DOI: 10.2478/fas-2022-0006
FATIGUE OF AIRCRAFT STRUCTURES
Volume 2022: Issue 14, pp. 69-82

INSIGHT INTO DAMPING SOURCES IN TURBINES

Grzegorz Moneta,
Łukasiewicz Research Network, Institute of Aviation,
Al. Krakowska 110/114 Warsaw, Poland

grzegorz.moneta@ilot.lukasiewicz.gov.pl

Abstract

Blade vibrations in aircraft engines are a significant challenge that must be overcome during the design and development of modern turbine engines. Vibrations lead to cyclic displacements and result in alternating stress and strain in undesired environments (high temperatures, erosion, corrosion of the surface, etc.). Under resonance conditions, stress amplitudes can increase and exceed their safety limits, and in extreme cases, can lead to engine failure. One method to reduce resonance vibrations is to increase damping in the turbine assembly. This paper presents and describes vibration damping sources in the turbine, including aerodynamic, material, and friction damping. Additionally, typical damping values for each damping component are presented and compared.

Keywords: vibrations, blade, gas turbine, turbine engine, damping, FEM, Finite Element Method, transient analysis, explicit, friction damping, under-platform damper, optimization, sensitivity analysis
Article Category: research article

INTRODUCTION

The occurrence of vibrations in turbine blades is still one of the most serious difficulties today, which must be solved during the process of designing and improving a modern turbine engine (Moneta, et al., 2021). These vibrations lead to cyclic displacements, and thus also to cyclic stresses and strains, occurring in unfavorable conditions (elevated temperatures, erosion and corrosion of the material surface, active environment). The consequences of these phenomena include instability of the machine operation and accelerated wear of its components, mainly as a result of material fatigue and fretting (Jachimowicz, et al., 2000; Jachimowicz, et al., 2011; Wdowiński, et al., 2017). The extreme effect of vibrations is crack initiation and propagation, which can lead to blade loss (Figure 1). This phenomenon leads to damage to downstream stages of the rotor assembly by impacts of detached fragments of blades. Engine damage can in many cases lead to destruction of the aircraft (especially for single-engine aircraft).

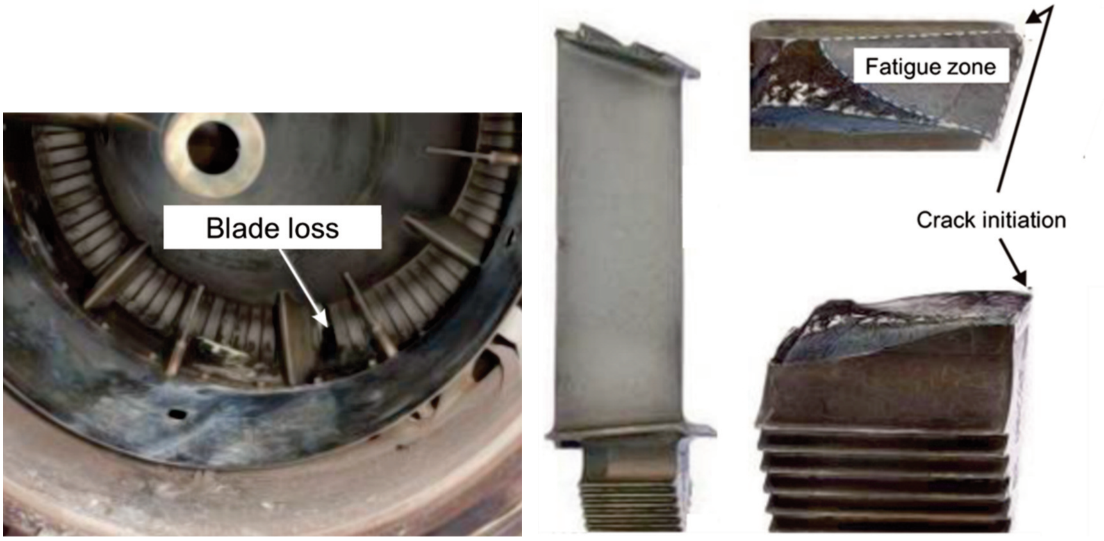

Figure 1. Turbine blade loss caused by HCF (DFS, 2014). HCF, high cycle fatigue.

