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EXPERIMENTALLY VERIFIED NUMERICAL TECHNIQUE FOR GAS-TURBINE BLADES DAMPING MODELLING BASED ON SUBSTRUCTURE METHOD

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Friction dampers are commonly used in jet and turboprop engines to decrease vibration stresses of turbine blades by irreversible conversion of mechanical energy into heat. Effectiveness of the energy transfer depends on dampers design, material and operational conditions. Optimization of dampers design is a complex task usually associated with nonlinear contact problems analysis and complicated experimental verification. It also coupled with high time costs causing the engineering task to be modified into compound research problem. The paper presents time effective numerical technique for gas turbine blades underplatform dry-friction dampers efficiency study. The presented technique is based on finite-element method with a numerical model reduction by Craig–Bampton and Guyan algorithms. The model consisted of dummy blades pair with wedge damper between the platforms. Technique allows considering three dimensional shapes of parts, nonlinear contact forces, friction, various operational loads and conditions. The model was verified by experiment completed on special test rig based on vibration shaker and 3D laser vibrometer. Relationship between the blades oscillation amplitude and the centrifugal load on the damper was obtained in the range of centrifugal loads 0...700 N. Optimal operational range with maximum damper efficiency was defined. Impact of so called anti-phase and in-phase modes of adjacent blades on damper efficiency was investigated. Impact of dynamic friction coefficients was investigated to identify contact parameters. For the identification purposes special experiment was performed using similar test rig and set up.

Keywords: substructures, dry-friction dampers, gas-turbine blades

1. Introduction

The design of underplatform turbine dampers is an iterative process, during which designers select the shape, material, stiffness and dynamic behavior of the components. The main objective of that process is to minimize blades vibration amplitudes through the engine regimes. To solve that problem it is necessary to use convenient numerical models, which are able to take into the account most of the regularities of blade and damper dynamic interaction. In the past decade a lot of works were dedicated to the mentioned problem presenting numerical models and techniques for the dampers design selection. Due to nonlinearity of the task and time consuming calculations, researches used different numerical approaches to reduce models order and verify these models by special experimental study. In Gola, Firrone, Zucca [1, 2] numerical model for damper design optimization based on harmonic balance method was presented. Verification of the model was based on experimental data of damper kinematics during the forced oscillations and understanding of the basic blade-damper interaction regularities. The experiments show that there is a nonlinear relation between damper efficiency and centrifugal load acting on it. Moreover it was noticed that dampers efficiency depends on system vibration modes during their operation.

In Petrov [3] a method was proposed for reducing models of joined structures using frequency domain analysis of nonlinear steady-state forced response. The method was applied for nonlinear forced response analysis of bladed disks with contact interfaces. In the further works of this author in cooperation with Ewins and Schwingshackl [4, 5] the special attention was given to the study of contact parameters effect on damping efficiency. It was highlighted that selection of correct input parameters for nonlinear contact interface model is essential. So the calculated damping characteristics depend on such variables as: friction coefficient, tangential contact stiffness and finite element discretization accuracy of contact surfaces. Summarizing that brief overview it is worth to note that developing of convenient numerical model for turbine blade damper efficiency prediction is an actual problem. Also it is necessary to develop not just models but simplified engineering technique allowing taking into account the shape of the damper, normal force acting on it during main operational regimes, characteristics and specifics of blade-damper contact and overhaul disk-blades-damper system vibration modes. Considering mentioned above it was decided to create the finite element numerical model for transient blade-damper interaction calculations based on reduction technique via substructures. Two dummy blades and damper were used for demonstration of following presented approach.

