Project Review: Elastic Decoupling of a large Roller Grinder

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ABSTRACT
In this conference-contribution an insight-view of a project for elastic decoupling is given. The project is about the elastic decoupling of a roller grinder. The roller grinder is situated in a production hall; so the machine is exposed to high levels of shock and vibrations caused by the surrounding production facilities. To ensure the accuracy of this high-precision-machine over the total service life, a vibration protection with polyurethane-bearings was requested. Studies were performed and a solution was suggested. This solution was approved by the customer and the installation was done in April 2015.

Keywords: vibration isolation, elastic decoupling, Polyurethane

I-INCE Classification of Subjects Number(s): 14.1.6 and 46.2

1. INTRODUCTION

During operation of machines, carrying out forklift-work, service of gantry cranes etc. dynamic forces inevitably occur. These dynamic effects cause noise and vibrations and can have negative effects on the manufacturing processes. They can be revealed in a reduced accuracy of the high-precision production-process, can lead to an increased mechanical wear or can reduce the lifetime of machine-components.

Therefore elastic decoupling as a efficient vibration isolation can be recommended. This is a commonly used measure. In doing so the machine is separated completely from the surrounding with elastic material. Proper elements for this purpose should feature elastic and damping characteristics.[1] In the following the theoretical background of vibration isolation, polyurethane (PUR) as a vibration isolation material itself as well as a realized project is explained.

2. ELASTIC DECOUPLING WITH POLYURETHANE

2.1 Mechanism of Action
Vibration-isolating effect of an elastic bearing bases on the physical principle of a compensation of mass-forces (so-called “harmonic oscillator” – see figure 1). Dynamic forces of the machine are shown with force \( F(t) \), machine and foundation are represented with mass \( m \); elastic decoupling is shown with a spring-damping-unit (stiffness \( k \) and damping factor \( \eta \)).

Generally a stiff subsoil and rigid bodies are assumed (no relative movement within the oscillating body). Furthermore only the steady-state condition caused by harmonic excitation is considered.

Exemplary for the vertical axis the natural frequency \( f_0 \) as one of the characteristic parameters of the system can be calculated with mass \( m \) and dynamic stiffness \( k' \) with equation 1.

\[
f_0 = \frac{1}{2\pi} \sqrt{\frac{k'}{m}}
\]  

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The transmission function $L$ (amplitude-frequency response or more precise “amplitude transfer function of the foundation force during force excitation”) expresses the ratio of maximum amplitude to static amplitude. For a harmonic oscillator it can be calculated with natural frequency $f_0$ and damping ratio $\eta$ as follows (equation 2):

$$L(f) = 20 \cdot \log \left[ \sqrt{\frac{1 + \eta^2 \cdot \left( \frac{f}{f_0} \right)^2}{\left(1 - \left( \frac{f}{f_0} \right)^2 \right)^2 + \eta^2 \cdot \left( \frac{f}{f_0} \right)^2}} \right]$$

(2)

Transmission function (equation 2) displayed in dependency of the frequency and standardized to the natural frequency results in figure 2 (logarithmic scale).

In this transmission function (figure 2) different areas can be obtained:

- amplification where $\frac{f}{f_0} < \sqrt{2}$
- resonance where $\frac{f}{f_0} = 1$
- no change where $\frac{f}{f_0} = \sqrt{2}$
- reduction (isolation) where $\frac{f}{f_0} > \sqrt{2}$

When regarding the elastic bedding of more complex structures not only the vertical direction but also the other directions including rotational degrees of freedom have to be considered. See figure 3 therefore.[2] In addition resonance effects on elastic parts of the structure have to be taken into account to avoid high amplifications of the induced vibrations.
2.2 Dimensioning and Design

First of all the elastic material has to withstand the occurring compression stress caused by the static loads (see section 2.3.2 “loading capacity”). In the second step the efficiency of vibration isolation has to be evaluated. As shown in the previous section the natural frequency is a relevant parameter and can be calculated with mass and stiffness. Thus a statement to vibration isolation (“degree of isolation”) can be calculated as a function of the transmission function (equation 2).

Generally elastic decoupling can be realized with or without an additional inertia mass on top of the vibration isolation (see figure 4). A solid inertia mass is preferred because it leads to a uniform pressure distribution and the structural vibrations of the foundation are in a higher range. Vibration amplitudes can be reduced (system is “calmed”). Recommendations for the mass-ratios of machine to foundation are available. [3]

Figure 3. Rigid-body-model (six degrees of freedom).

Figure 4. Design with and without additional inertia mass.

Figure 5. Discrete and full-surface bearings below a foundation.