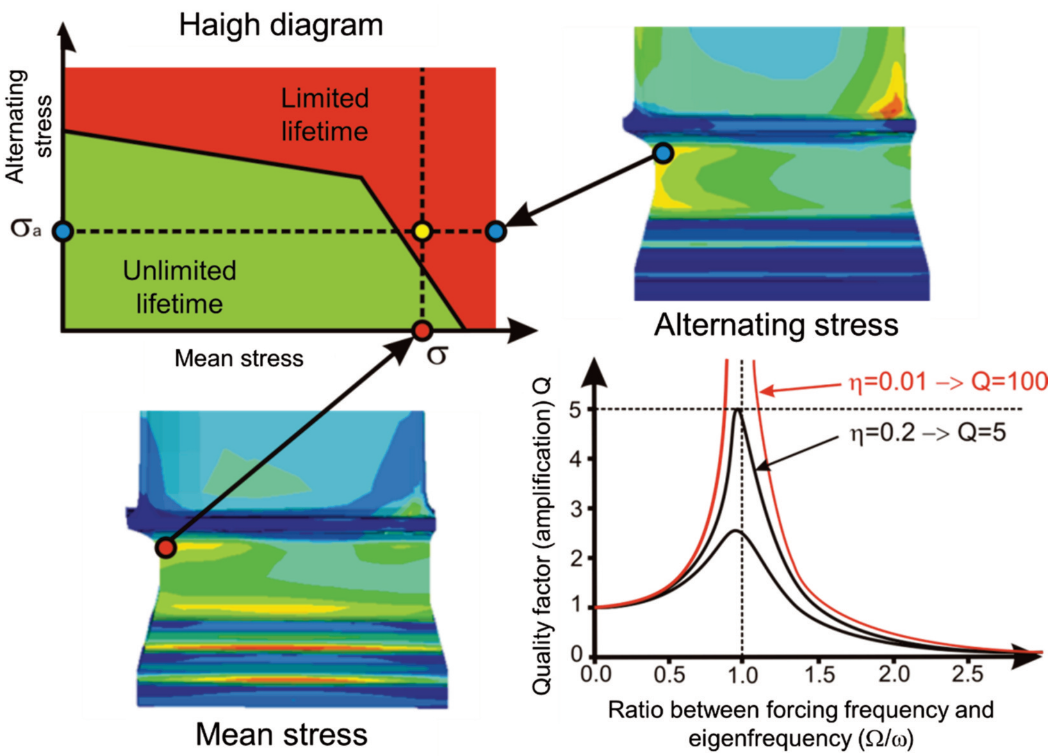

Figure 2. Scheme of blade fatigue assessment using the Haigh diagram.

There are several methods to increase blade fatigue resistance, which include as follows:

- reducing the amplitude of the excitation force (in the case of vibrations of lower frequencies by limiting the casing ovality by design, and for higher vibration frequencies by increasing the distance between rotating and stationary blade rows);

- changing the excitation frequency (by changing the number of blades and vanes in front of and behind the blade row or by modifying the characteristic rotational speeds of the rotor);
- changing of resonant frequencies of blades (optimization of natural frequencies);
- reducing the scatter of blade dimensional deviations (increasing the manufacturing accuracy) in order to narrow the dispersion of their natural frequencies;
- reduction of dynamic and static stresses in places of their concentration (shape optimization);
- increasing damping (e.g., by optimizing dampers, using material with higher damping, increasing the number of friction surfaces in locks and their optimization, etc.);
- shaping the contact surface of the blade with the disk in order to avoid fretting.

Each of the above-mentioned methods leads to transfer of a combination of static stresses (usually the maximum principal stress) and dynamic stresses into the area of infinite fatigue life (or at least as close as possible to this area) – Figure 2.

DESCRIPTION OF DAMPING SOURCES IN THE TURBINE

Gas turbine blades are characterized by very low damping and therefore, when passing through resonances by critical rotational speeds, the stress amplitudes are easily amplified by even more than a 100 times (Rao, 2011). This sometimes leads to very sudden failures and plane crashes caused by high cycle fatigue (HCF). For this reason, damping assessment is an important part of blade design. It should be highlighted that the resultant damping is not the same for all blades of a given row – it depends, among others, on the following:

- mode shape;
- rotational speed (and related to this, centrifugal load, pressure load, and temperature);
- manufacturing tolerances;
- assembly tolerances;
- contact conditions on shroud, sealings, and friction dampers;
- contact surfaces (area, state);
- material imperfections.

In addition, damping changes over time due to frictional wear, fretting, oxidation, corrosion, aging, and creep. Sometimes the phenomenon of blocking or welding of elements appears, which prevents frictional energy dissipation in locks and dampers, reducing the resultant damping of the assembly. The measurements of the damping of the blades are always characterized by dispersion of their values between parts, and it depends on the quality and repeatability of components (e.g., for SO-3 turbine engine, the measured damping ratio of the turbine blades without friction dampers were in the range of $0.66\% < \zeta < 1.08\%$, giving amplification $46 < Q < 76$) (Klepacki, 1978). For these reasons, any reliable solution to increase turbine damping is highly desirable. Several basic components of blade damping can be distinguished (Figure 3).

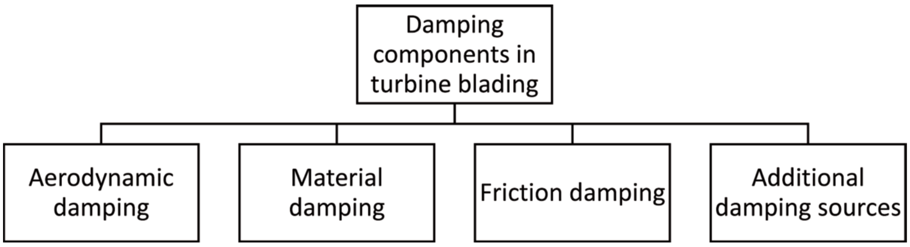

Figure 3. Damping components in turbine blading.