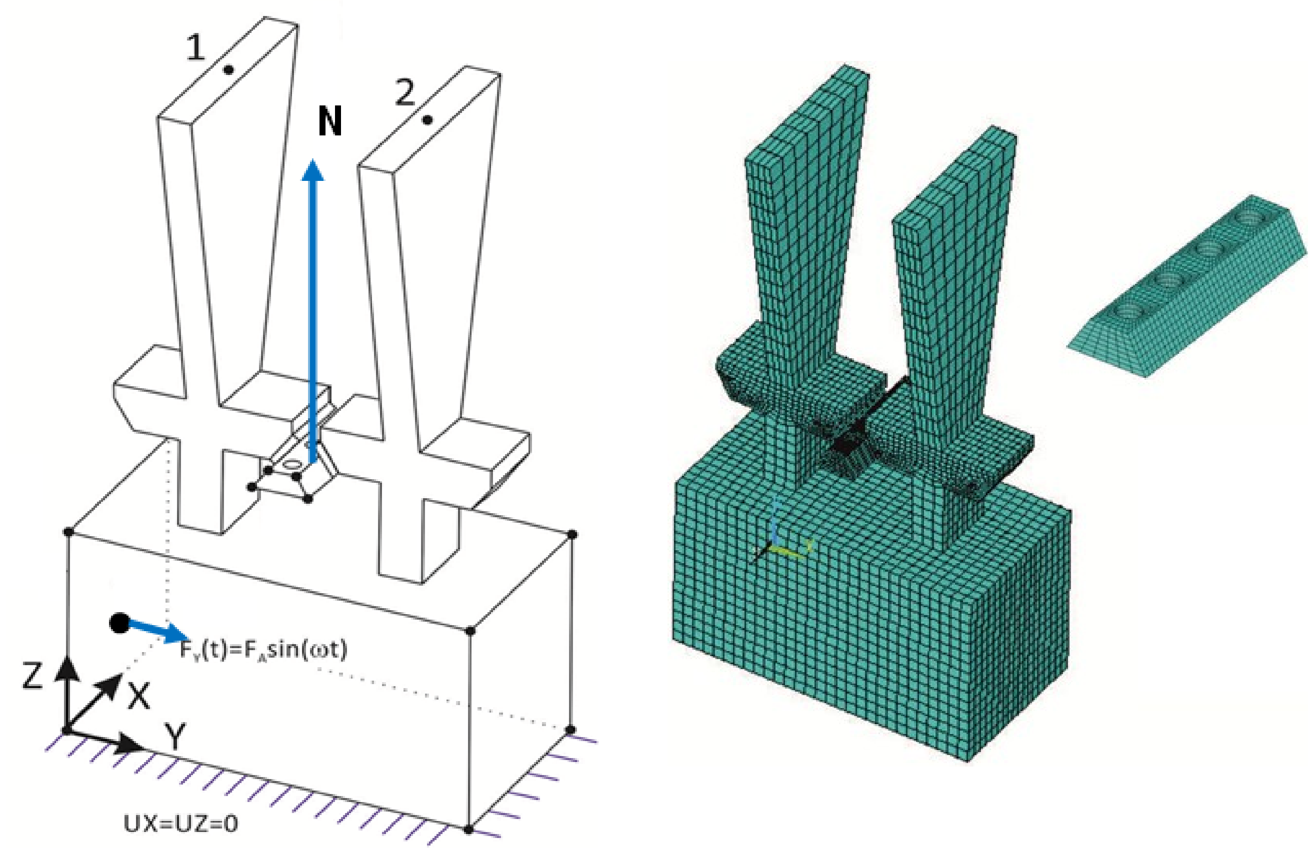
2. Numerical model

Pair of dummy blades with wedge shape underplatform damper was used as the investigation object. Model's dimensions were equivalent to manufactured experimental samples described below to provide adequate verification. Model consisted of two dummy blades, two plates and damper positioned under platforms as shown on Fig. 1, a. The purpose of the plates was to provide possibility of contact parameters variation during the experimental investigation. The pair of blades was connected to a massive base which was linked to the ground by a spring with stiffness adjusted to the experimental measurements. The base was excited by a harmonic force with amplitude adjusted to reproduce exciting acceleration of 8g which was applied in the experiment. The base was restrained in two directions from translation and all rotations were prohibited. Points 1 and 2 (Fig. 1,a) positioned on the blades tips were used for responses analyses after the numerical calculations. Contact between the plates and the blades platforms was considered as bonded while the contact between the damper and the plates was considered as frictional. During the analysis the damper was pressed to the plates by normal force which was transferred through two rods with stiffness same as in the experiment (Fig. 1). Damper's displacements in X axis were constrained but other types of motions were permitted. Such boundary conditions were applied intentionally based on conclusions made in [1] showing that rotational motions of the damper has noteworthy effect on its efficiency. The main purpose of the numerical modelling was to analyse impact of various dampers operational regimes on blades damping efficiency. Based on the prior investigations described in the Introduction it is important to capture effect of real three dimensional shapes of the parts, various damper operational regimes and contact parameters, blades natural mode shapes and rotational movements of the damper. Full finite element (FE) model is presented on the Fig. 1, b. The model is consisted of 17947 isotropic prismatic elements with 8 nodes and had 62527 degrees of freedom (DOF). Minimum element dimension at contact area was 1 mm. Maximum dimension of an element at base area was 3.3 mm. Material properties applied for the model were elastic, material damping was not taken into account.