Furthermore elastic bearings can be implemented discretely (point-bearings) or as a full-surface application – see figure 5. Full-surface-bearing applications offer advantages during construction as the elastic bearings can be used as lost formwork. Discrete bearings are more elaborate at the construction sites (use of pre-fabricated-plates). In exchange the dimensions of the discrete bearings can be customized according to
the occurring loads so that the bearings can be utilized optimally. Therewith very low natural frequencies can be achieved (see section 2.3.3).

2.3 Polyurethane Material (PUR)

2.3.1 Chemistry

For producing spacious PUR a reaction mixture out of isocyanate and polyol is applied on a continuously moving carrier tape. Caused by chemical crosslinking with additional foaming agents the mixture foams up and PUR develops. Cell-structure, density and other parameters can be regulated precisely within the process. Therefore also the mechanical properties like stiffness, loading-capacity and damping can be specified individually.

2.3.2 Loading capacity

Materials for vibration isolation have to withstand the admissible permanent loads during the entire service live without any damage and without immoderate increase of the deflection (see section 2.2). It is important that the material is utilized optimally (see section 2.3.3).

By adapting the chemical recipe the loading-capacity of PUR can be adjusted precisely. Depending on the density PUR has a permissible compression stress from $0.01 \text{ N/mm}^2$ to $6 \text{ N/mm}^2$. The term “static load limit” is used herefore (see datasheets [4]).

2.3.3 Stiffness

Dynamic stiffness of the elastic bearing together with the oscillating mass determines the natural frequency of a system (thus determines the efficiency of the vibration isolation - see section 2.1).

A low Young’s Modulus leads to a low tuning frequency thus to an efficient vibration isolation. Therefore the material has to be used at its optimum range of use (see the nonlinear behaviour of the Young’s Modulus in figure 6).

![Figure 6. Young’s Modulus of PUR Sylomer® SR110.](image)

Young’s Modulus (progression and absolute value) depends on the chemical recipe and density: e.g. for PUR with a low density the value for the dynamic (10Hz) Young’s Modulus is in a range of $0.15 \text{ N/mm}^2$ to $0.6 \text{ N/mm}^2$ and for PUR with a very high density is in between $10 \text{ N/mm}^2$ to $12 \text{ N/mm}^2$. 

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Materials with a lower Young’s Modulus do have a lower loading capacity (see section 2.3.2). If a higher-density-material with a higher Young’s Modulus is used a low natural frequency can be achieved with an increased thickness of the pad (stiffness is like reciprocal to the thickness of the pad).

Another approach for the explanation of the optimal “load point” works over the load-deflection-diagram (see figure 7). The tangent of the load-deflection-diagram has a low gradient in the optimal load-point. That means the material reacts “softly” at this pressure and provides a low natural frequency.

So the non-linear behaviour of PUR is excellent for a vibration isolation.

Figure 7. Load-deflection-diagram of PUR Sylomer® SR110.

2.3.4 Damping

Damping can be characterized with the so-called mechanical loss factor (dimensionless). The value depends on the chemical recipe and ranges for mixed-cell PUR from 0.11 to 0.25 and for closed-cell PUR from 0.07 to 0.10. Damping has effect on the behaviour of the transmission-function (amplification at resonance frequency and gradient of the diagram in isolation-area; see figure 2).

Adequate damping is essential for a correct design of elastic bearings. During run-up and run-down of the machine the excitation frequencies may pass the natural frequency of the system (resonance). Depending on the speed of this “passing” the damping has to be adjusted in a way that the vibration amplitudes do not exceed the given limits.

3. PROJECT ROLLER GRINDER

3.1 Situation

Valmet is a leading supplier of technology for pulp and paper industry. At the plant in Cernay (France) calender rolls as essential components for the production-lines are built. An excellent geometric precision of these rolls (high-precision accuracy in terms of form and shape) is essential to achieve high quality in the production process of paper. Especially for modern paper production this is becoming increasingly important.

For the finishing-process of these rolls a roller-grinder is used. As the capacity of machinery in Cernay was reaching the limits a new roller-grinder had to be installed. Therewith rolls up to 2.5 m of diameter and up to 12 m of length with maximum weight of 90 t can be processed. This roller-grinder fullfills the required accuracy of ±3 μm on the whole length of the rolls.
Machine loads:

<table>
<thead>
<tr>
<th>Component</th>
<th>Weight:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motor</td>
<td>5.0 t</td>
</tr>
<tr>
<td>Headstock (gear box)</td>
<td>7.5 t</td>
</tr>
<tr>
<td>Bed of workpiece A and B</td>
<td>16.0 t</td>
</tr>
<tr>
<td>Bearing pedestals</td>
<td>5.0 t</td>
</tr>
<tr>
<td>Bed of carriage</td>
<td>43.0 t</td>
</tr>
<tr>
<td>Carriage</td>
<td>6.5 t</td>
</tr>
<tr>
<td>Workpiece</td>
<td>90.0 t</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>173.0 t</td>
</tr>
</tbody>
</table>

The required foundation has dimensions 17 m x 4.60 m with a depth of 2.50 m which results in a foundation-weight of approximately 490 t. The foundation block was carried out as a deep-seated foundation in a concrete pit.