Resultant damping can be determined from the vibration measurements (e.g., using half-power bandwidth method). In the case of a linear nature of all damping components, the resultant value can be calculated as the sum of individual damping component (aerodynamic, material, in the lock, etc.) (Salzmann & van der Tempel, 2005). In the case of a turbine assembly, the damping components have a nonlinear character and the summation method cannot be used here.

Aerodynamic damping

Aerodynamic damping is related to the motion of the blade in the direction perpendicular to the flow of the medium (here exhaust gases). Determining the value of aerodynamic damping is difficult and is usually affected by high error. It can be done using analytical formulas (Hanson et al., 1953; Klepacki, 1975), computational fluid dynamic (CFD) methods (Lampert et al., 2004), experimental measurements (Brown, 1981), or estimated based on the damping values achieved in similar engines.

Klepacki and Pastorius (Klepacki, 1975; Pastorius, 1969) propose an analytical method for determining the aerodynamic damping based on an empirical formula from Hanson's paper (Hanson et al., 1953). After conversion from imperial units to SI units and checking, this formula takes the following form:

$$\delta_a = 1{,}12 \cdot \frac{\dot{u}_{g-b} \cdot c_{prof}}{\frac{\rho_m}{\rho_a} \cdot \left(0{,}2288 \cdot A_0 + 1{,}0444 \cdot A_t\right) \cdot f}, \tag{1}$$

$$\zeta_a = \frac{\delta_a}{\sqrt{4\pi^2 + \delta_a^2}}, \tag{2}$$

where:

δ_a [–] – logarithmic decrement of aerodynamic damping;

ζ_a [–] – aerodynamic damping ratio;

$\dot{u}_{g-b}\left[\frac{m}{s}\right]$ – relative velocity between exhaust gas and blade profile;

$c_{prof}\left[m\right]$ – chord length of the blade profile;

$\rho_m \left[\frac{kg}{m^3} \right]$ – density of blade material;

$\rho_g \left[\frac{kg}{m^3} \right]$ – average gas density;

$A_0 \left[m^2 \right]$ – cross-sectional area of the profile at the hub height;

$A_t \left[m^2 \right]$ – cross-sectional area of the profile at the tip height;

$f \left[\frac{1}{s} \right]$ – frequency of the first mode of the blade.

This formula applies to flutter-free flow. Below is presented the aerodynamic damping ratio comparison between the calculated value for SO-3 turbine and the values found in the literature (Table 1). The experimental values of aerodynamic damping were determined by using the half-power bandwidth method, comparing values obtained in vacuum and air. The applied empirical formula gave the value of the aerodynamic damping ratio in the range observed for similar structures.

Table 1. Comparison of the aerodynamic damping ratio of turbine blades

Object	**Aerodynamic damping ratio** ζ_a [-]	**Source**
SO-3 turbine blades (free-standing)	0.14% (calculated)	Formulas (Eqs. 1 and 2)
High pressure turbine blades of Honeywell TFE 731-2 engine (free-standing)	0.14%–0.26% (±0.11%)	Kielb & Abhari, 2001
Steam turbine blades (free-standing)	0.21%–0.40%	Brown, 1981

Material damping

Material damping is related to the elastic deformation of the material and so-called internal friction causing hysteresis effect (Figure 4). These changes are irreversible and cause heat generation.

Strains occur throughout the volume of the element, but their distribution is uneven. Material attenuation is defined in the literature in several ways, including the following:

- material damping loss factor η_m;
- material damping ratio ζ_m;
- logarithmic decrement of material damping δ_m;
- parameters of material damping J_m and n_m.

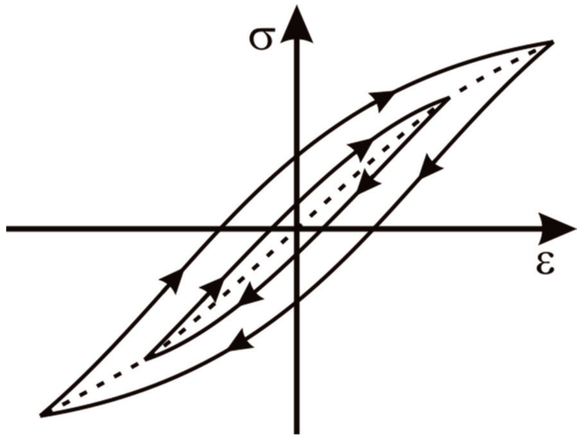

Figure 4. An example of a hysteresis loop with cyclic deformation of the material.

Material damping ratio can be determined using the following formula (Lazan, 1968):

$$\zeta_m = \frac{\int_V J_m \left(\sigma_a\right)^{n_m} dV}{2\pi W_V}, \tag{3}$$

where:
s_a – alternating stress,
W_V – maximum potential energy of the system in a cycle.