Equation of motion for the FE model can be represented as:

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\}+[K]\{u\}+\{F_n\}=\{F\} \tag{1}$$

where [M] – mass, [C] – damping and [K] – stiffness matrices, {Fn} – nonlinear force, {F} – external force, {ü} - accelerations, {u̇}– velocities and {u} – displacements vectors.Whereas numerical analysis considered material behavior as a linear, model was reduced using well known algorithms: Craig-Bampton formulation and Guyan condensation. First approach was used for the blades model while Guyan condensation was chosen for the damper model. All reduction methods [6] are based on elimination or transformation of the real physical coordinates of linear part of the model to decrease the number of DOF`s. Craig-Bampton formulation [6] is called as fixed-interface component mode synthesis method. According to that approach model reduction is made by transformation of physical coordinates into generalized coordinates. This method allows taking into account dynamic properties of the structure and keeping ability to connect different models by unaffected interface nodes. The main benefit of the method is that it is not sensitive to a number of nodes kept as master nodes after the transformation [6]. This allows having dynamic properties very close to a full model. The approach was applied to blades model which dynamic response presents main interest in the investigation.



a) b)

Figure 1: (a) Model for investigation; (b) Finite element model.

Equation of motion for the blades after the reduction can be presented as:

$$\begin{bmatrix} M_{mm} & M_{ms} \\ M_{sm} & M_{ss} \end{bmatrix}\{[B] \quad [\Phi]\}\begin{Bmatrix} \ddot{u}_m \\ \ddot{q}_{sp} \end{Bmatrix}+\begin{bmatrix} K_{mm} & K_{ms} \\ K_{sm} & K_{ss} \end{bmatrix}\{[B] \quad [\Phi]\}\begin{Bmatrix} u_m \\ q_{sp} \end{Bmatrix}+\{F_n\}_m=\{F\} \tag{2}$$

where index ***m*** defines interface or master nodes, index ***s*** defines inner or slave nodes. Matrix [B] corresponds to boundary coordinates definition. Matrix [Φ] relates to transformation of physical coordinates into modal, $\{F_n\}_m$ is non-linear contact force and $\{F\}$ is external exciting force. Transformation matrices [Φ] and [B] are defined during reduced model creation. Eigenvalue matrix [φ_s] is the part of the matrix [Φ] and it is defined by modal analysis.

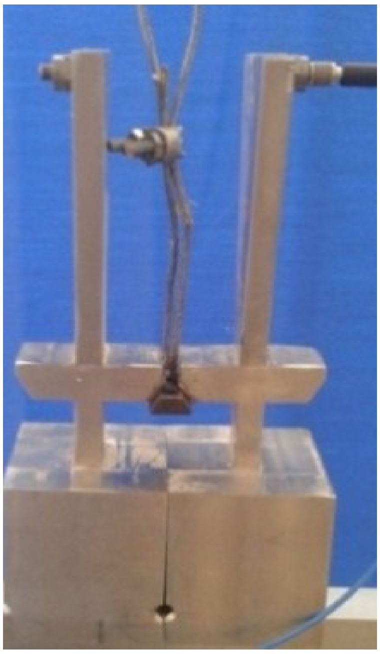
In Guyan condensation method physical coordinates are eliminated. So called master nodes represent inertia properties of the model itself. Therefore this method is sensitive to master nodes amount and distribution. Main benefit of this method is that it does not require structure dynamic properties definition. This approach was applied to the damper model. Equation of motion for damper reduced model can be presented as

$$\lfloor\hat{M}\rfloor\{\ddot{u}_m\}+\lfloor\hat{K}\rfloor\{u_m\}+\{F_n\}_m=\{\hat{F}\} \tag{3}$$

where $[\widehat{K}]$ – transformed stiffness matrix $\{\widehat{F}\}$ – damper`s load vector, $[\widehat{M}]$ – transformed mass matrix.

