

Investigation of the Influence of ODE Based Friction Models on Complex FEM Brake Models in the Frequency Domain

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ABSTRACT

In today's research and development of brake systems the model-based prediction of complex vibrations and NVH phenomena plays an important role. Despite the efforts, the high dimensional computational simulation models only provide a limited part of the results gained through experimental measurements. Several reasons are discussed by the industry and academic research.

One potential source of these inadequacies is the very simple formulation of the friction forces in the simulation models. Due to a significant shorter computation time (by orders of magnitude), the complex eigenvalue analysis has been established, in comparison to the transient analysis, as the standard method in the case of industrial research, where systems with more than one million degrees of freedom are simulated. The coefficient of friction in the models is usually assumed to be constant, although it is known that the coefficient of friction is dependent on various system variables like temperature, pressure and velocity. Furthermore, well-known friction phenomena such as time lag behaviour, hysteresis, and the fading effect, can be observed in experimental measurements. For this purpose, new and complex friction laws can be found in the literature, making use of additional parameters in differential formulations. To use such friction laws in linear systems, the initial system must be embedded in an expanded state space. For this purpose, the state vector is extended by the new state variables like coefficient of friction or friction force, pad and disc temperature, wear and so on. This method will be referred to in this paper as the "Method of the Augmented Dimensioning" (MAD).

The method was used to perform principle investigations on the influence of various friction models onto the stability behaviour of friction-induced vibrations in the case of minimal models. This paper analyses embedded ODE based friction models in common FEM-tools, in order to describe the influence of dynamic friction laws on the stability in commercial FEM brake models of high order.

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INTRODUCTION

In today's development of automotive brake systems two general simulation methods for stability analysis are used: Integration in the time domain and complex eigenvalue analysis. Due to significantly shorter computation time (by orders of magnitude), the complex eigenvalue analysis has been established as the standard method in the case of industrial research, where systems with more than one million of degrees of freedom are simulated.

Despite much effort, the high dimensional computational simulation models only provide a limited part of the results gained through experimental measurements. It is known that the number of instabilities of computational simulation models is greater than that of experimental measurements [1].

The coefficient of friction is a system variable with remarkably dynamic properties. Recent measurements have demonstrated that the coefficient of friction is dependent on various system variables such as temperature, pressure and velocity [2, 3, 4, 5]. Although the contact between the brake pad and disc is one of the most essential

components of the physical system considered, the friction contact is usually only rudimentarily formulated in NVH simulations, for example, using the Coulomb friction model. An implementation of appropriate friction models can greatly enhance the quality of the computational simulation models.

Generally, modelling the friction boundary layer is commonly based on either continuous macroscopic or discrete approaches. The macroscopic approaches are characterized by an integral description of the friction coefficients with regard to the contact surface, with the aid of one or several algebraic and/or differential equations. For example, with a set of two first-order differential equations, including the friction value and patch temperature as state variables, the friction model from Ostermeyer [6] can reproduce unsteady, temperature dependent friction force characteristics. Well known friction phenomena such as time lag behaviour, hysteresis, and the fading effect, can be simulated with this friction model [7]. For a detailed explanation of the patch theory, see [6] and [7].

To use friction models in the frequency domain which are described by differential equations, the initial system must be embedded in an expanded space. This common method has been used to approximate nonlinear friction effects in a linear space and will be referred to in this paper as the “Method of the Augmented Dimensioning” (MAD). The method was used in the EKS “Expert Group Simulation Brake Noise” to investigate the behaviour of dynamic friction models in the frequency domain for various established minimal models. The MAD and exemplary results of the stability behaviour based on the expanded minimal model of Hamabe by Hoffmann and Gaul [8] was published recently [9]. One further aim of the project within the EKS is the implementation of the MAD using the two commercial FEM-tools ABAQUS and PERMAS. The friction model from Ostermeyer [7] has been selected for implementation in commercial FEM-tools for the following reasons: it represents unsteady temperature-dependent friction characteristics, and it includes delayed adjustment processes for the friction coefficient and temperature using two time constants [9]. The current implementation will be the focus of the paper.

