Verification of a Mathematical Model of a Dry Friction Damper for a GTE Blade

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Abstract. This paper presents the verification of the mathematical model utilized to calculate the damping capacity of a dry friction damper. The mathematical model is based on determining natural frequencies and mode shapes, as well as relative characteristics of the friction energy losses. To estimate the applicability of the mathematical model to full-scale assemblies of a gas-turbine engine, the model was verified by the results of the experiment. The experiment was carried out in a laboratory where the damping capacity of the gas-turbine engine blade with a model under-platform damper was examined at constrained and damped oscillations. During the experiment multiple materials of contact pair were examined to determine how microslip affects the damping capacity. Based on the results of comparing calculations with the experiment, by selecting the dry friction coefficient, a good convergence on the oscillation decrement and natural frequency was achieved. The conducted research has shown that the mathematical model provides reliable results with Coulomb friction and can consider microslip effects on the damper capacity.

1. Introduction

The level of dynamic stresses in gas turbine engine components (GTE) defines the high-cycle fatigue life. Often, the resonance oscillations are the main cause of high dynamic stresses. One way to reduce resonance stresses is to increase the damping capacity of the system by using a dry friction damper with its efficiency depending on the proper setting of the system. For complex and critical applications, in particular GTE, experimental adjustment is often money and labour-consuming. Because of this, it is reasonable to carry out mathematical modeling of the experiment prior to full-scale testing.

The most appropriate method for mathematically presenting a complex full-scale structure, such as the GTE blade, is the finite element method (FEM). For an accurate representation of the system oscillations with dry friction it is enough to use the complete equation of motion:

\[
[M] \cdot \{\ddot{X}\} + [C] \cdot \{\dot{X}\} + [\alpha] \cdot \{X\} + \{f_{\text{tr}}(X, \dot{X})\} = \{Q\}
\]

where: \([M]\), \([C]\) are generalized matrices of masses and stiffness; \(\{\ddot{X}\}, \{\dot{X}\}, \{X\}\) are generalized vectors of acceleration, velocity and displacement; \(\{f_{\text{tr}}(X, \dot{X})\}\) is the vector of nonlinear contact forces (dry friction force); \(\{Q\}\) is the vector of constraining harmonic force; \([\alpha]\) is the matrix of dissipative forces different from the dry friction. The dot means the time \(t\) integration.

The results of the technical literature review show that two numerical methods are currently used to calculate the damping of the GTE assemblies:
- direct time integration method
- harmonic balance method.

Contact dynamical problems solved by the direct method [1, 2, 3] are still highly computationally-demanding [4, 5]. The application of the matrix reduction procedure and the use of contact elements such as node-to-node (GAP) elements [1] slightly reduces the calculation time. The direct method is often used as a reference to set or verify an alternative damper model [4, 8, 9].

In order to reduce the required computational resource, various simplifications of the mathematical model are implemented. The most popular developments include the method of harmonic balance [9, 10] using several harmonics (and refining the friction force in a nonlinear formulation) [4, 6, 8, 11], which partially addresses the computational resource issue, especially when binding with the matrix
reduction procedure and using GAP elements [7, 6, 8, 12]. In both cases, the mathematical model contains a set of secondary components that improve the accuracy of representation of the damped oscillatory system behavior, but at the same time make the system more complicated. As a consequence, when solving practical engineering problems, there are difficulties in understanding the process of damping the structure, because a system with a damper contains too many independent variables.

Therefore, today it is essential to seek for opportunities to bring the problem to a general form and to carry on the development of analytical approaches. A mathematical model which will make it possible to cut the problem dimension and computation time down with no compromise in quality of the results obtained is required. A mathematical model that meets these requirements was described in [13].

In this model secondary components were dropped in favor of the principal ones. Primary parameters are mass-stiffness property of the system and stiffness of the damper, which are sufficient to define the damping capacity of the damper. Secondary parameters are the damping level and the resonance amplitude level of oscillations of the system without the damper, the constraining force or a friction coefficient. Secondary parameters are taken into account at the stage of post-processing of the calculated data and allow to determine the optimal damper clamping force and the decrease of oscillation amplitude when the damper is used. The implementation of the model is presented in solving the problem of defining the natural mode shapes and frequencies. For the specific design of the damper and the system vibration mode, a relationship between the friction damping level and the parameter combining the friction force and the level of resonance oscillations amplitude (or dynamic stresses) is derived [13].

