Adaptive control of normal load at the friction interface of bladed disks using giant magnetostrictive material

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Abstract
A novel application of magnetostrictive actuators in underplatform dampers of bladed disks is proposed for adaptive control of the normal load at the friction interface to achieve the desired friction damping in the structure. Friction damping in a bladed disk depends on operating parameters, such as rotational speed, engine excitation order, nodal diameter normal contact load, and contact interface parameters, such as contact stiffness and friction coefficient. The operating parameters have a fixed value, whereas the contact interface parameters vary in an unpredictable way at an operating point. However, the ability to vary some of these parameters such as the normal contact load in a controlled manner is desirable to attain an optimum damping in the bladed disk at different operating conditions. Under the influence of an external magnetic field, magnetostrictive materials develop an internal strain that can be exploited to vary the normal contact load at the friction interface, which makes them a potentially good candidate for this application. A commercially available magnetostrictive alloy, Terfenol-D is considered in this analysis that is capable of providing magnetostrain up to $2 \times 10^{-3}$ under prestress and a blocked force over 1500 N. A linearized model of the magnetostrictive material, which is accurate enough for a direct current application, is employed to compute the output force of the actuator. A nonlinear finite element contact analysis is performed to compute the normal contact load between the blade platform and the underplatform damper as a result of magnetostrictive actuation. The nonlinear contact analysis is performed for different actuator mounting configurations and the obtained results are discussed. The proposed solution is potentially applicable to adaptively control vibratory stresses in bladed disks and consequently to reduce failure due to high-cycle fatigue. Finally, the practical challenges in employing magnetostrictive actuators in underplatform dampers are discussed.

Keywords
Giant magnetostrictive material, Terfenol-D, actuators, friction damping, high-cycle fatigue, bladed disk

I. Introduction
Dry friction damping is widely used to reduce vibratory stresses in bladed disks to avoid failure due to high-cycle fatigue (Griffin, 1980; Srinivasan and Cutts, 1983). Friction damping is the most popular solution in this field, since it works effectively in harsh environmental conditions and it is very economical. Different designs of friction damper, such as underplatform dampers (UPDs), shroud and snubber contacts, lacing wire, ring dampers, blade roots, strip damper, and multiple friction contacts, are investigated in the literature to characterize their damping potential and to design the best possible solution for a given bladed disk (Afzal et al., 2016; Laxalde et al., 2007, 2010; Lesaffre et al., 2007; Petrov and Ewins, 2006, 2007; Szwedowicz et al., 2008b; Wu et al., 2012; Yang and Menq, 1997b; Zucca et al., 2012). In addition, several numerical tools (Petrov and Ewins, 2002; Sextro, 2000; Yang and Menq, 1997b) are developed to predict the nonlinear response of a bladed disk with friction contacts. Nevertheless, the experimental studies (Sanliturk et al., 2001; Sever et al., 2008; Szwedowicz et al., 2008a) performed by several researchers indicate that the simulated and measured nonlinear response curves often differ in amplitudes and resonance frequencies, which

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is attributed to the complicated nature of measurement, contact modeling approach, and the variability in the input data.

The nonlinear forced response curve depends on operating parameters (rotational speed, engine excitation order, nodal diameter, and normal load), contact interface parameters (normal and tangential contact stiffness and friction coefficient), and relative motion at the friction interface. The operating parameters have a fixed value at an operating point, and therefore, friction dampers are often tuned for a single nodal diameter and engine excitation order (Szwedowicz et al., 2005). The UPDs that are commonly adopted to mitigate resonant vibration are often optimized with respect to their mass that determines the normal contact load at the friction interface. A typical variation of the nonlinear forced response curve with normal contact load, is shown in Appendix 1, indicates that the optimum damping in the system occurs at a particular normal load. However, due to uncertainty and variability of the contact interface parameters, the friction damping obtained, in practice, may differ significantly from the optimum value. Moreover, in the real operating conditions, the nonlinear forced response curve can deviate from the predicted response curve, due to parameters that are not considered in the simulation. Therefore, this article proposes a novel method to control the normal contact load adaptively, in a way that the optimum friction damping in the system can be achieved despite the variation in the interface parameters and other unknown factors. Moreover, resonances at different operating conditions are possible to damp using adaptive control of the normal load.

To achieve an adaptive control of the normal load, the nonlinear forced response curve can deviate from the predicted response curve, due to parameters that are not considered in the simulation. Therefore, this article proposes a novel method to control the normal contact load adaptively, in a way that the optimum friction damping in the system can be achieved despite the variation in the interface parameters and other unknown factors. Moreover, resonances at different operating conditions are possible to damp using adaptive control of the normal load.