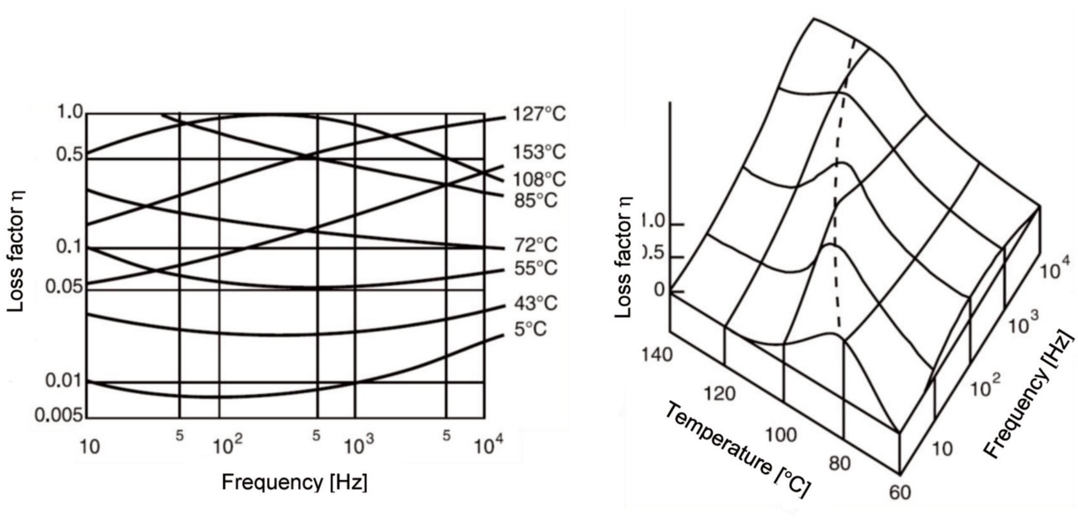

Figure 5. Example of temperature and vibration frequency influence on material damping (Braun et al., 2002).

The methods for determining the material damping parameters are standardized and must be carried out in vacuum, under strictly defined conditions (ANSI, 1998; ASTM, 1998; DIN, 1971; ISO, 1991).

Material damping depends on the following:

- type of material and its condition;
- average stresses and stress amplitude;
- temperature;
- frequency of vibrations.

Examples of the influence of temperature, vibration frequency, and stress amplitude on material damping are presented below (Figures 5 and 6).

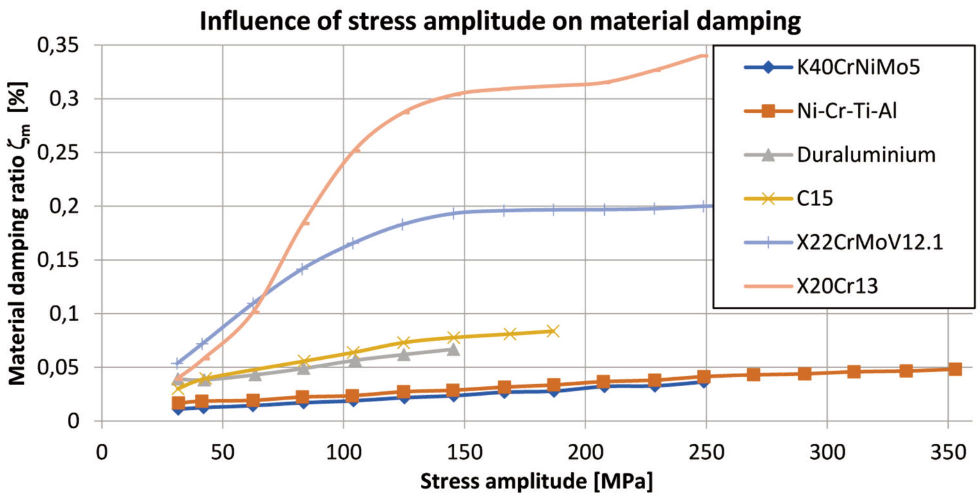

Figure 6. Influence of stress amplitude on material damping of selected alloys at room temperature (Szmid, 1962; Szwedowicz, 2012).

Friction damping

Friction damping results from energy dissipation due to friction in the turbine assembly. It occurs at the contact of the surfaces of movable connections, mainly in the following:

- roots;
- friction dampers;
- coupling elements – on contact surfaces of shrouds, snubbers, lacing wires, bolts, and rods (depending on the design).

Friction damping has a highly nonlinear character and depends mainly on the following:

- materials (friction coefficient, contact stiffness);
- temperature;
- wear of contact surfaces;
- manufacturing and assembly tolerances (especially for damping in the root);
- rotational speed (changes in contact pressure and force required to change stick-slip status).

