The advantage of linear viscoelastic material behavior in passive damper design—with application in broad-banded resonance dampers for industrial high-precision motion stages

Citation for published version (APA):
The advantage of linear viscoelastic material behavior in passive damper design—with application in broad-banded resonance dampers for industrial high-precision motion stages

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A R T I C L E   I N F O

Article history:
Received 2 November 2015
Received in revised form
17 May 2016
Accepted 18 May 2016
Available online 7 October 2016

Keywords:
LVE material
Passive damping
Motion stage design
Optimal
Structural damping

A B S T R A C T

In this paper we demonstrate the advantage of applying viscoelastic materials instead of purely viscous materials as damping medium in mechanical dampers. Although the loss modulus decreases as function of frequency in case of viscoelastic behavior, which can be interpreted as a decrease of damping, the viscoelastic behavior still leads to an increased modal damping for mechanical structures. This advantage holds for inertial-mass-type dampers that are tuned for broad-banded resonance damping. It turns out that an increase of the storage modulus as function of frequency contributes to the effectiveness of mechanical dampers with respect to energy dissipation at different mechanical resonance frequencies. It is shown that this phenomenon is medium specific and is independent of the amount of damper mass.

1. Introduction

Motion stages are mechanical structures that are position controlled. This concept finds a wide range of applications in production and measurement equipment [3]. The performance of a complete system depends on the properties of the position controlled closed-loop system, of which the performance is usually limited by the flexible (non-rigid) behavior of the mechanics [2]. This flexible behavior associated with the mechanical resonance frequencies, is excited during acceleration of the motion stage. One solution to improve the closed-loop performance is by increasing the modal damping at these resonance frequencies [17].

A Robust Mass Damper (RMD) is a passive device which has the ability to dissipate kinetic energy at a broad range of resonance frequencies [21], where the Tuned Mass Damper (TMD) acts at a specific single resonance frequency [5,12]. The dissipation of energy at a resonance frequency is called modal damping and is related to the amplification factor of the resonances of a mechanical system. In order to maximize the damping properties for a single resonance, i.e., reduce the amplification factor, active and semi-active variants of the TMD are developed [4,6], as well as strategies with multiple
tuned dampers have been invented [7,8]. Other principles [16] and materials are also explored [1,9,14,23,24]. RMD is relatively new in the field of motion stages and differs from the TMD by its robustness against parameter variations and the frequency range in which the dampers are effective due to the high damping value. This large damping value implies that the damping becomes a more important parameter than in case of a TMD application.

This paper presents the influence of linear viscoelastic behavior on the amount of modal damping that can be obtained over a broad range of resonance frequencies of a mechanical structure. In high-precision motion stage design, the linearity of the damping material is an advantage with respect to the design of the motion controller [15], which is often based on a linear time-invariant (LTI) approach. This implies that nonlinear damping materials, although the damping performance might be good [11,20,25], are suited less for improving the damping properties without introducing side-effects.

In general, the existing idea is that the phase of a damping medium should show a constant 90 deg over a large frequency range, i.e. behave purely viscous, and so obtain large damping values in a mechanical structure. Linear viscoelastic material properties show a decreasing phase angle for increasing frequencies and this is thought to result in a loss of damping for higher frequencies [5]. This paper presents the advantageous application of linear viscoelastic behavior in inertial mass-based damper design, i.e., making use of the decrease of the loss modulus as function of frequency and turning it into an advantage for the mechanical system.

2. Motion stage model and RMD effect

Fig. 1a shows an example of a motion stage as applied in the lithographic industry for processing wafers. This photo shows a ceramic structure, which has a low material damping, and, therefore, a low modal damping. The new RMD devices on the four corners are applied to improve these damping values. The arrow in Fig. 1a depicts the location of the actuation force and position sensor. An exploded view of the Robust Mass Damper is shown in Fig. 1b. A dynamic model of these dampers is depicted in Fig. 2a. Parameter \( x_f(t) \) represents the vibration displacement of the stage corner, which enters the damper base mass \( m_w \) with displacement \( x_{wd}(t) \) via the stiffness \( c_f \) and damping \( d_f \) of the stage-damper interface. The effective damper mass \( m_d \) with displacement \( x_d(t) \) is connected to the damper base mass by stiffnesses \( c_d \) and damping \( d_d \) which represent the stiffness and damping of the leaf spring. Parallel to the leaf springs, the linear viscoelastic damping behavior is described by the spring-dashpot elements, indicated by the dashed line box, with parameters \((m_n \cdots m_n c_1 \cdots c_n)\).

