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Modelling of an Innovative Technology for Pavement Milling

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Abstract: A new, dynamical approach for increasing the effectiveness of the pavement milling process is shown in this paper. The technology is derived from a dynamical model of a two-mass oscillator and a simple model of the pavement material. Both models are combined to simulate the milling operation. Different excitation methods are used and combined to find the optimal operation point of the model by taking the effectiveness into account. The simulation results are validated on a test stand with suitable proximity settings. With the model being validated, it can be scaled to a more complex model that also takes the geometry into account. The benefit of a more effective milling process results for example in a reduction of the machine size with maintaining the same work output or in less fuel consumption of existing machines.

Keywords: Pavement Milling; Dynamical Non-linear Modelling; Material Modelling; Effectiveness; Excitation Models; Oscillator; Industry Application; Signal Processing; Model Scaling

1. INTRODUCTION

The state of the art process in pavement milling nowadays is the cold milling method. It was introduced after the hot milling process, which needed a preheating of the pavement surface for easier removal of the material. Due to the poor overall energy effectiveness of the whole process (heating of the pavement!), it was not sustainable. New forms of rotary grinders and drill bits were developed to increase the cutting performance of the tools. Soon the tools were powerful enough to remove the pavement without the need to preheat the surface. This optimization of the drill bits with respect to wear and the reduction of the cutting forces has made a giant leap in pavement milling and mining applications. Modern cold milling machines come in all different sizes. They are available from small surface milling machines, which weigh 4 tons, to 40 ton machines which are designed for the complete removal of highway pavements. Despite all the progress that has been made in this field, the core principle has always stayed the same. All machines use a rotary grinder that is statically mounted to the chassis of the machine. The more material that has to be milled, the heavier the machines that are designed and the more fuel they consume.

The new technology in this paper can enable a new design generation of milling machines that has a higher performance to weight ratio. The key element for achieving this goal lies in the potential energy of dynamic vibrations. The paper aims to show a possibility of how to apply vibrational techniques to a two mass oscillator in a way that the mean forces between the masses can be reduced while maintaining the same work output. E.g. milling with the same speed while the mean occurring forces decrease. Lower mean forces per definition imply lower work and therefore less energy consumption.

2. DESCRIPTION OF THE DYNAMICAL MILLING PROCESS

The starting point of the investigation is the modelling of the dynamic process. The milling situation is simplified. Instead of a rotating drill bit on a rotary grinder, a translational drill bit that is attached with a spring and damper system to a chassis is modelled. The chassis acts as the heavy weight that is being driven by a motor. The spring and damper system between the chassis and the drill bit represent the flexibility in the system and the drill bit acts as the cutting tool while neglecting its geometry.

On the right side of figure 1 there is the pavement material model. The overall behaviour of the material is modelled with a massless spring and damper system that has the special ability of being able to break when the reaction force reaches a defined limit. When this happens the material model is moved forward by a prefixed increment.

The combination of these two parts represent the simple model of the milling process. A schematic of the model is shown in fig. 1.

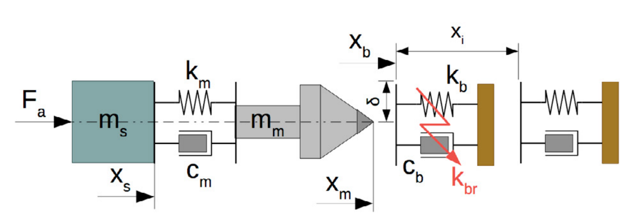

Fig. 1. Model scheme of the milling process

The variables and parameters used in this model can be seen in table 1.

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Table 1. Model variables and parameters

Name	Unit	Description
x_s	$[m]$	Position of the chassis
x_m	$[m]$	Position of the drill bit
x_b	$[m]$	Position of the material element
x_i	$[m]$	Increment of the material or chip size
δ	$[m]$	Single cutting depth
m_s	$[kg]$	Mass of the chassis
m_m	$[kg]$	Mass of the drill bit
k_m	$[N/m]$	Spring constant of the drill bit
c_m	$[Ns/m]$	Damping constant of the drill bit
k_b	$[N/m]$	Spring constant of the material
c_b	$[Ns/m]$	Damping constant of the material
k_{br}	$[N/m]$	Breaking constant of the material
F_a	$[N]$	Driving force of the chassis
v_c	$[m/s]$	Set point for cutting speed