2. GMM

The GMM Terfenol-D (Tb0.3Dy0.7Fe1.95) is a compound of the rare earth metal terbium, dysprosium, and iron. It is used in a wide range of quasi-static and dynamic applications, for example, active noise and vibration control, acoustic devices, linear and rotational motors, ultrasonic cleaning, micro-positioning, infrastructure applications, and many other actuation and sensing devices (Dapino, 2004). The magnetostrictive property of GMM involves a bidirectional energy exchange between magnetic and elastic states of the material, and therefore, it provides mechanisms both for actuation and sensing (Olabi and Grunwald, 2008). Strain generated in the GMM is caused by the re-orientation of the small magnetic domains in the direction of the applied magnetic field, which is known as the Villari effect (Calkins et al., 2007; see Figure 1(a)). Conversely, mechanical stress imposed on the GMM can produce measurable changes in magnetization, known as the Joule effect (Calkins et al., 2007) which is used in sensing applications. In this article, we focus on the Joule effect for the purpose of quasi-static actuation. Nominal material properties of Terfenol-D are listed in Table 1. Material properties such as the elastic modulus E often vary substantially during operation (Kellogg and Flatau, 1999), and the magnetostrain (\(\lambda = \Delta L/L\)) is a nonlinear function of the applied magnetic field H (see Figure 1(b)).

Hysteresis behavior (not shown in the figure) is also present in the \(\lambda-H\) curve (Zhou et al., 2006). These variations make the design of a magnetostrictive actuator challenging and require accurate assessment of material property behavior under varying operating conditions. Nevertheless, Terfenol-D has a high load-bearing capacity, high reliability, and unlimited life cycle, and its magnetostriction does not deteriorate with time and number of cyclic stress (Rajapati et al., 1996), thus making it attractive in many applications. Furthermore, Terfenol-D drive elements have a very high yield strength in compression (700 MPa) compared to the tension (28 MPa) and exhibits enhanced magnetostriction under moderate compression, and as a consequence, Terfenol-D actuators are operated almost exclusively under a compressive load (Calkins et al., 1997; Grunwald and Olabi, 2008).
3. Linear modeling of magnetostrictive actuator

Computation of the stress and strain generated by the input current $I$ in a GMM actuator is a complex task due to its bidirectional energy exchange characteristic and nonlinearity present in the $\lambda - H$ curve. Furthermore, the response behavior and mechanical properties (elastic modulus and magnetoelastic coupling) of Terfenol-D transducers vary significantly with mechanical prestress, magnetic bias, applied magnetic field, and frequency (Calkins et al., 1997; Dapino et al., 1996; Huang et al., 2007). Despite all these challenges, several physics-based (Dapino et al., 2000a, 2000b, 2002; Gu et al., 2013) and phenomenon-based (Kaltenbacher et al., 2009; Tan and Baras, 2004) models describing the magnetostriction are available in the literature. Physics-based models are based on Jiles–Atherton model of ferromagnetism (Jiles and Atherton, 1986) and require substantial physics knowledge and input parameters. On the contrary, phenomenon-based models do not provide physical insight into the behavior of the material but are commonly adopted when material behavior is not the main focus. Although these models are useful for dynamic applications, their level of detail and complexity is too high for the quasi-static (direct current (DC)) application proposed here. Therefore, a linearized magneto-elastic model proposed in Braghin et al. (2011) is employed in this article, and it is validated against the commercially available software COMSOL® com (COMSOL, 2016).

The primary components of a magnetostrictive actuator are shown in Figure 2 and consist of a cylindrical Terfenol-D rod, a wound coil, an enclosing permanent magnet, and a prestress mechanism. The rod is manufactured such that magnetic moments are primarily oriented perpendicular to the rod axis. Moreover, a prestress mechanism is required to fully align the distribution of magnetic moments perpendicular to the rod axis and to allow the transducer to be operated in compression (Zhang et al., 2004). A supplied current, flowing through the coil, generates a magnetic field inside the Terfenol-D rod. Consequently, this leads to an alignment of the magnetic domains along the longitudinal axis and thus generates a strain inside the rod and a high output force ($F_m$). Note that the output force is only generated if the end of the rod is constrained and is known as a blocked force ($F_b$) at an infinite blocking (see Figure 5(b)). Since infinitely rigid constraint is not practically possible in real life and therefore $F_m < F_b$ in practice.

Linearization of GMM behavior is performed around the bias magnetism ($H_b$) under the following assumptions (Figure 1(b)):

- Low working frequencies close to DC application;
- Reversible magnetostriction processes (hysteresis can easily be avoided in a DC application);
- Stress and strain distribution uniform throughout the magnetostrictive rod.