Many methods of friction damping analysis are commonly used, which are based on different friction contact models (Figure 7) (Csaba, 1998; Srinivasan, 1997), among others:

- rigid contact;
- macro-slip (assuming an even distribution of pressures in the combination and simultaneous transition of the entire contact surface into contact or slip);
- micro-slip (assuming uneven distribution of pressures in the combination and uneven and progressive transition of the contact surface into contact or slip).

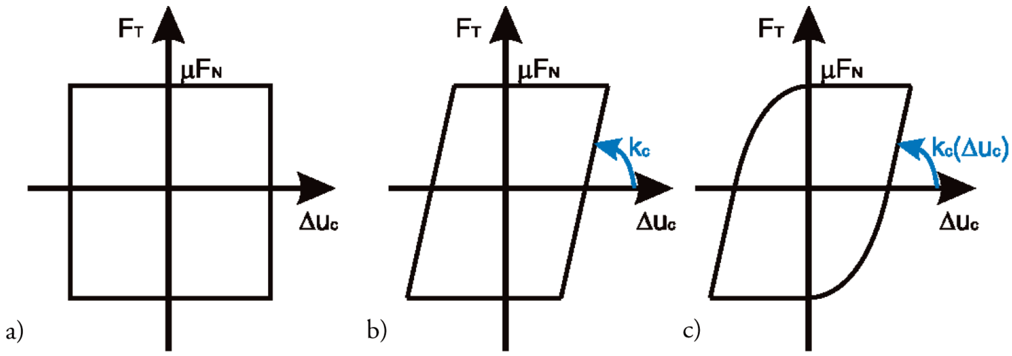

Figure 7. Friction models: a) rigid contact, b) macro-slip, c) micro-slip.

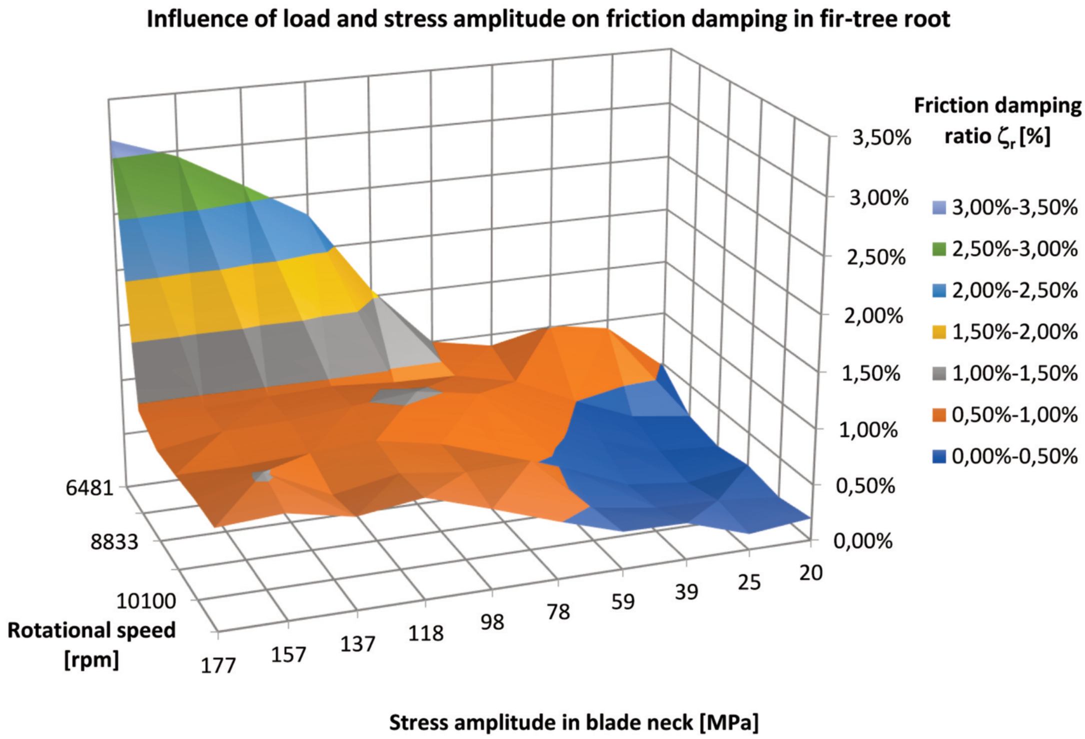

Figure 8. Influence of load and stress amplitude on friction damping in fir-tree root (Klepacki, 1975).

Friction damping in the turbine lock has a strongly nonlinear character. On the basis of the results of measurements of this parameter on one blade – groove pair of fir-tree lock, the damping in the lock reached the values 0.12% < ζ_r < 2.95% (0.737% < δ_r < 18.55%) (Klepacki, 1975). The results of the measurements of the impact of the load and the vibration amplitude on the friction damping in an example of fir-tree lock are presented below (Figure 8). As the load increases, the contact conditions change: individual teeth successively come into contact and its actual surface changes as well as the values and distribution of contact pressure. This results in sticking contact, which reduces damping in the lock. With increase in the stress amplitude, the damping also increases, which is caused by greater relative displacements in the contact pairs (greater dissipation of energy caused by friction).

