightarrow La_Z = La_r + 5 \text{ dB} = L_a(\text{unwanted})$  (where  $L_{\text{value}} = 20 \log |\text{value}|$ ). So Eq. 4 can be rewritten as

$$\Delta La_{(t-r)} \stackrel{\text{def}}{=} La_{t+r} - La_r \geq 20 \text{ dB}. \quad (5)$$

## 3. Measurements

An active control unit, with on-line system identification, was used to suppress  $a_r$  at discrete frequencies. A primary sine was created by a signal generator and was fed to the primary exciter which yielded a primary force  $F_p(t)$ . The signal generator also forwarded a synchronisation signal to the control unit. Finally the control unit also received the error signal  $a_r = \frac{a_1 - a_2}{2}$ . A control signal was sent from the control unit to the secondary exciter. This second exciter was mounted at the upper side of the upper block and applied the secondary force  $F_s(t)$ , see Figure 1.

The accelerations  $a_1, a_2, a_3$  and  $a_4$  were measured for 17 frequencies. The upper left plot of Figure 2 shows the influence of the active control on  $a_r$  and  $a_t$ . The translation remains globally the same, whereas the rotation decreases approximately 20 dB for all frequencies.

The lower left plot shows  $\Delta La_{(t-r)}$ . It can be clearly seen that  $\Delta La_{(t-r)}$  increases when control is used. At 16 frequencies  $\Delta La_{(t-r)}$  exceeds the required 20 dB. At the remaining frequency (14 Hz)  $\Delta La_{(t-r)} = 18 \text{ dB}$ . The effect of the suppression of  $a_r$  on  $k$  can be seen in the right plot of Figure 2 for discrete frequencies from 40 Hz. For lower frequencies Eq. 2 is not valid. The solid line is the curve that is found when a chirp is used as a driving signal (no control is used). Using control clearly suppresses the disturbances caused by  $a_r$ . Around  $f = 100 \text{ Hz}$  the influence of a reduction of  $a_r$  can clearly be seen. For higher frequencies the curve increases somewhat, which is the flank of the first standing wave in the resilient element under test.

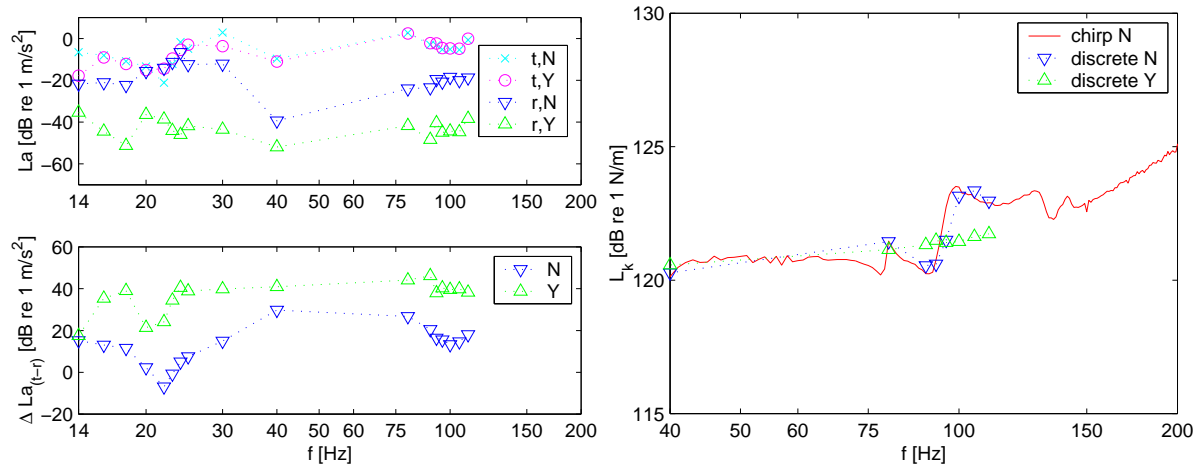


Figure 2: Influence of active control at discrete frequencies on rotation and translation of upper block (left) and dynamic transfer stiffness  $k$  (right).  $r$  = rotation,  $t$  translation,  $N$  = no control and  $Y$  = control.

The procedure of using active control looks very promising, however it is rather time consuming, because for each desired frequency a measurement must be performed. Using a chirp as the driving signal can enhance this.

Recent tests, with a controller based on off-line secondary path identification, show that control can suppress  $a_r$  sufficiently up to 250 Hz at discrete frequencies. Current research focuses on using a state-of-the-art controller, which will enable using chirps as a driving signal and which will further broaden the applicable frequency range.

## Conclusions

Applying active control at discrete frequencies, in order to reduce unwanted rotation, can be used very well when measuring dynamic stiffness of resilient elements. Typical reductions of 20 dB were achieved up to 250 Hz.

In practice it is not convenient to perform measurements at discrete frequencies, because this is very time consuming. Therefore current research focuses on applying a state-of-the-art controller in order to allow a chirp as the driving signal. Another goal of using such a controller is to further broaden the applicable frequency range.

## References

1. ISO 10846, "Acoustics and vibration – Laboratory measurement of vibro-acoustic transfer properties of resilient elements", 1997.