These viscoelastic elements are also known as Maxwell elements [13]. The details of the modeling and design optimization are given in [21]. These elements enable to describe the damping behavior as function of frequency. The storage and loss moduli of a realistic fluid (Rocol Kilopoise 0868) are given in Fig. 2b [13]. The damper is of the shear plate type, as depicted in Fig. 1c [19] and enables to define geometrical damper parameters that make sense in practical RMD design: the system damping parameter in Ns/m is calculated by the total effective area divided by the gap width between the shear plates, in combination with the fluid’s storage and loss moduli. The ratio between total effective area and gap width is called geometrical damping factor (GDF) and is used in this paper as parameter to adjust the damping. A frequency response function (FRF) in z-direction (direction of the arrow in Fig. 1a) of the mechanical stage without dampers is shown in Fig. 3 by the dashed line for the input-output location indicated by the arrow in Fig. 1a. The solid curve presents the FRF of the stage with dampers added. The two discontinuous lines which connect the peaks between frequencies \( f_1 \) and \( f_2 \), of respectively the undamped and damped FRF’s, show the achieved amplitude reduction at the resonance frequencies and, therefore, represent the modal damping improvement. The damped curve is calculated by using optimization techniques, which determine the optimal stiffness and damping values to obtain maximal suppression over the frequency interval \( f_1 - f_2 \). This paper investigates how the linear viscoelastic fluid behavior as function of frequency influences the maximal achievable amount of resonance suppression.
3. Linear viscoelastic damping materials

With the linear viscoelastic behavior, see Fig. 2b, known [22] we are able to calculate the achievable modal damping of mechanical structures. In addition to linear viscoelastic fluids, linear viscoelastic solids, i.e. rubber-like materials, can be applied for this purpose. Linear viscoelastic behavior is captured with a multi-mode Maxwell model. Each mode, i.e. serial spring-damper combination has its characteristic frequency $f_c$. If $f \ll f_c$, the fluid acts like a damper, with 90 deg phase. For $f \gg f_c$, the fluid acts as a spring, which means that only the stiffness remains and that the phase has decreased to 0 deg. At the characteristic frequency $f_c$, the phase equals 45 deg.

The difference between linear viscoelastic fluids and solids is the low frequency behavior. An linear viscoelastic fluid shows viscous behavior for low frequencies, whereas an linear viscoelastic solid shows elastic behavior for these frequencies. This static stiffness appears in the multi-mode Maxwell model as a parallel spring. This mode at zero frequency is not present in a linear viscoelastic fluid model, which shows steady state viscous behavior.

Fig. 2. (a) Damper model with multi-mode Maxwell fluid model included. (b) Storage and loss modulus of a 3 mode Maxwell fluid model of Rocol Kilopoise 0868.

Fig. 3. Collocated frequency response function of the motion stage of Fig. 1. The result is a range of resonances that are suppressed by the dampers.
For a flexure based RMD design, the difference between fluids and solids result in a worse decoupling between stiffness and damping for linear viscoelastic solids: the stiffness of linear viscoelastic material contributes significantly to the RMD stiffness. This contribution is compensated by a decrease in flexure stiffness to maintain the optimal natural frequency. The second drawback of linear viscoelastic solids is that they usually contain modes with relative low characteristic frequencies with compared to fluids. These modes do not contribute to the performance within the frequency range in which the damping of the motion stage has to be increased. However, these modes contribute to the time response of the dampers.