This paper is structured as follows: First, the main aspects of the MAD will be discussed based on a node-to-surface approach in the case of a penalty-based contact formulation. Afterwards, the initially intended workflow for the implementation of the MAD will be explained. The corresponding workflow will be presented for the two FEM-tools ABAQUS (6.14-1) and PERMAS (v15). The challenges arose in the course of the implementation will be discussed. Furthermore, the solutions and their implementations will be discussed. In addition to the implementations, examples will be presented which have been used (among other things) for the validations of the implementations. At the end of this paper, the content will be summarized and further steps will be introduced.

THE METHOD OF AUGMENTED DIMENSIONING BASED ON A NODE-TO-SURFACE APPROACH

In this section, the most important aspect of the MAD will be discussed by considering selected FEM contact formulations. In the case of the MAD, the initial system must be embedded in an expanded state space. For this purpose, the state vector is extended by the new state variables such as the coefficient of friction or friction force, pad and disc temperature, wear and so on. Therefore, the system matrices must be extended by the new entries of the considered dynamic friction model. This coupling method is often called “strong coupling”, in contrast to “weak coupling”. Further fundamental details and the influence onto the stability behaviour in the case of minimal models were published in [9].

In the following, the MAD will be discussed using a node-to-surface approach and a penalty-based contact formulation in the case of the normal and the tangential directions. For this purpose, the contact between the slave nodes and the master surface will first be considered in detail (see Figure 1). An exemplary master segment of the disc is illustrated in black (see Figure 1, bottom). In this case, four master nodes define the illustrated master segment. The local clockwise Cartesian contact coordinate system is defined with respect

to the master segment in the case of a contact, where n represents the normal direction and (γ_1, γ_2) represent the two tangential directions. An associated slave node could come into contact with the master segment. This contact (between a slave node and the master segment) causes a certain penetration $h = n_s - n_M$.

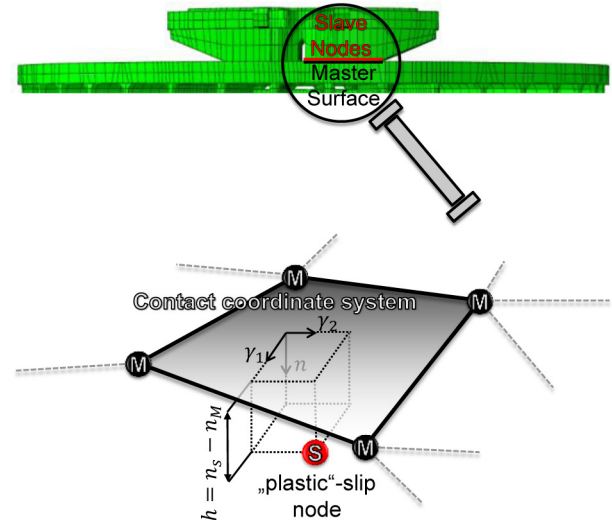


Figure 1. Contact element using a node-to-surface approach and a penalty-based contact formulation.

Only the slave nodes with a (plastic) slip state are used for the extension of the system matrices (see Figure 1, bottom and Figure 2, top). Without going into further details, it is important to mention that the rest position of the dynamic friction model considered is used to determine the critical shear stress in the nonlinear static analysis (see the following sections for more details). Therefore, the nonlinear static analysis can be performed on the basis of the dynamic friction model.

Considering such a contact, as illustrated in Figure 1 (bottom), the resulting relative velocity can be defined for the local coordinate system, as seen in Figure 2 (bottom). As is well known, the resulting tangential stress acts opposing the relative velocity. The relative velocity based on the local coordinate system is defined by the velocity components of the slave nodes, the master segment, and the motion contributions. The motion contribution represents a predefined vector field and is defined in the nonlinear static analysis step (see the following section).