In the case of constrained oscillations, the equation of motion (1) is divided into two linear equations that are solved independently:

\[
[M] \cdot \{\ddot{X}\} + [C + k] \cdot \{X\} + [\varepsilon \cdot \{\dot{X}\} = \{Q\} \tag{2}
\]

\[
[M] \cdot \{\ddot{X}\} + [C + k + \xi] \cdot \{X\} = 0 \tag{3}
\]

From Eq. (2) we get the resonance oscillation amplitude at the known constrained force \(\{Q\}\) and damping (decrement of the system without \(\delta_0\) dry friction element). In Eq. (3) the effect of the friction force \(F_{FR}\) contained in \(f(X, \dot{X})\) is modeled by the elastic element \(\xi\). Through the stiffness variations of the elastic element \(\xi\), from Eq. (3) we get all possible values of oscillation decrement \(\delta_{FR}\) depending on the friction energy loss.

In order to demonstrate the applicability of the developed mathematical model to full-scale GTE components, the computational and experimental study of the model damper positioned under the platform’s inner surface of the GTE compressor blade was conducted. The experiment was carried out using a shaker (Fig.1), where the blade oscillated transversally in first mode.

Due to vibration levels of GTE assemblies, the slipping does not exceed micron orders in the damper contact areas, thus leading to microslip effects [12, 15]. When it comes to damping, the negative effect of microslip is a reduction of the hysteresis loop area, i.e. a decrease in the damper efficiency, sometimes a significant one [15]. Therefore, most authors take this effect into account during computations using the Dahl model [14] instead of the Coulomb friction, and also conduct experimental studies on special rigs to define contact stiffness, friction coefficients, microslip effects, and etc. Experimental studies were described by Griffin, Petrov, Swchedovich, Panning, Gola, Zusa [16-22]. Therefore, it is concluded that the mathematical model should be able to take the effects of microslip into account.

2. Experimental study. A rig was designed for the experimental study so the efficiency of the model damper could be examined in the shaker system conditions. The full-size blade with a platform was used as a damped object (Figure 1: image, Figure 6: model). The LDS V875-440 Shaker was used.

The test rig is shown in Figure 1. The damper was clamped and pulled by a line (line diameter is 0.6 mm), line tension was adjusted by a stack of weights. The length of the line from the damper to the block was 1.5 m, it provided unlimited extension of the line under damper sliding: tensile stiffness of
the one-line meter is \( \sim 860 \) N/m, which is negligible. The angle of the line to the horizon is 45°. A strain gauge was adhered to a suction side of the airfoil to control the level of dynamic stresses.

**Figure 1.** GTE blade with a model damper as part of the test rig on the shaker

2.1 *The first stage.* During the first stage of the experimental study, the oscillation decrement \( \delta_0 \) of the system without a damper was determined as function of the blade dynamic stresses level \( \sigma \). The methods described by V.V. Matveev [23] were used to calculate the reliable decrement value. Figure 2 shows the dependence of \( \delta_0(\sigma) \), which is further used to calculate the friction decrement \( \delta_{FR} \) in the system with a damper.

**Figure 2.** Oscillation decrement of a system without a damper, calculated from damped oscillations (solid line) and constrained oscillations (markers).

2.2 *The second stage.* During the second stage, an experiment with constrained oscillations was conducted. Figure 3 shows the amplitude-frequency response of the signal recording from the strain gauge at constrained oscillations of the blade with various normalized clamping forces of the damper (chromium-coated). The results of the experiment interpretation are provided in Section 4.
In the process of experiment at constrained oscillations, the authors discovered a decrease in damping capacity of the damper compared to the expected calculated one (see Figures 5 and 11). The damping capacity deterioration factor was $\delta_{FR\text{Test}}/\delta_{FR\text{Calc}}=0.8$. A decrease in damping capacity indicates a decrease in friction force, therefore, there was a reduction in the hysteresis loop area from the theoretical potential one at Coulomb friction. The amplitude of damper slipping was in a micron order, so, at an optimum normalized clamping force of 14 the amplitude of slipping was $\sim2\mu m$ (measured by the PDV-100 Laser Vibrometer). It is assumed that the deterioration occurred due to the significant effect of microslip.