Damping in fir-tree locks is characterized by a fairly large dispersion of values caused by the manufacturing tolerances of each pair. The isochromatic patterns in two similar fir-tree locks subjected to the same load are shown in Figure 9. There are significant differences in the distribution of contact pressures.

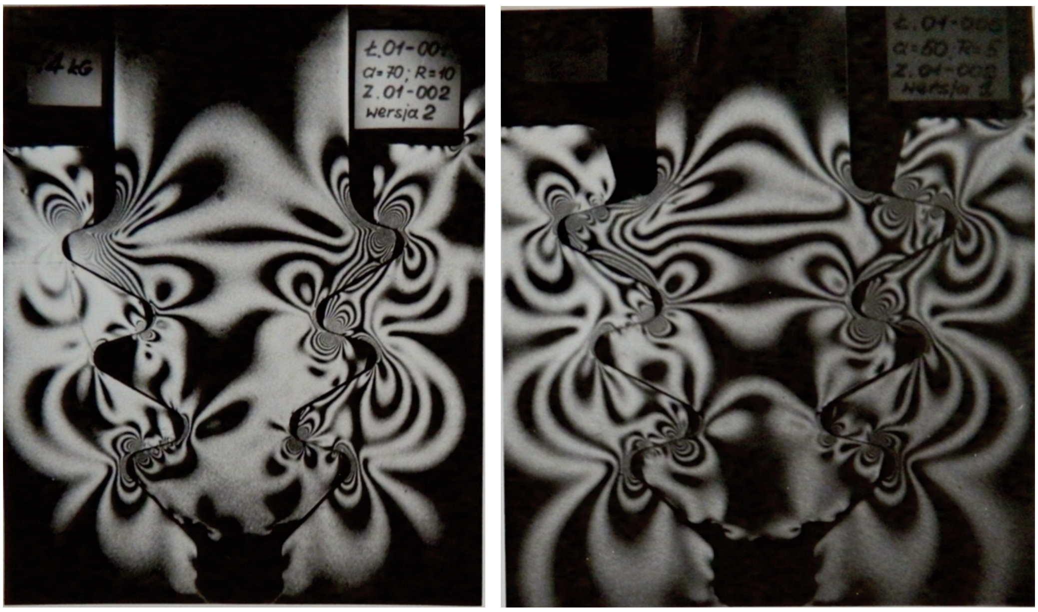

Figure 9. Isochromatic pattern in two different pairs of fir-tree locks (Klepacki, 1975).

Friction damping in the damper has a fundamental influence on the stress amplitude in the resonance. The impact of the use of under-platform dampers in SO-3 turbine assembly is presented in Figure 10. A significant reduction in the stress level in resonance is visible, making the structure safer and more durable. The use of a damper also increases the values of the natural frequencies of the blades, allowing in the case of the discussed turbine assembly structure to avoid resonance with the 5th harmonic of the engine speed.

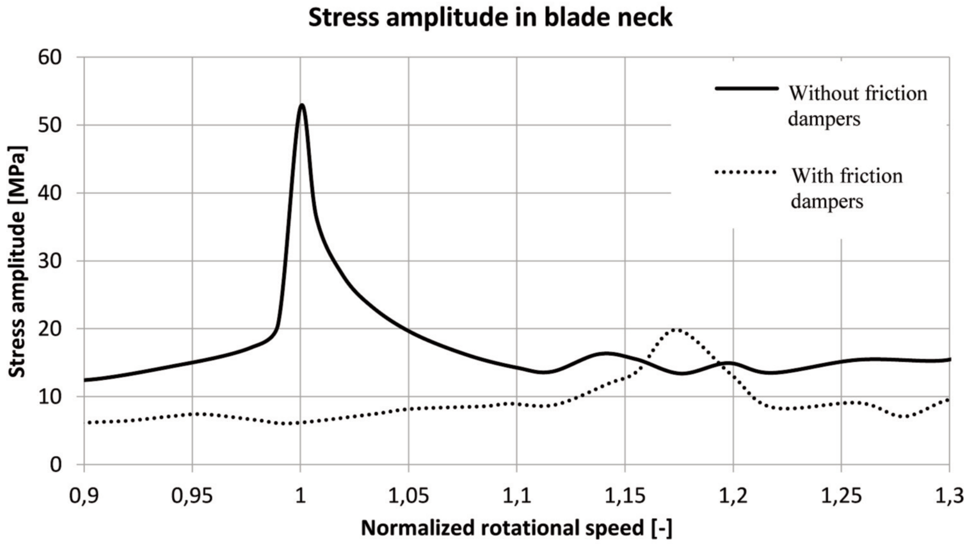

Figure 10. Stress amplitudes in the blade with and without friction dampers as a function of rotational speed (Klepacki, 1978).

The results of using friction dampers in the discussed turbine with other solutions described in the literature are presented in Table 2. Based on the analysis of the achieved reductions in the amplitude values, it can be concluded that the originally used design of the damper is probably not optimal for the discussed design. This gives one the opportunity to improve it and achieve better results.