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**Table 1.** Material properties of Terfenol-D.

| Elastic modulus | Density       | Tensile strength | Compressive strength | Curie temperature | Magnetostrain |
|-----------------|---------------|------------------|----------------------|-------------------|--------------|
| 30–90 (GPa)     | 7870 kg/m³    | 28 MPa           | 700 MPa              | 380°C             | 1000–2000 ppm |

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**Figure 1.** (a) Joule magnetostrain produced by a magnetic field $H$. (b) Magnetostrain $\Delta L/L$ produced by varying the magnetic field and bias magnetic field $H_b$ provided by a permanent magnet.
All the above assumptions are valid for a quasi-static application, and thus, the linear coupled magnetomechanical equations (Clark, 1980) read as

\[ e = s^H \sigma + dH \]

\[ B = d^* \sigma + \mu^\sigma H \]

where \( e \) is the strain, \( \sigma \) is the stress, \( s^H \) is the mechanical compliance at constant applied magnetic field strength \( H \), and \( d = (\partial e/\partial H)\sigma \) and \( d^* = (\partial B/\partial \sigma)H \) are the linear magnetoelastic cross-coupling coefficients, while \( B \) and \( \mu^\sigma \) denote the magnetic flux and permeability, respectively, at a constant stress. If the magnetostrictive process is assumed to be reversible as in the present case, then \( d^* = d \). This is normally true for low-level driving currents and magnetic fields. Note that the generated strains in equation (1) depend upon both the elastic properties of the material (modeled by the term \( S^H \sigma \)) and the magnetic inputs (modeled by \( dH \)). Equation (2) models the converse magnetostrictive effect in which a magnetic flux is generated by stresses in the material. The output force \( F_m \) exerted by the magnetostrictive rod (Braghin et al., 2011) on the attached body (in this case UPD) is

\[ F_m = \sigma A \]  

where \( A \) is the cross-sectional area of the rod and \( \sigma \) is a function of the magnetic field strength \( H \) and strain \( e \) that is rewritten as

\[ \sigma = \frac{-e + dH}{s^H} \]  

Furthermore, Ampere’s law describes a simple relationship between the generated magnetic field \( H \) and the current \( I \) in a coil winding that reads

\[ H = \frac{n}{s^H} I \]  

where \( n \) is the number of turns in the coil and \( l \) is the elongated length of the magnetostrictive rod. Since deformation \( x < < l \) and therefore \( l \) is approximated as

\[ l = L + x \approx L \]  

where \( L \) is the initial length of the rod at bias magnetism \( (H = H_b) \) and prestress \( (\sigma = \sigma_0) \). The coefficient \( \delta \) is introduced to account for the total length of the magnetic circuit (see Figure 3).

Since the magnetic lines flow inside the ferromagnetic materials (soft iron and Terfenol-D) and due to the shape of the actuator, the suggested length of the magnetic field lines in Braghin et al. (2011) is twice the length of the magnetostrictive rod, that is, \( \delta \approx 2 \). Substituting equation (5) to equation (4) results

\[ \sigma = -\frac{e}{s^H} + \frac{nd}{s^H \delta(L + x)} = -\frac{x}{s^H \delta(L + x)} + \frac{nd}{s^H \delta(L + x)} \]

Thus, equation (3) reads

\[ F_m = -K_r x + \frac{dAH}{s^H} \approx -K_r x + \frac{ndA}{s^H \delta L} \]  

where \( K_r = A/Ls^H \) is the mechanical stiffness of the magnetostrictive rod. Note that the calculated magnetic
field using equation (5) is an approximation and therefore may differ from the accurate solution. To verify the approximation, the actuator is modeled in COMSOL as a two-dimensional (2D) axisymmetric model (Figure 4(a)) with the properties listed in Table 2. In this model, the differential form of Ampere’s law is solved numerically in the magnetic circuit and the accuracy of the result is only limited by the mesh size. The mesh size is determined by a mesh convergence analysis in the present case and it is chosen sufficiently small to increase the accuracy of the result.

The computed distribution of the magnetic field using COMSOL at $I = 3$ A is shown in Figure 4(b) and depicts a discontinuity at the end of the Terfenol-D rod, while the magnitude of $H$ inside the rod is very high compared to its casing, which is made of soft iron. This is due to the large difference in relative permeability between soft iron ($\mu_r = 60,000$) and Terfenol-D ($\mu_r = 12$), which causes the magnetic field produced by the input current to be confined inside the Terfenol-D rod only. Furthermore, the numerically obtained magnetic field inside the rod is twice as large in magnitude as the magnetic field obtained using equation (5). The main reason appears to be the proposed path length of the magnetic lines in Braghin et al. (2011) and this explains the discrepancy in the result (see Figure 3). Therefore, the actual magnetic field generated inside the Terfenol-D rod is simply equal to the magnetic field produced by a solenoid and equation (5) should be used with $\delta = 1$ (see Figure 5). Nevertheless, numerically obtained $I - H$ curves can also be employed in equation (8) and most likely preferred in the general case, since $\delta$ may have a different value depending upon the material properties of the casing.

As mentioned before, the blocked force can be obtained by constraining the rod to an infinitely rigid support. This is achieved in the simulation by fixing both ends of the rod and solving the magnetomechanical equation (1), which is solved for the Terfenol-D rod only (see Figure 4(a)). The blocked force for the linear model is obtained by substituting $x = 0$ in equation (8). The obtained results are compared in Figure 5(b), and the curves are similar to Figure 5(a) and the results are in agreement for $\delta = 1$. This is also anticipated since COMSOL uses the same linearized equation with $s^H$ and $d$ as input parameters. Now, the linearized

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**Figure 4.** (a) 2D axisymmetric model of the actuator in COMSOL Multiphysics and (b) the magnetic field distribution at $I = 3$ A, where the lengths and magnetic field are in millimeter and Ampere/meter, respectively.