Both the penetration h and the relative velocity v (see Figure 3), based on the local coordinate system, are used to define the reactive power $r(t)$ within the considered dynamic friction model. The reactive power is used for the coupling between the mechanical brake system and the friction model. This power can be determined from the product between the normal contact stiffness k , the penetration h and the relative velocity v .

Based on this definition of the dynamic friction model in the local coordinate system, the linearization takes place. The extension of the system matrices (see red terms in Figure 3, bottom) takes place automatically for each sliding contact node, the corresponding workflow will be explained in the next sections. The resulting gradients of the linearization will be discussed briefly in the following.

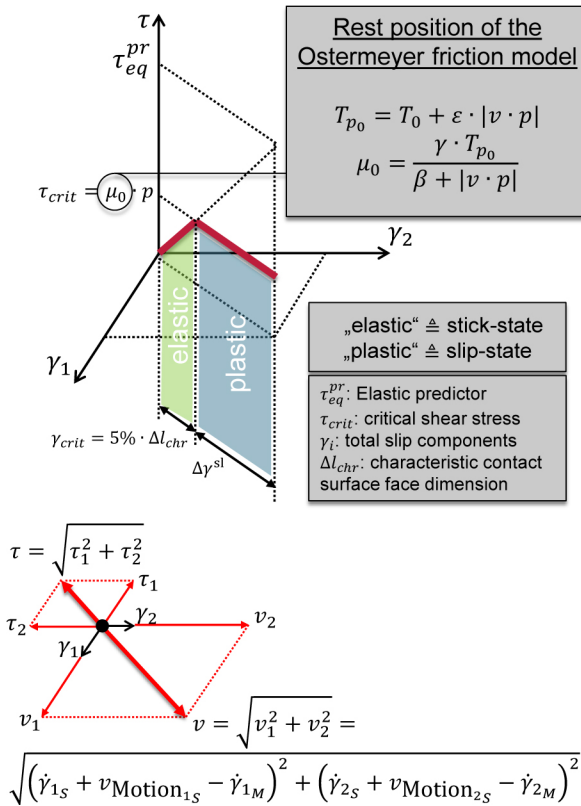
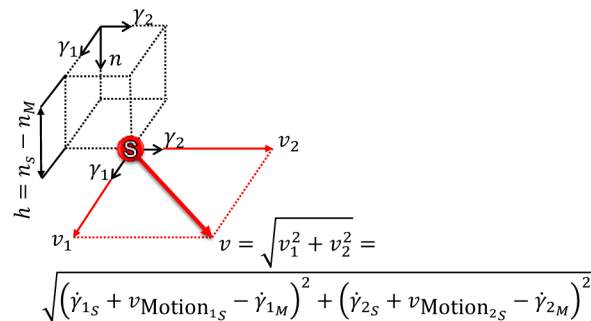


Figure 2. Definition of the tangential shear stress and relative velocity in the local contact coordinate system



Ostermeyer friction model:

$$\begin{aligned} \dot{\mu} &= -\alpha \cdot [(\beta + |r(t)|) \cdot \mu - \gamma \cdot T_p] \\ \dot{T}_p &= -\delta \cdot [T_p - T_0 - \varepsilon \cdot |r(t)|] \end{aligned} \quad \underline{\mu} := \begin{bmatrix} \mu \\ T_p \end{bmatrix}$$

$$r(t) = p \cdot v = k \cdot h \cdot v = k \cdot (n_s - n_M) \cdot \sqrt{(\dot{\gamma}_{1s} + v_{Motion_{1s}} - \dot{\gamma}_{1M})^2 + (\dot{\gamma}_{2s} + v_{Motion_{2s}} - \dot{\gamma}_{2M})^2}$$

$$\begin{bmatrix} \underline{M} \\ \underline{0} \end{bmatrix} \begin{bmatrix} \ddot{x} \\ \ddot{\mu} \end{bmatrix} + \begin{bmatrix} \underline{D} \\ -\frac{\partial f_{\mu}}{\partial \dot{x}} \bigg|_{\xi_0} \end{bmatrix} \begin{bmatrix} \dot{x} \\ \dot{\mu} \end{bmatrix} + \begin{bmatrix} \underline{C} \\ -\frac{\partial f_{\mu}}{\partial x} \bigg|_{\xi_0} \end{bmatrix} \begin{bmatrix} x \\ \mu \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