To illustrate the two statements above, examples are given with model parameters for four different linear viscoelastic materials, three fluids and a solid rubber material. For the rubber and one fluid, parameter values are obtained from real materials. The parameter values for the other two (hypothetic) fluids are chosen to demonstrate specific behavior. The results for these four materials are compared to show their different influences with respect to motion stage damping. The four materials are:

- A linear viscoelastic fluid that shows viscous behavior over the frequency-range of interest, i.e. up to 10 kHz. A single mode model with a time constant of \(10^{-6}\) s and viscosity of 220 Pas.
- An LVE fluid with the characteristic frequency at 5 kHz. Again a single mode model is used with a time constant of \(2 \times 10^{-4}\) s, so the phase angle crosses 45 deg at 5 kHz, and a viscosity of 220 Pas.
- A 3-mode model that captures the behavior of the commercial lubricant fluid Rocol Kilopoise 0868. The zero shear viscosity of this fluid is 220 Pas.
- An industrial rubber type (GWM-A04), described with 10 Mx modes. This material shows LVE behavior in the frequency range 1–4 kHz, the range of interest for motion stage damping.

The relaxation times and viscosities for the different modes are given in Table 1. All fluids have a zero shear viscosity (ZSV) of 220 Pas. This makes the fluids more comparable. The magnitude and phase of these materials are presented in Fig. 4.

### Table 1
The characteristic values of the storage and loss moduli of the four materials are listed in this table.

<table>
<thead>
<tr>
<th>Viscoelastic material list</th>
<th>Mode 1</th>
<th>Mode 2</th>
<th>Mode 3</th>
<th>Mode 4</th>
<th>Mode 5</th>
<th>Mode 6</th>
<th>Mode 7</th>
<th>Mode 8</th>
<th>Mode 9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rocol 3-mode fluid model</td>
<td>(G) 1.55e4</td>
<td>5.87e6</td>
<td>4.41e5</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3 Maxwell modes</td>
<td>(\eta) 66.3</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Viscous fluid to 1kHz</td>
<td>(G) 1.38e9</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1 Maxwell mode</td>
<td>(\eta) 220</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Viscous fluid to 5kHz</td>
<td>(G) 6.9e6</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1 Maxwell mode</td>
<td>(\eta) 220</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rubber GWM A04</td>
<td>(G) 3.6e4</td>
<td>3.6e5</td>
<td>1.3e6</td>
<td>4.3e6</td>
<td>9.0e6</td>
<td>3.6e7</td>
<td>1.7e8</td>
<td>3.2e8</td>
<td>7.2e7</td>
</tr>
<tr>
<td>9 Maxwell modes</td>
<td>(\eta) 5.7e5</td>
<td>5.7e5</td>
<td>2.0e5</td>
<td>6.9e4</td>
<td>1.4e4</td>
<td>5.7e3</td>
<td>4.5e3</td>
<td>2.6e3</td>
<td>1.1e2</td>
</tr>
</tbody>
</table>

For a flexure based RMD design, the difference between fluids and solids result in a worse decoupling between stiffness and damping for linear viscoelastic solids: the stiffness of linear viscoelastic material contributes significantly to the RMD stiffness. This contribution is compensated by a decrease in flexure stiffness to maintain the optimal natural frequency. The second drawback of linear viscoelastic solids is that they usually contain modes with relative low characteristic frequencies with compared to fluids. These modes do not contribute to the performance within the frequency range in which the damping of the motion stage has to be increased. However, these modes contribute to the time response of the dampers.

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- An LVE fluid with the characteristic frequency at 5 kHz. Again a single mode model is used with a time constant of \(2 \times 10^{-4}\) s, so the phase angle crosses 45 deg at 5 kHz, and a viscosity of 220 Pas.
- A 3-mode model that captures the behavior of the commercial lubricant fluid Rocol Kilopoise 0868. The zero shear viscosity of this fluid is 220 Pas.
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The relaxation times and viscosities for the different modes are given in Table 1. All fluids have a zero shear viscosity (ZSV) of 220 Pas. This makes the fluids more comparable. The magnitude and phase of these materials are presented in Fig. 4.