**Table 2.** Dimensions and main parameter values of Terfenol-D (the average value is used in the calculation).

| $s^H / 10^{-11}$ | Poisson’s ratio | $d / 10^{-8}$ | $\mu^* / \mu^0$ | Rod length (L) | Radius (R) | Number of turns (n) |
|------------------|----------------|--------------|----------------|----------------|-------------|-------------------|
| 1.2–3.4 (m²/N)  | 0.45           | 1.2–2.8 (m/A)| 3–20           | 50 (mm)        | 3 (mm)     | 1000              |

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equation is validated and the next step is the computation of the output force due to a longitudinal deformation of the rod \( x \). Note that the deformation \( x \) depends on the stiffness of the attached structure and requires an iterative solution that is described in section 5.

4. Proposed design of actuator mounting

To generate an output force and transform it to the normal contact load between UPD and the blade platform, the standard design of a wedge damper (Borrajo et al., 2006) is modified to a thin wall damper and the actuator is placed between the damper walls as shown in Figure 6(b) to (d). Note that a thin wall UPD has a good damping potential of bladed disks as presented in Szwedowicz et al. (2008a). The actuator is fixed on the disk using a rigid support and the disk is assumed to be rigid enough in the radial direction. Three actuator mounting designs (\( dsg_1 \) to 3) and two boundary conditions between the UPD walls and the actuator ends are proposed. The magnetostrictive rod length is 50 mm (Table 2) for \( dsg_1 \) and 2 and it is half of this length for \( dsg_3 \). In the first boundary condition (BC1), there is a frictional contact and in the second boundary condition (BC2), the UPD walls and the actuator ends are bonded. In other words, in the first case, relative motion between the actuator ends and the UPD walls is allowed and therefore the normal contact load on the blade platform has a contribution from the centrifugal load; whereas in the second case, the UPD is fixed with actuator and therefore the normal load is only controlled by the actuator output force.

In the present analysis, the different designs aim at maximizing the transmissibility ratio of the normal load over the output force. Note that with the proposed mounting configurations, normal contact load can only be increased on the blade platform. Therefore, BC1 is potentially applicable where a high normal contact load variation is required on the blade platform such as for a large turbine with low rotational speed and high excitation amplitude. However, in the case of a medium-size bladed disk, BC2 is potentially applicable, where required variation in the normal load is moderate.

5. Adaptive control of normal load using output force

A simplified model of two blades is considered to demonstrate the proposed concept and to compute the resulting normal load \( (N_0) \) between the blade platform and the UPD, as shown in Figure 7(a). The model is meshed with SOLID186 quadratic elements in ANSYS® and consists of 9834 nodes. The UPD is designed to accommodate the actuator and meshed separately with 4763 nodes. The material properties of the structural steel \((E = 200 \text{ GPa} \text{ and } \rho = 7850 \text{ kg/m}^3)\) is employed for both bladed disk and UPD. The centrifugal force \( F_c \), which depends on UPD mass, angular speed, and radius, is chosen to be 1200 N in this analysis for BC1 and \( F_c = 0 \) for BC2. A representative value of \( F_c \) is chosen for BC1 and it can significantly vary with angular speed. Three examples are discussed here to analyze the variation in \( N_0 \) and pressure distribution on the platform due to the variation in the input parameters.

(a) In the first example, the input current, \( I \), is varied from 0 to 3 A and its influence on the change in the \( N_0 \) is discussed.

(b) In the second example, the effect of friction coefficient on the \( N_0 \) is analyzed for a constant current.
In the last example, the input current and the friction coefficients are varied and their influence on the pressure distribution and the contact status of the UPD are analyzed.

5.1. Influence of input current on the normal load

The input current is altered from 0 to 3 A and its influence on $N_0$ is analyzed in this example. The friction coefficient ($\mu$) between the blade platform and the UPD is set to 0.3. Computation of $N_0$ is performed in two steps for BC1 and the first step is omitted for BC2, since the centrifugal force is zero in the latter case. In the first step, the nonlinear static contact analysis is performed using the centrifugal force only and the reference results are stored (see Figure 7(b)). After performing the first step, it is assumed that the actuator ends are in contact with the UPD walls and in the second step, instead of considering a full nonlinear contact analysis together with the actuator, only the output force generated by the actuator is considered (see Figure 7(c)). This is a realistic assumption, since actuator transmits the output force only along the rod axis and the tangential motion between UPD walls and the actuator ends is assumed to be negligible. However, the output force $F_m$ depends on the deformation $x$ of the rod as described before (see equation (8)). Therefore, to compute the exact value of $F_m$, the displacement of the actuator contact nodes is monitored as shown in Figure 7(c). The computed deformation $x$ due to the force $F_m$ in the second step should satisfy equation (8), and thus, it requires 2–3 iterations to compute the final value of $F_m$.