The two mass oscillator is made of the chassis with position x_s and mass m_s and a single drill bit with position x_m and mass m_m. The two masses are connected with a spring with constant k_m and a damper with constant c_m. To move the chassis forward, a force F_a is applied to it. This force is regulated by a PI Controller that keeps the chassis at a designated cutting speed v_c. The differential equation for the chassis is in eq. (1) and for the drill bit in eq. (2). Similar dynamical equations are also used in Balachandran (2001), Balachandran and Zhao (2000), Zhao and Balachandran (2001), Araujo et al. (2009), Araujo et al. (2010), Sutherland and DeVor (1988) and Altintaş and Budak (1995).

$$m_s \cdot \ddot{x}_s = F_a - F_m \qquad (1)$$

The driving force for the chassis is the applied force F_a (eq. (3)) from the controller and the opposing force F_m comes from the drill bit (eq. (4)).

$$m_m \cdot \ddot{x}_m = F_m - F_b \qquad (2)$$

Equation 2 is the one for the drill bit. Its driving force is the drill bit force F_m (eq. (4)) and the opposing force is the ground force F_b (eq. (5)).

The applied force F_a from the PI Controller is modelled as in eq. (3).

$$F_a = K_p \cdot (v_c - \dot{x}_s) + K_i \cdot \int (v_c - \dot{x}_s)dt \qquad (3)$$

The force F_m between the two masses is modelled as in eq. (4).

$$F_m = c_m \cdot (\dot{x}_s - \dot{x}_m) + k_m \cdot (x_s - x_m) \qquad (4)$$

To describe the ground reaction force F_b a non-linearity has to be introduced. The ground force can only occur while there is contact. Additionally, the contact force can not become negative. Therefore eq. (5):

$$F_b = \begin{cases} c_b \cdot \dot{x}_m + k_b \cdot (x_m - x_b), & \text{if } x_m > x_b \text{ and eq.} > 0 \\ 0, & \text{else} \end{cases} \qquad (5)$$

The modelling of the non-linear contact is the same as a jumping ball in Nollau (2009).

Next to the two parameters k_b and c_b, the behaviour of the material is described with three additional parameters. These are the breaking constant k_{br}, the cutting depth δ and the material increment x_i. To allow the material to break it is necessary to define a maximum stress force of the material. The maximum material force $F_{b,max}$ is assumed to be proportional to the cutting depth δ as in Araujo et al. (2009) and Araujo et al. (2010) or Vlasblom (2007). It follows:

$$F_{b,max} = k_{br} \cdot \delta \qquad (6)$$

The maximum material force therefore depends on the cutting depth. To keep the model simple the cutting depth is assumed to be constant over time. When the ground force F_b reaches the maximum force $F_{b,max}$ the material element moves by the increment x_i. This is described in eq. (7). Another approach is to take the cutting kinematics into account as in section 5.

$$x_b = \begin{cases} x_b + x_i, & \text{if } F_b > k_{br} \cdot \delta \\ x_b, & \text{else} \end{cases} \qquad (7)$$

The breaking constant k_{br} is not to be confused with the spring constant k_b. The difference can be seen in the stress-strain diagram for brittle materials (compare fig. 2). Vlasblom (2007) also uses brittle failure for the modelling of rock.

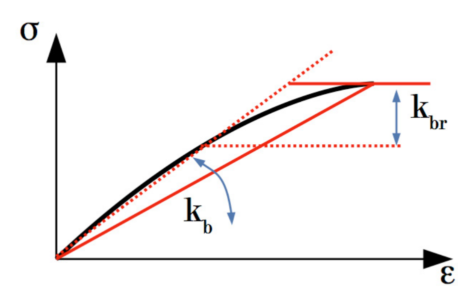

Fig. 2. Stress-strain diagram for the modelling of brittle material with two coefficients k_{br} and k_b

The spring constant k_b describes how fast the breaking force $F_{b,max}$ is reached and the breaking constant k_{br} describes the limit of F_b or in other words, the value of $F_{b,max}$

There are two non-linearities which have different behaviours. The first non-linearity comes from the contact condition and the second one from the breaking of the material.

The described system can now be excited with different methods.

3. EXCITATION OF THE MODEL

The excitation of the model influences the systems behaviour. Since it contains two non-linearities, the systems response is not linear and can drift into chaos. To investigate the systems behaviour, two different excitations are being tested. These are:

- Normal excitation and
- Waypoint excitation

To be able to compare the different simulation results, it is necessary to introduce an effort coefficient for each parameter setting. The effort coefficient η is calculated