Experimental studies confirm the conclusion that it is important to consider such factors as microslip effects, wear and tear, and the instability of the dry friction coefficient. These factors have a negative impact on the repeatability and results of the experiment. In support of the facts mentioned above, the results of the experiment with different contact pairs (see Table 1) are given in Figure 5. For reference: TsVSP-3s coating is the soft wear-resistant coating, and chrome is the hard wear-resistant coating. It is apparent from Figure 5, that microslip effects have a strong influence on damping. It is proposed that an increase in the amplitude of relative displacement in contact, an increase in surface stiffness (hardness) and the implementation of wear-resistance actions will reduce the negative impact of microslip effects.

| Table 1. Contact pair characteristics |
|--------------------------------------|
| Contact pair                        | Cr-V-Mn quenching, uncoated | Cr-V-Mn chromium coating | Cr-V-Mn quenching, TsVSP-3s coating |
| Hardness of material                | 60 HRC                      | 20 HRC                   | 56 HRC                               |
| Hardness of surface                | -/-                         | > 60 HRC                 | not measured                         |
| Actual roughness of friction surface | Ra 0.06-0.12               | Ra 0.55-0.59             | Ra 1.26-1.77                         |
| Friction surface after test         | fretting corrosion          | light wear               | no change                            |
2.3 The third stage. During the third stage, an experiment aimed at achieving the maximum calculated level of friction decrement $\delta_{FR}$ for the friction pair with chromium coating was carried out. This experiment showed the best results (see Figure 5). The actions consisted of doubling the level of displacements and friction force in contact compared to the previous experiment. Thus, the hysteresis loop area would be increased by 4 times, which will reduce the microslip effects. In order to complete the objective, an experiment with damped oscillations was carried out, as in case of constrained oscillations it was hard to provide a combination of these parameters.

Figure 4 shows the signal recording from the strain gauge that was taken during the experiment. It depicts the moment of steady-state resonance oscillations of the blade with a subsequent break-down to damped free oscillations. The results of comparison with the computations are provided in Section 4.2.

3. Computing the damping capacity of the damper.

3.1 The ANSYS finite element package was used to calculate the damping capacity of the damper. The model consisted of 113346 nodes and 61828 Solid187 elements with intermediate nodes. Fastening of the rig in the computation was the same as the actual fastening to the shaker. The result of the computations of the blade without a damper is shown in Figure 6, where the first mode lateral oscillations are illustrated.

The damper position and the scheme of damper operation are shown in Figures 7 and 8, respectively. At vibrations of the blade airfoil, the platform in an area of the damper’s location moves mainly in a direction of $0Z$ axis, based on this feature the damper location and a friction zone have been defined when designing the rig. The damper is in contact with the blade platform along the line forming a conventional hinge (see Figure 8).
The contact was modeled with elastic elements (Bonded contact type in ANSYS). The damping capacity was calculated in several formulations that reflect the effect of stiffness of normal elastic elements (fkn) in contact on the result of calculation. The stiffness of normal elastic elements can be selected from the specific conditions of damper operation. It is found that the steady-state solution is achieved at high normal stiffness from ~ 100000 MPa, which is not higher than the Young modulus of steel. The question about the correctness of fkn parameter selection remains open. The stiffness of tangential elastic elements (fkt) simulates the friction force effect, i.e. the variable parameter that varied from 10 to 20000 MPa.

The results of computations are shown in Figure 9 and are the \( \delta_{FR}(P \cdot k/\sigma) \) function, where \( P \) is damper pressure force (N), \( k \) is friction coefficient, \( \sigma \) is the level of dynamic stresses in a strain gauge. Maximum friction decrement was \( \delta_{FR} = 5.9 \% \).