Table 2. Comparison of the impact of friction dampers on blade vibrations.

Object	**Amplitude reduction (%)**	**Source**
SO-3 turbine (free-standing blades, baseline under-platform dampers)	-62.0	Klepacki, 1975
SO-3 turbine (free-standing blades, optimized under-platform dampers)	-74.0	Moneta, 2019
Axial compressor (free-standing blades, roller under-platform dampers)	-80.0	Hanson, 1956
Volvo RM6/RM8 turbine (free-standing blades, under-platform dampers)	-80.0	Csaba, 1997
Steam turbine (last stage blades coupled by damping bolts)	-83.3	Szwedowicz, 2008

Other damping components

In addition to the already mentioned damping components, other phenomena can also be used to reduce the amplitude of vibrations, including as follows:

- impact damping – energy is dissipated through elastic impacts occurring during vibrations (Jones et al., 1975) (including the phenomenon of detuning, which additionally reduces the amplitudes of displacements in resonance);
- damping related to the following phenomena: piezoelectric, electrostatic, and magnetostriction – these elements, by controlling the appropriate electric (or magnetic) signal, change their volume and counteract vibrations (Braun et al., 2002) (it is possible to use reverse phenomena to generate signals from the deformation of elements and use them to stimulate other active elements within the autonomous dampers);
- eddy current damping – relative displacements of strong magnets and good current conductors cause eddy currents to counter this motion (Laborenz, 2014);
- damping with the use of shape memory materials – the use of phase transformations caused by strains and temperature changes (Braun et al., 2002);
- inherent damping in the unfused powder during the additive manufacturing process (Moneta et al., 2022; Moneta et al., 2022; Scott-Emuakpor et al., 2021).

All these methods have their applicability limitations (related mainly to the operating temperature), but their application in prototypes and technology demonstrators is becoming more and more common. Particularly promising and future-oriented is their use in the creation of integrated, intelligent structures (e.g., in the form of advanced, intelligent damping composites) (Braun et al., 2002).

CONCLUSIONS

To satisfy the market demand for higher performance, modern engines need to have lower mass, operate at higher temperature, and with a larger mass flow, to startup and shutdown more frequently. These requirements lead to increase the HCF degradation of machines (higher excitation, more frequent passing through resonances). The so-called resonance-free design of turbine blading will be more difficult to achieve and these components will be required to design using a resonance-proof approach. To face this problem, damping optimization will be more and more required during the design process. Efficient use of available damping sources in turbines requires better understanding of the occurring phenomena in turbine. Additive manufacturing provides new opportunities for damping optimization, giving additional design space for blade design and providing very promising results in resonance amplitude reduction.

REFERENCES

ANSI. (1998). *Single cantilever beam method for measuring the dynamic mechanical properties of viscoelastic materials*. (S2.23-1998).

ASTM. (1998). *Test method for measuring vibration-damping properties of materials* (E756-98).

Braun, S., Ewins, D. J., & Rao, S. S. (2002). *Encyclopedia of vibration*. (G. Simon, Ed.). Academic Press.

Brown, W. G. (1981). Determination of damping values for turbine blades. *Proceedings of the Design Engineering Technical Conference*, Article ASME Paper 81-DET-131.

Csaba, G. (1998). *Modelling microslip friction damping and its influence on turbine blade vibrations* [PhD dissertation, Linköping University]. http://urn.kb.se/resolve?urn=urn:nbn:se:liu:diva-181151

Csaba, G. & Andersson, M. (1997). Optimization of friction damper weight, simulation and experiments, presented at the ASME TurboExpo 97, Orlando, FL, 97-GT-115, 2-5 June, 1997.

Deutsches Institut fuer Normung. (1971). *Biegeschwingungsversuch. Bestimmung von kenngroessen schwingungsgeda empfter mehrschichtsysteme* (*Flexural vibration test. Determination of parameters of vibration-damped multi-layer systems*) (DIN Standard 53 440).

Director of Flight Safety. (2014). *Canadian forces flight safety investigation report* (CT155201). DND.

Hanson, M. P. (1956). A vibration damper for axial-flow compressor blading. *Procedings of the Society for Experimental Stress Analysis*, *XIV*.

Hanson, M. P., Mayer, A. J., & Manson, S. S. (1953). A method of evaluating loose-blade mounting as a means of suppressing turbine and compressor blade vibration. *Proceedings SESA*, *10*(2).

ISO. (1991). *Damping materials: Graphic presentation of complex modulus* (ISO 10112:1991).

Jachimowicz, J., Karliński, W., & Szachnowski, W. (2000). Blade and disc in-lock co-operation – selected problems. *Machine Dynamics Problems*, *4*(24), 71–86.

Jachimowicz, J., Kozłowski, P., Moneta, G., Szymczyk, E., & Kaniowski, J. (2011). Zjawisko frettingu w konstrukcjach lotniczych. *Prace Instytutu Lotnictwa*, *206*, 36–58.