### 4. Frequency dependent behavior

Here we show the differences between the four materials when applied in the complex stage design as presented in Fig. 1a. Optimal damper parameters – natural frequencies and geometrical damping factor are determined. The results are listed in Table 2. The effective mass of an RMD is 65 g and the stage mass amounts 22.9 kg. The damper location is the same for every case. A gradient-based optimization algorithm is used to converge towards optimal solutions [18].

Two different cases are studied to show the influence of the linear viscoelastic properties on the results that can be obtained:

- An optimization with a cost function that is defined for suppression between 1 and 2 kHz. This frequency range covers only the first resonance, and, therefore, only suppresses this one. This cost function is not visualized. The resulting parameter values represent typical TMD behavior.
- An optimization with a cost function that is defined for suppression between 1 and 4 kHz, which includes four resonance frequencies, see Fig. 3. This choice aims to obtain broad-banded and robust damping. The resulting parameter values represent typical RMD behavior.

The results of the small-banded optimization show that the suppression factors for the three fluids are comparable, approximately 40 dB. Looking top-down in Table 2, the TMD natural frequency is decreasing and the geometrical damping factor is increasing. Fluid 1, although it contains perfectly viscous damping up to 10 kHz, does not perform best. However, the difference is only 0.34 dB, which is meaningless for practical use. However, the effect is noticeable. Fig. 4 shows that the phase at 1500 Hz, the resonance frequency, is decreasing for the different fluids: fluid 1 = 90 deg, fluid 2 = 75 deg and fluid 3 (Rocol) shows about 60 deg. This decrease in damping is compensated by larger geometrical damping factors: respectively
The fluid stiffness due to the larger geometrical damping factors is compensated by a decreased optimal stiffness for the leaf springs, leading to lower natural frequencies, respectively, 1391 Hz, 1326 Hz and 1270 Hz. The rubber shows a lower suppression factor combined with a very low natural frequency. The geometrical damping factor (GDF) is lower than for the fluids. The reason is seen in the magnitude plot of Fig. 4, the values are an order larger than for the fluids. The optimal geometrical damping factor is, therefore, more than a magnitude smaller than for the fluids. The problem of this specific rubber material is the high stiffness-damping-ratio; it is too high for optimal resonance suppression. A larger geometrical damping factor introduces more damping, but, it increases the stiffness too and, therefore, the natural frequency. A rubber material with this phase characteristic and a lower zero frequency stiffness would have performed much better in this case.

Table 2
Suppression factors for the resonances for the small banded case (cost function between 1 and 2 kHz) and the broad banded case (cost function between 1 and 4 kHz). The damper mass is 65 g for all results listed. The stage mass is 22.9 kg.

<table>
<thead>
<tr>
<th>Small banded optimization</th>
<th>Suppr. [dB]</th>
<th>Nat. Fr. [Hz]</th>
<th>GDF [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid (45 deg @ 1e6 Hz)</td>
<td>– 39.41</td>
<td>1391</td>
<td>0.81</td>
</tr>
<tr>
<td>Fluid (45 deg @ 5e3 Hz)</td>
<td>– 39.75</td>
<td>1326</td>
<td>0.85</td>
</tr>
<tr>
<td>Rocol KP 3 mode model</td>
<td>– 39.62</td>
<td>1270</td>
<td>2.10</td>
</tr>
<tr>
<td>Rubber</td>
<td>– 30.96</td>
<td>10</td>
<td>0.054</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Broad banded optimization</th>
<th>Suppr. [dB]</th>
<th>Nat. Fr. [Hz]</th>
<th>GDF [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid (45 deg @ 1e6 Hz)</td>
<td>– 22.76</td>
<td>2136</td>
<td>4.87</td>
</tr>
<tr>
<td>Fluid (45 deg @ 5e3 Hz)</td>
<td>– 24.81</td>
<td>1437</td>
<td>5.54</td>
</tr>
<tr>
<td>Rocol KP - 3 mode model</td>
<td>– 23.66</td>
<td>1271</td>
<td>14.32</td>
</tr>
<tr>
<td>Rubber</td>
<td>– 24.02</td>
<td>0</td>
<td>0.144</td>
</tr>
</tbody>
</table>

0.81, 0.85 and 2.10 m.