Linear parts of the blades and damper FE models were compound into separated linear elements named as substructures created by described approaches. Substructures were connected by contact elements built on interface master nodes between the models. Non-linear contact forces were linearized by Pure Penalty algorithm with fixed contact stiffness defined based on full FE model static contact analysis. Friction was described by Coulomb approach. During numerical modelling dynamic friction coefficient μ was supposed to be equal to static. Range of coefficients μ = 0.2; 0.4; 0.6; 0.8 were investigated. The numerical model was verified by experimental results. Experimental technique is presented below.

3. Experimental technique



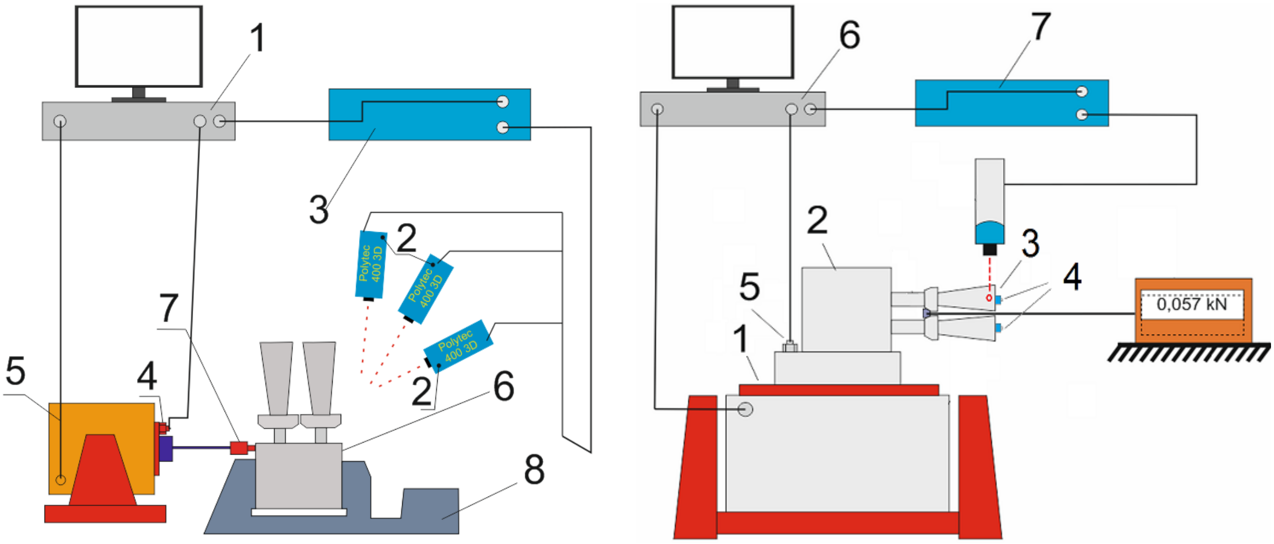
a)

b)

Figure 2: (a) Block of two dummy blades; (b) Wedge damper.

The main goal of the experimental modelling of the turbine blade and dry-friction damper interaction is to obtain data for numerical model verification. The following experimental models were used as a study objects: block of two dummy blades (Fig. 2, a) and wedge shape damper (Fig. 2, b). The part of the rotor disc was simulated by massive block cubic base.