The resulting output force and the normal load on the blade platform as a function of input current for different mounting designs are depicted in Figure 8(a) and (b) for BC1 and BC2, respectively. The obtained relationship between the input current and the normal load is almost linear despite a nonlinear friction contact existing between UPD and the blade platform, which is a quite beneficial result from the computational perspective. For BC1, the computed normal load on the blade platform due to the centrifugal load only ($N_{01}^{ref}$) is 653 N and the change in the normal load due to the mounting $dsg1$ ($N_{01}^{dsg1}$) is approximately 200 N at $I = 3$ A. The change is adequate for the proposed application; however, it is relatively small compared to the other mounting designs ($dsg2$ and 3), since the horizontal output force is converted to the normal load that
is substantially influenced by the UPD stiffness, which is anticipated as well. The change in the normal load may be further increased by designing a stiffer UPD compared to the actuator stiffness for this design. Furthermore, a significant change in the normal load is obtained due to the mounting designs $dsg_2$ (757 N at $I = 3$ A) and $dsg_3$ (584 N at $I = 3$ A). The change in the normal contact load is substantially large in these cases, since the output forces are directed toward the centrifugal force and the normal load for the mounting designs $dsg_2$ and $dsg_3$, respectively.

For BC2, the change in the normal load due to the mounting $dsg_1$ is negligible, and therefore, this design is not recommended for a practical application. The change in the normal load due to the mounting designs $dsg_2$ and $dsg_3$ are 760 and 585 N, respectively, at $I = 3$ A and the obtained values are the same as for BC1. In other words, the change in the normal load is

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**Figure 7.** (a) Simplified model of two blades with UPD, (b) UPD with the centrifugal force ($F_c$), and (c) UPD with the centrifugal force and the output force for the actuator mounting $dsg_3$.

**Figure 8.** Variation in the normal load ($N_0$) and the output force ($F_m$) with input current ($I$) at $\mu = 0.3$: (a) for BC1 and (b) for BC2. $N_{01}^{ref}$ represents the normal load at the blade platform due to the $F_c$ only and $N_{01}^{dsg1}$ represents the normal load due to the $F_c$ and $dsg_1$ for BC1. Similarly, $N_{02}^{dsg1}$ represents the normal load due to the $dsg_1$ for BC2 and other notations are interpreted in the same way.
independent of $F_c$ for the mounting designs $dsg2$ and $3$, but it is influenced by $F_c$ for the $dsg1$.

For BC1, the design configuration $dsg1$ is the most preferable from the implementation point of view, since the radial and the axial motions of the UPD will not change the contact condition between the actuator ends and the UPD wall, but a smaller change in the normal load is transmitted to the blade platform. The other mounting configurations ($dsg2$ and $dsg3$) can provide a significant change in the normal load at the blade platform. However, they require the same contact condition between UPD and actuator before and after applying the centrifugal load. This may be a challenge in practice. The selection of the mounting design and the boundary condition depends on the dynamics of the bladed disk and the required variation in the normal contact load. Nevertheless, BC1 should be potentially employed where a high normal load is required to achieve the optimum damping in the system and the BC2 (with $dsg2$ and $3$) is recommended for a low normal load case.

The transmissibility ratio of the normal load over the output force is presented in Figure 9(a) and (b) for BC1 and BC2, respectively. In both cases, mounting $dsg3$ has the highest transmissibility ratio and $dsg1$ has the lowest value for the range of input current. The results are in agreement with the previous discussions, and the obtained curves are horizontal for mounting $dsg2$ and $dsg3$. This means that the transmissibility ratio is independent of the input current, and therefore, the normal contact load can be determined without performing a nonlinear contact analysis if the ratio is known for one current value. Note that the line between 0 and 0.5 A has a slope since all forces are zero at $I = 0$ A. Furthermore, the transmissibility ratio for mounting $dsg1$ with BC1 increases with input current that indicates that mounting $dsg1$ will have a better performance at high input current.

5.2. Influence of the friction coefficient

To analyze the effect of the friction coefficient on the normal contact load, the nonlinear static contact analysis is performed at a constant input current, $I = 3$ A. The corresponding blocked force for $dsg1$ and $dsg2$ is 2000 N and for $dsg3$ is 1000 N since the magnetostrictive rod is half of the original length in this case (Figure 6 and Table 2). The friction coefficient is altered from 0.01 to 1 and other parameters and the boundary conditions are the same as in the previous example.