### 3.2 Solution – Calculation Approach

The mass-ratio of \(\frac{490}{173} = 2.9\) is a reasonable value for this type of machine. In this way the system has enough inertia to keep the vibration amplitudes at low values. Based on experience the customer demands a fundamental vertical frequency of 3 Hz at first. Therefore steel-springs would have been chosen. As this would have lead to additional constructional complexity (accessibility to the spring unit for maintenance) alternative options were discussed. Finally the requirement was restated to a vertical natural frequency of <7 Hz which is achievable with PUR-bearings. Therefore 66 discrete PUR pads of type Sylodyn® NE in thickness 75 mm and base dimensions 390 mm x 390 mm were chosen and a vertical natural frequency of 6.0 Hz was achieved. The damping-behaviour of these pads is described with a loss factor of 0.09 (corresponds to Lehr damping factor of 0.045).

The change of the center of gravity due to operation conditions is not relevant since the foundation block is significantly heavier than the moving parts.

As a calculation model the rigid-body-model with six degrees of freedom was used. See section 2.1 respectively [2]. The mass-matrix refers to the total center of gravity and is oriented in the direction of the inertia principal axis. For the total stiffness-matrix the stiffness-matrix of each bearing is accumulated and then transformed so that they refer to the center of gravity and are oriented to the inertia principal axis as well. So the transformation-matrices include translations and rotations. Based on the dynamic stiffness-matrix and the mass-matrix of the overall system the eigenfrequencies were calculated and the vertical natural frequency can be identified.

An evaluation of the vibration amplitudes was done by an external consultant who was involved in this project.

### 3.3 Solution – Constructional Approach

Natural frequency of <7 Hz is achievable with PUR-bearings, but in general not with a full-surface approach. For this reason a solution with discrete bearings was necessary. The discrete bearings were arranged according to an installation plan (figure 8). Surface has to be smooth and plane (see installation instructions [4]). The dimensions of the discrete bearings can be choosen in order to make the optimum use of the bearing material (capacity reaches 100 %; thus the materials are used in the range of the lowest dynamic stiffness).

Discrete bearings require another constructional approach. The elastomer cannot be used as lost formwork anymore - an additional shuttering has to be provided which has to resist against the actions during the concrete works of the foundation block. In this case the customer realized that with steel-plates of 25 mm thickness. The steel-plates were arranged on top of the elastomer-bearings and then welded together at their joints. See figure 10.
The sides of the foundation-pit have to be decoupled as well. Theoretically an air-gap shows the best performance, but is hard to realize. So at first a very soft PUR-product was suggested but due to the big lateral area there was a considerable impact on the overall dynamic stiffness and thus to the fundamental vertical frequency. As an alternative a special cardboard-product (product “Alvaplaque” with thickness 60 mm) was...
chosen. During the concreting of the foundation block the cardboard shows sufficient rigidity to withstand the concrete pressure. After the maturation of the concrete block the cardboard was watered which led to a complete loss of the (already-low) stiffness over a short time period. So this is an interesting approach to create this solution with a perimetal air gap. See figure 11.

3.4 Fact box

<table>
<thead>
<tr>
<th>Machine:</th>
<th>roller grinder</th>
</tr>
</thead>
<tbody>
<tr>
<td>Machine weight:</td>
<td>173 t</td>
</tr>
<tr>
<td>Foundation:</td>
<td>17 m x 4.6 m with depth 2.5 m</td>
</tr>
<tr>
<td>Foundation weight:</td>
<td>490 t</td>
</tr>
<tr>
<td>Vibration isolation:</td>
<td>discrete polyurethane-bearing (type Sylodyn® NE in thickness 75mm)</td>
</tr>
<tr>
<td>Commissioned:</td>
<td>September 2015</td>
</tr>
</tbody>
</table>

3.5 Conclusion

A strong cooperation between the plant-operator, the material-provider and the engineering consultant is necessary for an effective vibration isolation solution (see also [5]). Plant-operator defines his requirements with the technical constraints, material-provider provides material and specific know-how and planning-consultant performs the analysis and the verification of the required efficiency. This way of collaborative working has been successfully implemented in this project.

Furthermore the awareness of the workers on site is an important key factor for the correct installation. Workers have to be skilled and have to be instructed properly.

REFERENCES