The presented experimental technique assumes two steps approach of the study. At the first step of the experiment the modal analysis of the two dummy-blades block was performed using three dimensional scanning laser vibrometer Polytec PSV-400-3D. Frequency response was obtained by three reference axes at 40 points (20 points on each blade), which is enough to achieve the lower natural modes of the blades system. Scanning was made in frequency range of 250…350 Hz with frequency scanning step of 0.488 Hz. Excitation force had sine-wave form with constant amplitude and increasing frequency. The scheme of first-stage experimental test rig is presenter on Fig. 3,a.



a) b)

Figure 3 – The scheme of experimental test rig for: a) natural modes and frequencies evaluation: 1 – shaker control system; 2- scanning heads; 3- laser signal decoder; 4- shaker control accelerometer; 5- magnet shaker; 6- blades block; 7 – force transducer ; b)force vibration response analysis: 1 – vibro table, 2 – blades block, 3- scanning point, 4- accelerometers for response measurement, 5 – accelerometer for shaker control

During the investigation two resonance frequencies of the blades were defined in selected frequency domain: the in-phase bend mode 306.3 Hz and anti-phase bend mode 335 Hz. Those two modes were used for numerical model verification. At the second stage of the investigation the blade damping efficiency was studied. The investigation was made by force resonant vibration response analysis of the blade-damper system at various levels of centrifugal load for two previously defined natural frequencies. To achieve vibration amplitudes of blades tips up to 1 mm the LDS v 650 magnet shaker was used. As it is shown on Fig.3, b the dummy blades block – 2 with wedge shape damper were clamped on the table of magnet shaker – 1.The response registration by 1-D laser vibrometer was made at point – 3 (Fig. 3, b) at the centre of the peripheral section of the dummy blade, where the maximum vibration displacement of first bending mode was observed. Moreover accelerometers – 4 were mounted for additional amplitude and phase control. During the second step of the experiment five centrifugal loads levels were applied on the damper: 0, 45, 187, 414 and 700 N. The relations between forced vibration amplitudes and simulated centrifugal loads were obtained.

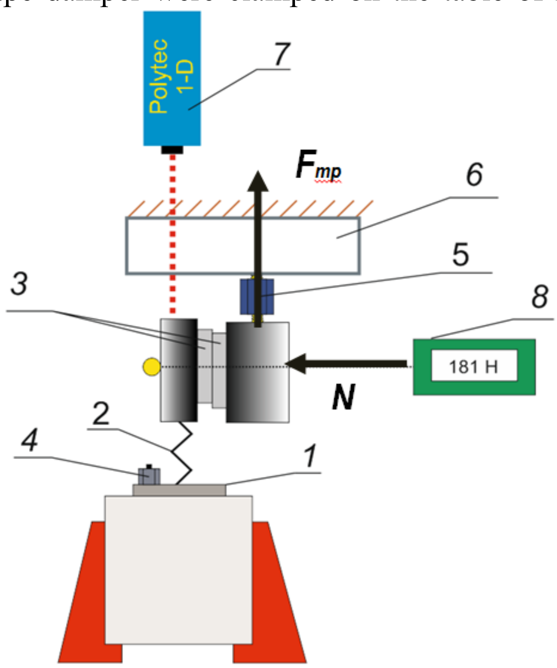


Figure 4. Experimental test rig for contact properties evaluation : 1 – vibro table , 2 – spring element, 3 – contact pair; 4- – accelerometer, 5 – force piezosensor ; 6 – metal base; 7 – vibrometer 8 - dynamometer

Impact of dynamic friction coefficients was also investigated to identify contact parameters. For the identification purposes special experiment was performed using similar test rig and set up. The objective of the experiment was to establish static and dynamic friction coefficients by contact hysteresis. Pair of blocks made from same material as blade's plate and damper were used as study objects. The general view of experimental test rig is presented on Fig. 4. The main idea of the experiment was to excite the vertical relative oscillations of the block pairs (Fig. 4 - 3) under the normal load N, acting in the horizontal direction and controlled via dynamometer (Fig. 4 - 8). The main benefit of this method is direct measurement of the frictional force F_{fr} by piezoelectric sensor (Fig 4 – 5) and relative displacement by means of 1-D laser vibrometer.

Test-rig allows measuring friction forces between two oscillating contact blocks in frequency range to 3000 Hz with relative displacements of contact surfaces from 1 µm to 10 mm. Besides friction coefficient and tangential contact stiffness dependencies on normal force so as work of friction force during one vibration cycle could be obtained through the test. During the experiment friction forces in range of -488 N to 488 N were recorded. Relative displacements with accuracy of 1 µm were registered. Normal forces were measured in range from 0 to 1 kN with accuracy of 1N. Friction coefficients and tangential contact stiffness depending on normal contact load were obtained.