The computed response curves are presented in Figure 10(a) and (b) for BC1 and BC2, respectively. In contrast to the previous example, the obtained curves are nonlinear in nature. Moreover, the normal contact load is significantly influenced by the friction coefficient and decreases as the friction coefficient grows. For example, the normal contact load ($N_{ref}$) due to the centrifugal force only reduces by 50% (840–425 N) as the friction coefficient changes from 0.01 to 1 (see Figure 10(a)). This happens mainly due to the emergence of the tangential contact force, and therefore, the amplitude of the normal load decreases and the total contact force on the blade interface also changes its direction (see Figure 11). These findings indicate that a change in the friction coefficient could have a significant influence on the nonlinear dynamic response of the system. Furthermore, the other contact interface parameters, such as the normal and the tangential contact stiffnesses (not investigated in this article), also change significantly due to vibration since the vibration leads to a change in the contact condition and contact load between the blade platform and UPD (Asai and Gola, 2015), and its estimation is also a challenging task (Koh and Griffin, 2006). A change in the tangential contact stiffness and the friction coefficient due to the wear cycles is presented in Botto et al. (2012), and the variation in the interface parameters with contact load, contact area, and
temperature is investigated in Schwingshackl et al. (2012). All these physical phenomena manifest a need to control the normal load at the interface adaptively such that the vibrations of the bladed disk can be minimized despite a variation in the contact interface parameters and contact conditions. A typical variation in the friction coefficient values for steel-to-steel contact is 0.3–0.7. In this range of variation, the change in the normal load is of the order of 200 N and this variation is necessary to take into account while optimizing the friction damping.

The normal contact load \( (N_{01}) \) due to the \( F_c \) and \( d_{sg1} \) is almost independent of the friction coefficient values (see Figure 10(a)). This is primarily due to the UPD design and the mounting configuration. Nevertheless, a change in the friction coefficient will influence the dynamic nonlinear forced response of the bladed disk even though \( N_0 \) is constant on the blade platform. Consequently, to attain the optimum damping in the system, the normal contact load is required to either increase or decrease on the interface and that can be achieved by changing the input current in the actuator. It should be emphasized here that the goal is not to keep \( N_0 \) constant on the friction interface. The main aim is to achieve the maximum possible damping in the system by altering the value of \( N_0 \) under different operating conditions. In conclusion, the above results and discussions demonstrate that a change in the friction coefficient will alter the nonlinear dynamic response of the bladed disk that can be potentially controlled by varying the current in the actuator.

### 5.3. Influence of input current and friction coefficient on the pressure distribution

In this example, the pressure distribution and the contact status of UPD are analyzed for varying input current and friction coefficient. All the parameters are the same as in the first example and only BC1 is investigated in this case. The resulting pressure distribution and the contact status of UPD due to \( F_c \) and \( d_{sg1} \) are depicted in Figures 12 and 13, respectively. These results are obtained at increasing input currents and at a constant friction coefficient, \( \mu = 0.3 \). A noticeable shift in the pressure distribution is observed when the actuator is turned on and the highest pressure location moves toward the edge of the contact interface from the mid of UPD, which is due to the topology of this
design (see Figure 6(b)). Furthermore, an increase in the current amplitude leads to a smaller effective contact area and a higher maximum contact pressure as shown in Figure 12(c) and (d) that is likely to alter the tangential contact stiffness as presented in Asai and Gola (2015), Filippi et al. (2004) and Siewert et al. (2006) and discussed before. Therefore, the proposed solution will not only change the normal contact load at the blade platform but may also influence the contact interface parameters and thus the nonlinear response curve. Nevertheless, these variations can be countered by adjusting the input current amplitude in the actuator and thus vibratory stresses in the bladed disk can be minimized. A change in the contact status of the UPD is depicted in Figure 13, which is similar to Figure 12.

The obtained pressure distribution and the contact status of the UPD due to $F_c$ and $dsg$ are depicted in Figures 14 and 15, respectively. These results are computed at increasing value of the friction coefficient and at a constant input current, $I = 3$ A. The shift in the pressure distribution is not as drastic as in the previous case, and maximum pressure on the UPD decreases as friction coefficient increases. This is in line with the previous result (Figure 11) where the normal load decreases as friction coefficient increases due to the emergence of the tangential contact force. Furthermore, the contact area is almost constant with the increase in the friction coefficient, which is due to a constant input current, and the output force direction is toward the normal of the contact interface. Nevertheless, a part of UPD interface goes into sticking state at $\mu = 1$, which is as expected.

6. Conclusion

A novel application of the magnetostrictive actuator in controlling the normal contact load at the friction interface of a bladed disk is proposed. This is achieved by constraining the output rod of the actuator between the walls of the UPD. The computed results reveal that a change in normal load as high as 750 N can be obtained by properly designing the actuator mounting. It means that the change in the normal load can be from 0 to 750 N by varying the input current from 0 to 3 A in the actuator circuit, and thus, an optimum damping in the system can be achieved despite a variation in the contact interface parameters during vibration. Moreover, two boundary conditions (BC1 and BC2) between the actuator ends and the UPD walls are proposed, where the BC1 is potentially applicable where a high normal load is required to achieve the optimum damping in the system and the BC2 is recommended for a low normal load case. Furthermore, it is demonstrated that the friction coefficient has a significant influence on the normal contact load and thus on the nonlinear vibration prediction of the bladed disk with friction contact. Finally, it is possible to damp several resonances at different
Figure 13. Variation in the contact status of the UPD with input current caused by $F_c$ and dsg1: (a) $I = 0$ A, (b) $I = 1$ A, (c) $I = 2$ A, and (d) $I = 3$ A at $\mu = 0.3$ with BC1.