Figure 3. Definition of the dynamic friction model by Ostermeyer within the local coordinate system and the corresponding extension of the system matrices

The dependencies of the two first order differential equations of the dynamic friction model are summarized in Figure 4. The coefficient of friction has a pressure dependency (dn_s, dn_M), a velocity dependency ($d\dot{\gamma}_{1s}, d\dot{\gamma}_{2s}, d\dot{\gamma}_{1M}, d\dot{\gamma}_{2M}$), a temperature dependency dT_p (representing the coupling of the two ODEs), and a dependency on the friction coefficient history $d\mu$.

Friction contributions
(Ostermeyer friction model)

$$d\dot{\mu} = (\dots)dn_s + (\dots)dn_M + (\dots)d\dot{\gamma}_{1s} + (\dots)d\dot{\gamma}_{2s} + (\dots)d\dot{\gamma}_{1M} + (\dots)d\dot{\gamma}_{2M} + (\dots)d\mu + (\dots)dT_p$$

$$dT_p = (\dots)dn_s + (\dots)dn_M + (\dots)d\dot{\gamma}_{1s} + (\dots)d\dot{\gamma}_{2s} + (\dots)d\dot{\gamma}_{1M} + (\dots)d\dot{\gamma}_{2M} + (\dots)dT_p$$

Coordinate transformation:

$$\begin{bmatrix} n \\ \gamma_1 \\ \gamma_2 \end{bmatrix} = R^{-1} \begin{bmatrix} x \\ y \\ z \end{bmatrix} = R^T \begin{bmatrix} x \\ y \\ z \end{bmatrix} = R^T x_{Global}$$

Figure 4. Dependencies of the two differential equations of first order of the dynamic friction model by Ostermeyer defined in the local coordinate system

The temperature also has a pressure dependency (dn_s, dn_M), a velocity dependency ($d\dot{\gamma}_{1s}, d\dot{\gamma}_{2s}, d\dot{\gamma}_{1M}, d\dot{\gamma}_{2M}$), and dependency on the temperature history dT_p . Due to the pressure and the velocity dependency of the two differential equations, and due to the linearization of the two differential equations in the local coordinate system, a transformation into the global coordinate system is carried out (see Figure 4). The gradients of pressure dependency lead to asymmetric entries within the stiffness matrix (see Figure 3). The gradients of velocity dependency lead to asymmetric entries within the damping matrix. Finally, the gradients of the temperature and friction coefficient lead to entries on the main diagonal and to asymmetric entries within the stiffness matrix.

THE WORKFLOW TO EMBED THE INITIAL SYSTEM INTO AN EXPANDED STATE SPACE

In the following section, the initially intended workflow for the implementation of the MAD using commercial FEM-tool will be explained. Firstly, the well-known conventional workflow (see Figure 5) for the simulation of brake squeal in the frequency domain will be discussed briefly.

As well-known [10, 11], the previous conventional workflow of the brake squeal analysis can be subdivided into three steps (see Figure 5). It is to be mentioned that only the necessary facts in connection with this paper will be introduced here. In the first step, the brake pressure and the disc rotation are applied within a nonlinear static analysis (see Figure 6, top box). Therefore, the contacts between the pads and the disc are established and the disc's rotational velocity is induced using a predefined vector field. The resulting stress state is used in the second and third steps for the eigenvalue analysis.