Experimental results were used for numerical model verification.

4. Numerical and experimental modelling results

At the very beginning numerical model reduction was performed and verified by experimental results and full FEM modal analysis. Results of the blades modal analysis are presented in the Table 1. In the investigated system two vibration modes: in-phase and anti-phase were considered as modes of primary interest. FE models were adjusted by additional weights M1 and M2 added to blades tips to achieve same natural frequencies as in the experiment. It is also clear that reduced FE model was not affected by the transformation.

Table 1: Natural frequencies obtained by linear harmonic analysis

Mode	In-phase 	Anti-phase 
	Frequency, Hz	
Experiment	333	306,5
Full model	333	305
Reduced model	333	305

Results of linear harmonic analysis are shown in Table 2. The system of blades have two natural modes corresponding to the experiment: in-phase bend mode and anti-phase bend mode. It was observed that when “blades-damper” system shifts from full stick condition to full slip condition the order of modes shapes changes accordingly. Full slip condition modes order corresponds to experimental partial slip condition which is clearly visible from numerical and experimental results described below. Moreover it is necessary to notice that reduced model frequencies correspond to full model frequencies with maximum deviation of 3% and the model can be used for further studies.

Table 2 - Natural frequencies obtained by linear harmonic analysis

Load	Fullmodel				Reducedmodel			
	Fullstick		Fullslip		Fullstick		Fullslip	
	Anti-pase	In-phase	Anti-pase	In-phase	Anti-pase	In-phase	Anti-pase	In-phase
No	411	465	332	351	411	463	327	350
Yes	421	465	407	352	424	464	419	348

Then so called nonlinear harmonic analysis was performed. Nonlinear harmonic analysis was completed as a transient analysis with nonlinear contact forces taken into considerations. Excitation of the system was made via “chirp” signal in the frequency range from 300 to 500 Hz. Output from that step is a system’s spectrum built using fast Fourier transform (FFT) algorithm. Results of non-linear harmonic analysis are shown on the Fig. 5.

It was observed that presence of friction has certain correlation with vibration modes and load on the damper. As shown on the Fig.5 system has regimes where friction coefficient and increase of load has no impact on frequencies. Those slip regimes are characterized by friction coefficients lower than 0.4 or normal loads under 200 N. When friction resistance steps up together with load increase stick prevailing condition appears at the contact which impacts the frequencies.

Before the regime switch there is transition zone in loads range of 200…400 N where high friction coefficients give change in system’s frequencies. Experimental results are marked as points on the Fig. 5. Experimental and numerical results comparison shows that transition from slip prevailing condition to stick prevailing conditions is more significant in numerical calculations than in the experimental simulation. Calculated frequency increase for in-phase mode is 21% ad for the antiphase mode is 9%. Experiment shows more smooth transition: for in-phase mode frequency increases by 12% and for antiphase mode by 4%. For that particular step numerical model may be partially verified with appropriate accuracy. Numerical results obtained with dynamic friction coefficient 0.6 correlate with experimental results. Those results correspond to friction coefficient range of 0.6-0.8 obtained in contact pair properties evaluation experiment.