### 3.2 Simulation of microslip effects.

It is proposed to simulate the microslip effects by reducing contact stiffness as described below. In the simulation model, the stiffness of the damper’s surface layer was reduced in a tangential direction. Stiffness was reduced by adding an intermediate layer between contact pairs. The intermediate layer was simulated as thin, weightless and deformable. A contact (BONDED) with selected tangential elastic element stiffness \( fkt=13000 \) MPa was set between the intermediate layer and the damper, stiffness of normal springs was set high \( fkn=1e+07 \) MPa. The model was adjusted according to the maximum level of decrement \( \delta_{FR}^{exp}=4.7 \% \), achieved at constrained oscillations (Figure 11).

The results of computations with reduced stiffness of the damper surface layer are marked in Figure 9 by a dashed line, whereas the solid line is the initial solution. Consistent elastic coupling in the damper reduced its damping capacity, and the section of the graph with intensive damper slipping \( (P \cdot k/\sigma < 0.3) \) remained the same. This correlates with the data provided by Gu Weiwei and Griffin [9, 24].

![Figure 9. Friction decrement (%) vs P-k/σ at different values and ratios of contact stiffness.](image)

### 4. Verification.

The simulation model was verified for the chromium coated contact pair.

#### 4.1. Model verification at constrained oscillations.

Figures 10 and 11 show the results of the experiment being interpreted as a dependence of normalized dynamic stresses (resonance peak) and decrement \( \delta_{FR} \) on damper normalized clamping force \( P \). The experiment was repeated 4 times. Comparing the computations with the experiment shows that good convergence in terms of decrement level has been achieved: maximum experimental decrement was \( \delta_{R}=4.6 \% \), simulated decrement was \( \delta_{FR}=5.9 \% \). The results of the adjusted model and the selected friction coefficient computations are marked by a dashed line.
Figure 10. Simulation and experimental dependence of $\sigma(P)$ with selection of friction coefficient

The results of the selection of dry friction coefficient are given in Figure 12. It was determined that the friction coefficient varied from 0.4 to 0.8 with the increase of damper clamping force. Figure 13 shows the dependence of the friction coefficient on the parameter $P\cdot k/\sigma$.

Figure 12. Selection of the friction coefficient: the line is computations, and the marker is the experiment

Figure 13. Selection of the friction coefficient: the line is computations, and the marker is the experiment

Figure 14 shows how the blade's natural frequency changes with the $P\cdot k/\sigma$ parameter. Figure 15 shows the dependence of $\delta_{FR}(P\cdot k/\sigma)$ with selected coefficients of friction, where the markers signify experimental data, the dashed line signifies re-computations with reduced bonding, and the solid line is initial computation.
Figure 14. Dependence of $f(P \cdot k/\sigma)$. Markers are the experiment, and the line is computations.

Figure 15. Dependence of $\delta_{tr}(P \cdot k/\sigma)$. Markers are the experiment, the solid line is initial computations, and the dashed line is computations with reduced bonding of friction surface.

4.2 Verification of the model at damped oscillations.
Figure 16 shows the dependence of $\delta_{FR}(P \cdot k/\sigma)$, where markers signify experimental points with the selected friction coefficient, taken from the results of comparison at constrained oscillations (Figure 13). A trend line with cubic approximation (dashed line) was plotted by the experimental points. A solid line signifies calculation results. Figure 16 shows that the trend line falls well on the calculated curve. Uneven experimental decrement can be explained by the rig vibrating at its own frequency, which "merged" with the oscillations of the blade, as well as by the specific formula for decrement [23]. Thus, the calculated oscillations decrement $\delta_{FR}$ was achieved.

Figure 16. Dependence of $\delta_{FR}(P \cdot k/\sigma)$. Comparison of computations with the experiment at damped oscillations.

Conclusion.
A mathematical model developed by the authors in order to determine the damping capacity of the dry friction damper based on the calculation of natural frequencies and modes of oscillations has been tested in this paper. After interpreting experimental data and comparing the experiment with the computations, we can conclude as follows:

Quantitative convergence of computation results with the experiment on the level of the damper’s damping capacity, on the level of dynamic stresses decrease and on natural oscillations frequency is obtained. The model is adjusted by selecting the dry friction coefficient. It is shown that the model is able to consider microslip effects on the damping capacity of the damper. The conducted studies show that the mathematical model is sufficient for practical application, including the GTE components.

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