Figure 14. Variation in the pressure distribution on the UPD with input current caused by $F_c$ and dsg3: (a) $I = 0$ A, (b) $I = 1$ A, (c) $I = 2$ A, and (d) $I = 3$ A at $\mu = 0.3$ with BC1, where the pressure is in Pascal.
operating conditions by altering the normal load at the friction interface.

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**References**

Afzal M, Arteaga IL and Kari L (2016) Investigation of damping potential of strip damper on a real turbine blade. In: Proceedings of ASME turbo expo 2016, Seoul, South Korea, 13–17 June, Paper GT2016-57230.

Asai K and Gola MM (2015) Experimental verification of friction behaviors under periodically-varied normal force by developing a two-directional friction test system. In: ASME turbo expo: power for land, sea, and air—structures and dynamics, vol. 7B, Quebec, QC, Canada, 15–19 June.

Borrajo JM, Zucca S and Gola MM (2006) Analytical formulation of the Jacobian matrix for nonlinear calculation of the forced response of turbine blade assemblies with wedge friction dampers. International Journal of Non-Linear Mechanics 41: 1118–1127.

Botto D, Lavella M and Gola MM (2012) Measurement of contact parameters of flat-on-flat contact surfaces at high temperature. In: Proceedings of ASME turbo expo 2012, Copenhagen, 11–15 June, Paper GT2012-69472.

Braghin F, Cinquemani S and Resta F (2011) A model of magnetostrictive actuators for active vibration control. Sensors and Actuators A 165: 342–350.

Calkins FT, Dapino MJ and Flatau AB (1997) Effect of prestress on the dynamic performance of a Terfenol-D transducer. In: Proceedings of SPIE, symposium on smart structures and materials, San Diego, CA, 6 June.

Calkins FT, Flatau AB and Dapino MJ (2007) Overview of magnetostrictive sensor technology. Journal of Intelligent Material Systems and Structures 18: 1057–1066.

Clark AE (1980) Chapter 7: magnetostrictive rare earth-Fe2 compounds. In: Wohlfarth EP (ed.) Handbook of Ferromagnetic Materials. Amsterdam: Elsevier, pp. 531–589.

COMSOL (2016) COMSOL Multiphysics 5.2a. Available at: https://www.comsol.se/events?gclid=CJyuwuXFydiCFV

Dapino MJ (2004) On magnetostrictive materials and their use in adaptive structures. Structural Engineering and Mechanics 17: 303–329.

Dapino MJ, Calkins FT and Flatau AB (1996) Measured Terfenol-D material properties under varied applied magnetic

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**Figure 15.** Variation in the contact status of the UPD with input current caused by $F_c$ and $d_{sg}$: (a) $I = 0$ A, (b) $I = 1$ A, (c) $I = 2$ A, and (d) $I = 3$ A at $\mu = 0.3$ with BC1.
field levels. In: *Proceedings of SPIE, symposium on smart structures and materials*, San Diego, CA, 1 May.

Dapino MJ, Smith RC and Flatau AB (2000a) A coupled structural-magnetic strain and stress model for magnetostriuctive transducers. *Journal of Intelligent Material Systems and Structures* 11: 135–152.

Dapino MJ, Smith RC and Flatau AB (2000b) A structural-magnetic strain model for magnetostriective transducers. *IEEE Transactions on Magnetics* 36: 545–556.

Dapino MJ, Smith RC, Calkins FT, et al. (2002) A coupled magnetomechanical model for magnetostriective transducers and its application to Villari-effect sensors. *Journal of Intelligent Material Systems and Structures* 13: 737–747.

Filippi S, Akay A and Gola MM (2004) Measurement of tangential contact hysteresis during microslip. *ASME Journal of Tribology* 126: 482–489.

Griffin JH (1980) Friction damping of resonant stresses in gas turbine engine airfoils. *Journal of Engineering for Power* 102: 329–333.

Grunwald A and Olabi AG (2008) Design of a magnetostriective (MS) actuator. *Sensors and Actuators A* 144: 161–175.

Gu G, Li Z, Zhu L, et al. (2013) A comprehensive dynamic modeling approach for giant magnetostriective material actuators. *Smart Materials and Structures* 22(12): 125005.

Huang W, Wang B, Sun Y, et al. (2007) Investigation on dynamic properties of Terfenol-D actuators. In: *Proceedings of SPIE, international conference on smart materials and nanotechnology in engineering*, vol. 6423, Harbin, China, 1 November.