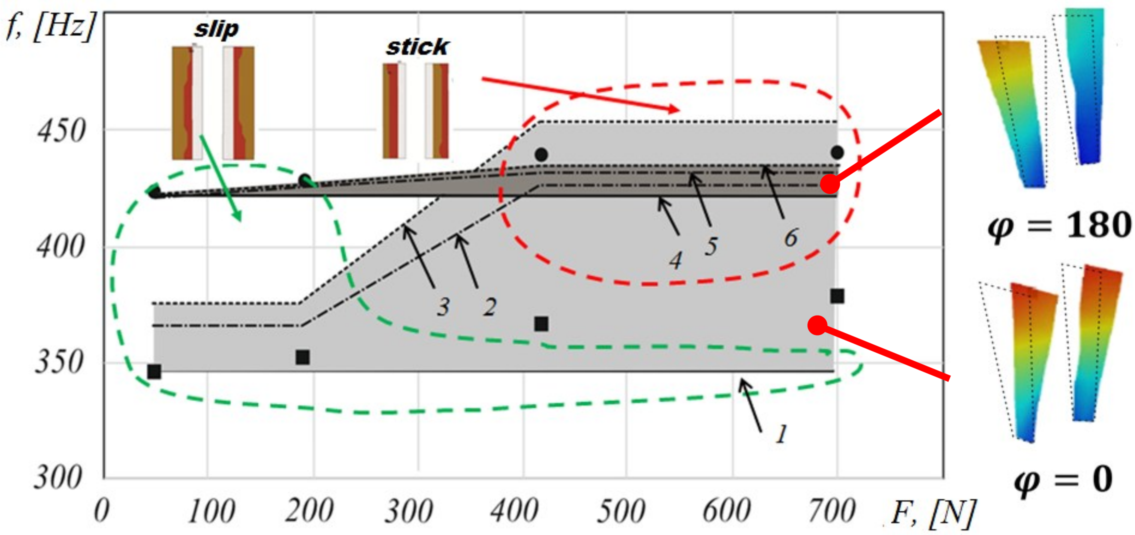


Figure 5. Nonlinear harmonic analysis results: 1, 4 - μ=0,2…0,4; 2,5 - μ=0,6; 3,6 - μ=0,8. Experimental analysis results: ● - anti-phase mode; ■ in-phase mope

When resonance frequencies where established system’s amplitudes response to harmonic excitation on resonance regimes was calculated via nonlinear transient analysis. During this analysis the impact of damper centrifugal loads, contact parameters and system vibration modes was investigated. System’s amplitude response at resonances is illustrated on Table 3. Numerical results are compared with experimental ones. It was observed that the system has three basic regimes: slip prevailing condition (largest zone in Table 3), stick prevailing condition (zone with 414 to 700N loads and friction coefficient of 0.6 to 0.8 in Table 3) and transition zone (dotted) where amplitudes had the minimum magnitudes and spectral peaks made by FFT were under confidence interval of 95%. Grey cells in Table 3 correspond to anti-phase mode and white cells highlight the in-phase mode. It is clearly visible the modes shift at stick prevailing regime. Let consider that damper efficiency achieves an optimum point when blades amplitudes are minimal for both system’s vibration modes. With such assumption we can see that for the experiment optimal operational regimes lie at centrifugal loads range 187…414 N, where we can observe minimal amplitudes on both modes.

For the numerical calculations the optimal range of 187…414 N was also observed for friction coefficient of 0.6. So the optimal damper operational regime is observed in transition zone and at lower boundary of stick prevailing zone, where both main bending modes give minimum blades amplitudes. It means that during damper design it is necessary to identify such a transition regimes to set-up operational parameters close to beginning of stick prevailing zone for maximum efficiency. Based on analyses provided we may conclude, that numerical model is verified by experimental results with appropriate accuracy. Model can be used for damper design optimization and prediction of its optimal operational regimes.

Table 3. System`s amplitude responses at resonances

μ	45 N		187 N		414 N		700 N	
	1	2	1	2	1	2	1	2
0.2	2.20	0.24	2.48	0.50	2.50	0.35	2.00	0.35
0.4	1.60	0.25	1.23	0.40	1.40	0.22	1.60	0.60
0.6	0.42	0.22	0.30	0.16	0.33	0.18	0.88	0.36
0.8	0.42	0.51	0.60	0.50	0.50	0.40	1.20	0.33
Exp. results	0.70	0.10	0.65	0.20	0.80	0.25	0.90	0.30

5. Conclusions

Dry friction underplatform dampers efficiency was investigated. Numerical technique based on substructure model reduction method was presented. Experimental verification was performed using developed methodology and special test rig.

During the investigation it was observed that: during blade-damper contact interaction there are three basic regimes: slip prevailing, stick prevailing and transition regime; - blades amplitudes at resonances are close to minimum for in-phase and anti-phase bending modes at transition regime and beginning of stick prevailing regime (187 N…414N); system’s (blades-damper) vibration modes have significant impact on damper design optimization and should be taken in account.

Presented numerical technique is time effective and can be used for analyzing more complex systems such as disk-blades-dampers. Model considers 3D shape of parts, nonlinear contact forces, rotational motion of the dampers and vibration modes.

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