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On the relevance of a microslip contact model for under-platform dampers

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A B S T R A C T

A crucial part of building reliable models for the design of under-platform dampers for turbine blades resides in the appropriate description of the contact conditions, both in the normal and in the tangential direction.

The aim of this paper is to determine to what extent microslip due to the combined non-linearities along the normal and the tangent of non-conforming contact surfaces influences the damper behavior. The ultimate goal is to determine whether introducing these features in the contact model would improve the performance of numerical routines used at the blade-damper design stage. In order to explore this problem, a purposely developed contact model is tuned on a single-contact test and then included in the numerical model of a curved-flat damper to simulate its cylindrical interface. The damper numerical routine is then validated against the results from an experimental device purposely developed to test the dynamics of a damper loaded between moving platforms.

It is shown that the validated numerical routine featuring the newly introduced contact model predicts, in comparison with the standard contact model (where partial slip and normal approach non-linearity are not considered), a lower dissipated energy by an amount that would not be justifiable to neglect.

1. Introduction

In the frame of damper design the main object in the literature is the development of a calculation procedure that integrates blades' FE model, rigid body model of the damper and contact model in order to predict the damper performance through the solution of the nonlinear dynamic response of the system. In the technical literature, the problem of modeling periodical contact forces at friction contacts is still ongoing [1] and has been addressed by several authors, leading to different contact models. These models belong to the "spring–slider" family, a class of displacement-dependent contact models [2] which neglect features like viscous forces along the normal direction and friction's velocity-dependence. These features, while relevant in other fields, are not normally considered in friction damping applications to turbomachinery.

The first "macroslip" (or "gross-slip") model used in friction dampers was proposed by Griffin [3] in 1980 and extended to include 2D motion on the contact plane [4,5]. The greatest advantage of the macroslip model is the low number of parameters required for its tuning, however it does not model non-linearities in the normal direction neither microslip effects which may become relevant in the case of small relative displacements or large normal loads.

An important contribution to friction interfaces modeling was given in 1938 by Cattaneo [6], who starting from Hertz [7] theory of normal contact of ellipsoids, extended it to a case of two elastic spheres in contact under the action of a constant normal force and a constant tangential force less than that of (Coulomb) limiting friction. Cattaneo showed that the effect of a tangential force smaller than the limiting friction force is to cause small relative motion, referred to as "microslip" (or "partial-slip") over a part of the interface, while the rest of the contact surface deforms without relative motion, a condition referred to as "stick". This microslip contact problem was further explored by Mindlin [8], who extended it to the case of periodically applied tangential loads. Experimental studies that support the theory have been reported by Mindlin [9], Johnson [10], and Goodman and Brown [11].

Menq et al. [12] in 1986 offer one of the earlier attempts at modelling microslip through a 1D spring–slider system to analyze the dynamic response of frictionally damped structures. Sanliturk et al. [13] later presented a microslip contact model constituted by an array of macroslip elements without normal contact stiffness and applied it to a wedge damper. The model is tuned against

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Nomenclature

α_i	share of N_{tot} on the ith macroslip element of the GG array
d	diameter of the non-conforming contact
E	Young's modulus
gap_i	gap of the ith macroslip element of the GG array
k_{ni}	normal spring constant of the ith macroslip element of the GG array
k_{ti}	tangential spring constant of the ith macroslip element of the GG array
l	length of the cylinder line contact
μ	friction coefficient
ν	Poisson's coefficient
N_{tot}	maximum value of normal load experienced by the GG array
$\tilde{N}^i$	normal force at the ith macroslip element of the GG array
N	normal force at the contact
n	normal relative displacement at the contact
$\tilde{T}^i$	tangential force at the ith macroslip element of the GG array
T	tangential force at the contact
t	tangential relative displacement at the contact
s_i	slider position of the ith macroslip element of the GG array

Symbols related to the damper routine

β_d	damper rotation, see Fig. 5
CF	magnitude of the centrifugal force applied to the damper
$\mathbf{F_c}$	vector of contact forces
$\mathbf{F_e}$	vector of external forces
I	moment of inertia
K_R,K_I	real and imaginary part of the complex spring used as an indicator of the damper performance
$\mathbf{K}$	stiffness matrix used to express contact forces $\mathbf{F_c}$ at a given instant in time
$\mathbf{M}$	damper mass matrix
m	mass
$\mathbf{s}$	vector of sliders positions
τ	time
$\mathbf{T}$	transformation matrix to switch from one reference system to another
u	horizontal displacement, see Fig. 5
$\mathbf{U}$	vector of damper displacements
$\mathbf{U}_P$	vector of platforms' displacements
V	vertical force at the contact
ω	frequency of vibration (rad/s)
θ	platforms angle, see Fig. 5
w	vertical displacement, see Fig. 5

Additional subscripts

D	damper
F	relative to forces
$G->L$	global to local
$L->G$	local to global
$L1$, $L2$	damper left contact points ID
L, R	left and right
P	platform
s	slider
U	relative to displacements

Additional superscripts

$1C$, $1S$	first harmonic Fourier coefficients

experimentally observed hysteresis loops however, given its lack of normal contact stiffness excludes any relation between normal force and number of macroslip elements in contact.

A different approach to model microslip requires instead of the discretization of the contact area [14,15]. In both cases the total normal and tangential stiffnesses are nonlinearly dependent on the normal relative displacements due to an increasing percentage contact area with higher normal forces. This approach has the advantage of being adaptable to any contact surface, however it introduces the simplification which sees each spring, corresponding to one point of the contact area, as decoupled from the other points. It is therefore impossible to mimic, at the same time, the same contact area and the same maximum pressure as the Hertzian solution [14].

In order to model the microslip behavior of friction contacts in FE numerical simulations of frictionally damped structures, some authors adopt a Dynamic Lagrangian approach on the contact patch [16], other authors [1,17–20] apply a macroslip friction element (with normal as well as tangential stiffness) to each meshed node belonging to the contact area. In the latter case contact parameters are evaluated for the whole contact by using simplified test arrangements [19]: the friction coefficient is assigned to each contact node, while contact stiffness values are evenly distributed among the contact nodes. It is important to notice how this method allows for the slipping area to grow inward, even if with a different pattern from the one predicted by Cattaneo and Mindlin [18].

Contrary to the traditional approach where the contact surfaces are divided in facing portions connected by one traditional macroslip element [1,17,18,20,21], the parallel array of macroslip elements presented by these authors in [22] allows the normal nonlinearity (and the connected tangential non-linearity) to be captured without a local FE mesh which would require an excessive refinement. The former approach is effective for contacts which are at least initially conforming (i.e. flat on flat as in [1,17,18,20]), and, as such, is applied also in our case on the flat side of the damper (see Sections 3–4), while the latter is much more convenient to model non-conforming contacts and, as such, is applied in our case on the cylindrical line contact of the damper. This model structure, here termed GG array, has three main advantages:

- unlike [21], it takes into account non-linearity along the normal direction when modeling non-conforming contacts;
- unlike Csaba's brush model [14], it makes no assumption on the contact area and does not require an excessively refined FE mesh close to the contact;
- if properly tuned it is capable of mimicking the load–displacement characteristic curves both in the normal and in the tangential direction.

Each macroslip element of the GG array is assigned its own set of