Jiles DC and Atherton DL (1986) Theory of ferromagnetic hysteresis. *Journal of Magnetism and Magnetic Materials* 61: 48–60.

Kaltenbacher M, Meiler M and Ertl M (2009) Physical modeling and numerical computation of magnetostriction. *The International Journal for Computation and Mathematics in Electrical and Electronic Engineering* 28(4): 819–832.

Kellogg R and Flatau A (1999) Blocked force investigation of a Terfenol-D transducer. In: *Proceedings of SPIE, symposium on smart structures and materials*, Newport Beach, CA, 9 June.

Koh KH and Griffin JH (2006) Dynamic behavior of spherical friction dampers and its implication to damper contact stiffness. In: *Proceedings of ASME turbo expo 2006*, Barcelona, 8–11 May, Paper GT2006-90102.

Laxalde D, Thouerez F and Lombard JP (2010) Forced response analysis of integrally bladed disks with friction ring dampers. *ASME Journal of Vibration and Acoustic* 132: 011013.

Laxalde D, Thouerez F, Sinou JJ, et al. (2007) Qualitative analysis of forced response of blisks with friction ring dampers. *European Journal of Mechanics—A/Solids* 26: 676–687.

Lesaffre N, Sinou JJ and Thouerez F (2007) Contact analysis of a flexible bladed-rotor. *European Journal of Mechanics—A/Solids* 26(3): 541–557.

Olabi AG and Grunwald A (2008) Design and application of magnetostriective materials. *Materials and Design* 29: 469–483.

Petrov EP and Ewins DJ (2002) Analysis of nonlinear multi-harmonic vibrations of bladed disks with friction and impacts dampers. In: *Proceedings of the 7th national turbine engine high cycle fatigue (HCF) conference*, Palm Beach Gardens, FL, 14–17 May.

Petrov EP and Ewins DJ (2006) Effects of damping and varying contact area at blade-disk joints in forced response analysis of bladed disk assemblies. *ASME Journal of Turbomachinery* 128: 403–410.

Petrov EP and Ewins DJ (2007) Advanced modeling of under-platform friction dampers for analysis of bladed disk vibration. *ASME Journal of Turbomachinery* 129: 143–150.

Rajapati K, Greenough R and Wharton A (1996) Effect of cyclic stress on Terfenol-D. *IEEE Transactions on Magnetics* 32(5): 4761–4763.

Sanliturk KY, Ewins DJ and Stanbridge AB (2001) Under-platform dampers for turbine blades: theoretical modeling, analysis and comparison with experimental data. *ASME Journal of Engineering for Gas Turbines and Power* 123: 919–929.

Schwingshackl CW, Petrov EP and Ewins DJ (2012) Measured and estimated friction interface parameters in a nonlinear dynamic analysis. *Mechanical Systems and Signal Processing* 28: 574–584.

Siewert C, Panning L, Schmidt-Fellner A, et al. (2006) The estimation of the contact stiffness for directly and indirectly coupled turbine blading. In: *Proceedings of ASME Turbo Expo 2006*, Barcelona, 8–11 May, Paper GT2006-90473.

Srinivasan AV and Cutts DG (1983) Dry friction damping mechanisms in engine blades. *ASME Journal of Engineering for Power* 105: 332–341.

Szwedowicz J, Gibert C, Sommer TP, et al. (2008a) Numerical and experimental damping assessment of a thin-walled friction damper in the rotating setup with high pressure turbine blades. *ASME Journal of Engineering for Gas Turbines and Power* 130: 012502.

Szwedowicz J, Mahler A, Slowik S, et al. (2005) Nonlinear dynamic analyses of a gas turbine blade for attainment of reliable shroud coupling. In: *Proceedings of ASME turbo expo 2005*, 6–9 June, Reno, NV, Paper GT2005-69062.

Szwedowicz J, Visser R, Sextro W, et al. (2008b) On nonlinear forced vibration of shrouded turbine blades. *ASME Journal of Turbomachinery* 130: 011002.

Tan X and Baras JS (2004) Modeling and control of hysteresis in magnetostrictive actuators. *Automatica* 40: 1469–1480.

Wu J, Xie Y, Zhang D, et al. (2012) Experimental friction damping characteristic of a steam turbine blade coupled by shroud and snubber at standstill setup. In: *Proceedings of ASME turbo expo 2012*, Copenhagen, 11–15 June, Paper GT2012-69472.

Yang BD and Meng CH (1997a) Characterization of contact kinematics and application to the design of wedge damper in turbomachinery blading: Part I—stick-slip contact kinematics. *ASME Journal of Engineering for Gas Turbine and Power* 120: 410–417.
Appendix I

A typical variation in the nonlinear forced response curve with normal load is shown in Figure 16. The curves reveal that an increase in the normal load initially leads to a decrease in the response amplitude and then the response amplitude increases. Consequently, the maximum response amplitude of the curve passes through a minimum that is known as optimum damping in the system.