Jones, D. I. G., Nashif, A. D., & Stargardter, H. (1975). Vibrating beam dampers for reducing vibrations in gas turbine blades. *Transactions of the ASME, Journal of Engineering for Power*, 111–116.

Kielb, J. J., & Abhari, R. S. (2001). Experimental study of aerodynamic and structural damping in a full-scale rotating turbine. *ASME Turbo Expo 2001*, Article 2001-GT-0262. https://doi.org/10.1115/2001-GT-0262.

Klepacki, W. (1975). *Nieklasyczne zagadnienia drgań łopatek turbin lotniczych* [Unpublished PhD dissertation]. Instytut Lotnictwa.

Klepacki, W. (1978). O pewnej metodzie zmniejszania drgań łopatek turbin lotniczych. *Prace Instytutu Lotnictwa*, *74*, 11–40.

Laborenz, J. (2014). *Eddy current damping concept for last stage steel blading*, 19th Blade Mechanics Seminar. Winterthur.

Lampert, P., Szymaniak, M., & Rzadkowski, R. (2004). Unsteady load of partial admission control stage rotor of a large power steam turbine. *Proceedings of the ASME Turbo Expo 2004: Power for Land, Sea, and Air*, *Volume 5: Turbo Expo 2004, Parts A and B*, Article ASME Paper GT2004-53886. https://doi.org/10.1115/GT2004-53886

Lazan, B. J. (1968). *Damping of materials and members in structural mechanics*. Pergamon Press.

Moneta, G. (2019). *Damping optimization of turbine blade vibration* [Unpublished PhD dissertation]. Warsaw University of technology.

Moneta, G., Fedasz, M., Szmidt, M., Cieslak, M., & Krzymien, W. (2022). Advantages of additive manufacturing technology in damping improvement of turbine blading. *Proceedings of the 2022 International Additive Manufacturing Conference. 2022 International Additive Manufacturing Conference*, Article IAM2022-96752. https://doi.org/10.1115/IAM2022-96752.

Moneta, G., Fedasz, M., Szmidt, M., Cieslak, S., & Krzymien, W. (2022). Increasing of damping in the turbine blade through multi-functional design and advantages of additive manufacturing technology. *Proceedings of the ASME Turbo Expo 2022: Turbomachinery Technical Conference and Exposition, Volume 8A: Structures and Dynamics — Aerodynamics Excitation and Damping; Bearing and Seal Dynamics*, Article GT2022-83889. https://doi.org/10.1115/GT2022-83889.

Moneta, G., Jachimowicz, J., Pietrzakowski, M., Doligajski, A., & Szwedowicz, J. (2021). Insight into vibration sources in turbines. *Fatigue of Aircraft Structures*, *2021*(13), 40–53. https://doi.org/10.2478/fas-2021-0005.

Pastorius, W. J. (1969). *Damping factors in turbine blade vibration* (Publication No. 6580) [Electronic Theses and Dissertations, University of Windsor]. https://scholar.uwindsor.ca/etd/6580

Rao, J. S. (2011). *History of mechanism and machine science*. Volume 20. Springer.

Salzmann, D.J.C. & van der Tempel, J. (2005). Aerodynamic damping in the design of support structures for offshore wind turbines. Offshore Wind Energy Conference, Copenhagen, Dennmark, 26-28 October, 2005.

Schmid, R. (1962). Resonanzverhalten und Schwingungssicherheit der Schaufeln von Turbomaschinen. *Maschinenbautechnik*, 11(Heft 12).

Scott-Emuakpor, O., Beck, J., Runyon, B., & George, T. (2021). Determining unfused powder threshold for optimal inherent damping with additive manufacturing. *Additive Manufacturing*, *38*, 101739. https://doi.org/10.1016/j.addma.2020.101739.

Srinivasan, A. V. (1997). Flutter and resonant vibration characteristics of engine blades: An IGTI scholar paper. *Proceedings of the ASME 1997 International Gas Turbine and Aeroengine Congress and Exhibition, Volume 4: Manufacturing Materials and Metallurgy; Ceramics; Structures and Dynamics; Controls, Diagn*, Article 97-GT-533.

https://doi.org/10.1115/97-GT-533.

Szwedowicz, J. (2012). *Bladed disks: Non-linear dynamics, structural design of aircraft engines*. Article RTO-AVT-207-09.

Szwedowicz, J., Secall-Wimmel, T., & Dünck-Kerst, P. (2008). Damping performance of axial turbine stages with loosely assembled friction bolts: The nonlinear dynamic assessment. *Transactions of the ASME, Journal of Engineering for Gas Turbines and Power*, *130*, Article 032505.

Wdowiński, W., Szymczyk, E., Jachimowicz, J., & Moneta, G. (2017). Design and strength analysis of curved-root concept for compressor rotor blade in gas turbine. *Fatigue of Aircraft Structures*, *2017*(9), 137–155. https://doi.org/10.1515/fas-2017-0011.