The fluid stiffness due to the larger geometrical damping factors is compensated by a decreased optimal stiffness for the leaf springs, leading to lower natural frequencies, respectively, 1391 Hz, 1326 Hz and 1270 Hz. The rubber shows a lower suppression factor combined with a very low natural frequency. The geometrical damping factor (GDF) is lower than for the fluids. The reason is seen in the magnitude plot of Fig. 4, the values are an order larger than for the fluids. The optimal geometrical damping factor is, therefore, more than a magnitude smaller than for the fluids. The problem of this specific rubber material is the high stiffness-damping-ratio; it is too high for optimal resonance suppression. A larger geometrical damping factor introduces more damping, but, it increases the stiffness too and, therefore, the natural frequency. A rubber material with this phase characteristic and a lower zero frequency stiffness would have performed much better in this case.
The comparable suppression factors in case of the broad banded optimization results show that all four materials perform well within this frequency range (1–4 kHz). However, small differences are present. The first fluid obtains the worst suppression value, although it shows a phase of 90 deg over the whole cost function frequency range, see Fig. 5. This is similar to the small banded suppression results. The other two fluids are performing better. Looking top-down in Table 2, the geometrical damping factor increases for the fluids, and the natural frequency decreases, as for the small banded case. The fluid with 45 deg phase at 5 kHz performs best again. In this case, also the rubber performs well in terms of suppression factor. However, due to the relatively high rubber material stiffness, a significant amount of stiffness is added in the direction of motion by the rubber material. This is compensated for by the optimization algorithm by providing a very low natural frequency of the flexures. This matches the stiffness and damping values in the direction of motion, however, it will be difficult to obtain high natural frequencies for the other directions. The problem in practice will be that these resonances will have cross-talk in all directions. Therefore, a split of stiffness (leaf springs) and damping (viscoelastic material) will improve the performance of the overall system.

5. Suppression as function of damper mass

Fig. 5 shows the maximal suppression factor, the natural frequency, and the geometrical damping factor as function of the damper mass for the broad banded case. These plots are the result of 50 optimizations for each linear viscoelastic material. The right figures are details of the left figures, the result for a damper mass of 65 g is indicated by the dashed line. The upper left figure shows straight lines as function of the damper mass. The lines do not intersect which clearly shows that the maximum obtainable suppression factor for a certain RMD mass and damping range solely depends on the linear viscoelastic fluid characteristics: the fluid that performs best with a certain damper mass is the best fluid for other damper masses too. These graphs show that (a) approximately pure viscous damping (Fluid 45 deg @ 1e6 Hz) does not provide optimal suppression. This holds for both the small banded case and the broad banded case, see Table 2, and (b) the broader the frequency range for the cost function is specified, the larger the advantage of linear viscoelastic behavior with respect to viscous behavior will be. The mechanism behind these results is explained in the next section.
6. Frequency dependent natural frequency

To explain the advantage of a linear viscoelastic material with respect to a purely viscous material, the RMD model of Fig. 2a is used. The corresponding values are given in Table 2, i.e. the broad banded optimization with Rocol Kilopoise. The damper mass of 65 g combined with the leaf spring stiffness results in an undamped natural frequency of 1270 Hz, with a theoretical stiffness of 4.13e6 N/m. This value is constant as function of frequency and shown in Fig. 6 as Flexure stiffness. The complex fluid damper response shows an in-phase component. This is due to higher order dynamic behavior of the multi-mode Maxwell fluid model and can be seen as a ‘frequency dependent stiffness’: each Maxwell mode contributes to the stiffness in a specific frequency range. This frequency dependent stiffness is shown in the Fig. 6 as fluid induced damper stiffness. The combined stiffness as function of frequency is the sum of the flexure and the fluid damper stiffness. This frequency dependent stiffness enables to calculate a frequency dependent natural frequency of the RMD. Fig. 7 shows the RMD natural frequency–frequency ratio for the flexure stiffness and the combined stiffness. The natural frequency for the flexure stiffness is 1270 Hz and is shown by the crossing of the curves with the natural frequency coefficient equal to 1.

The natural frequency for the combined stiffness is approximately 2100 Hz. The frequency range in which the modal stage damping has to be increased is indicated by the thicker line. A more flat slope contributes to the effectiveness of the damper. This explains why the small banded optimization is, in a relative sense, less advantageous: the change in natural frequency is smaller over the frequency range of interest, therefore, compared to the viscous fluid a smaller improvement is possible than in case of the broad banded resonance damping. For comparison: Table 2, broad banded optimization, Fluid (45 deg @ 1e6 Hz) shows a natural frequency of 2136 Hz for a fully viscous fluid. This natural frequency is roughly in the middle of the cost function range. This already indicates that this fluid does not benefit from linear viscoelastic stiffness (see the lower suppression factor).

![Diagram](image1.png)

**Fig. 6.** Stiffness as function of the frequency. The flexure stiffness is constant.

![Diagram](image2.png)

**Fig. 7.** Natural frequency coefficient as function of frequency. The circles indicate the static natural frequency and the actual natural frequency.
7. Time domain results

Using the parameters for four TMDs with different linear viscoelastic materials, the step responses are calculated to illustrate another important difference between linear viscoelastic solids and fluids. Fig. 8 shows normalized step responses for the four RMDs. The upper figure gives the response on a linear time scale and the lower one on a logarithmic time scale. The damper with the viscous fluid (45 deg at 1 MHz) shows the quickest response. This mechanism has the highest stiffness and the fluid has short time constants. The RMD with optimal fluid for suppression (45 deg at 5 kHz) also shows a fast response. The RMD with the Rocol fluid shows a slower response than the first two. The RMD with rubber applied shows the slowest response. The initial response is relatively good. However, the low-frequency linear viscoelastic modes cause a step response with a very slow component. Finally, the step response converges to 1 as in case of the fluids. Note that the performance of these dampers on the motion stage is comparable in the frequency range between 1 and 4 kHz.

8. Conclusions

Although counterintuitive, linear viscoelastic damping behavior results in larger resonance suppression factors, than purely viscous materials if applied in mass-based resonance dampers. Therefore, this result holds for RMD’s, applied to suppress resonances over a specified frequency range as discussed in this paper.

A linear viscoelastic fluid loses damping at high frequencies, which induces difficulties to design high damping values between two bodies at these frequencies. The effectiveness of the RMD’s, however, is, beside the amount of damping, also based on their natural frequency, which is composed of the RMD’s connection stiffness and the effective mass. It appears that, although the damping vanishes for high frequencies, i.e. the loss modulus decreases, the stiffness increases, i.e. the storage modulus increases as function of frequency. This means that the in-phase behavior of the complex fluid acts as a frequency dependent stiffness, which matches the RMD frequency in a better way to the natural frequencies of the mechanical structure than a viscous fluid does. This improves the effectiveness of the RMD’s over the intended frequency range. The improvement factor found in this paper is approximately 2 dB, which means a factor 1.25. Although this seems to be a relatively small factor, in motion controller design it can be used very well, because it improves the robustness of the closed loop system.

The amount of resonance suppression by applying a linear viscoelastic material instead of a purely viscous fluid is solely determined by

- the linear viscoelastic material properties, and
- the frequency range over which the RMDs have to be effective.
The improvement factor is independent of the damper mass which was visualized in Fig. 5. The advantage as described in this paper applies to both TMDs and RMDs but is more advantageous in case of broad banded damper design. The larger the frequency range is defined, the more effective this phenomenon can be utilized.

Although rubber materials work as well as damping material for high frequency resonance damping, they contain Maxwell modes with relatively long time constants and even static stiffness properties due to the characteristics of a solid material. The modes with long time constants are visible in the transient response of the RMDs, which lead to a long relaxation time of the damper after a certain displacement.

Acknowledgments

This project is supported by ASML Research Mechatronics (NL). We greatly appreciate the input of dr. ir. J.P.M.B. Vermeulen, dr. ir. M.M.J. van de Wal, and dr. ir. S.H. van der Meulen.

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