ISSN: 1231-4005
e-ISSN: 2354-0133
DOI: 10.5604/12314005.1217255

DRY FRICTION DAMPER FOR SUPERCRITICAL DRIVE SHAFT

Witold Perkowski

Institute of Aviation
Krakowska Avenue 110/114, 02-256 Warsaw, Poland
tel.: +48 22 8460011 ext. 373, fax: +48 22 8465774
e-mail: witold.perkowski@ilot.edu.pl

Abstract

In this article the construction and mathematical model of a dry friction damper with radial gap, designed for dumping flexural vibrations of a supercritical propulsion shaft (developed previously during design works made in Institute of Aviation on propulsion system for an ultralight IS-2 helicopter), while passing through the resonance is presented. Some results of mathematical and numerical analyses of such a system (supercritical shaft + damper) behaviour are also presented – among other things, very distinctive behaviour of the shaft, while passing through resonance, is shown. From a theoretical point of view, it is interesting that certain range of damper parameters, obtained in the course of numerical analysis, lead to chaotic vibrations of the system (they were also observed in practice). From a practical point of view, it will be interesting that the article shows a way how to create a dimensionless (and therefore general) parameters of the system: supercritical shaft and friction damper and also simple engineering methodology for selection of suitable (for the correct operation of the shaft) damper parameters depending on the parameters of the shaft, developed during series of analyses. The practical aspect of the problem seems to be particularly important, as the supercritical propulsion shaft equipped with a friction damper can be very structurally simple, light and inexpensive, and still it is not widely used, probably because of certain doubts aroused in the constructors by the term: "resonant vibrations".

Keywords: *supercritical drive shaft, supercritical propulsion shaft, dry friction damper*

1. Introduction

Supercritical drive shaft, when reaching its operating rotational speed, at least once passes through the critical speed. During this pass, it significantly increases the amplitude of its vibrations, which, without sufficient vibration damping, can lead to damage or destruction of the shaft. This is one of the reasons why supercritical shafts are not widely adopted in the machinery, although it is known how to prevent excessive vibrations in a dangerous area: ensure adequate dampening or to cross the dangerous speed range fast enough (the best is using both techniques simultaneously). Once danger area is exceeded and appropriately distanced, characteristic self-stabilization of the shaft appears and it rotates from now very stable, and the amplitude of the vibrations is small [5]. If we do not mind some "overhang" of the shaft under its own weight, well-designed supercritical shaft is a great option when we want to provide the drive for a relatively large distance to the power receiver operating mainly at constant speed – with the use of supercritical shaft it can be achieved at low weight, simple design and technology, and at low cost of the shaft. A classic example of such application may be providing power from the main gearbox of the helicopter, to its tail gearbox. The replacement of the classical solution (the shaft divided into several sub-critical sections connected one to another by means of flexible couplings, and with many supports on the helicopter tail beam) with "one-piece" supercritical shaft, can save not only a lot of mass and cost, but also improves security by eliminating the dozens of so-called "crucial elements", failure of which could lead to crash. The article describes a simple-construction dry friction damper for damping supercritical shaft flexural vibrations, describes its model as well as the proposition of a methodology for pre-selection of suitable dampers parameters depending on parameters of the shaft. This methodology provides simple, engineering guideline, developed on the basis of many simulations of the shaft-damper system dynamics, which, as far as the author's knowledge reaches, are lacking in the literature. All values used in the article are expressed with SI units.

2. The primary shaft parameters

The primary shaft was designed as tail transmission shaft for a prototype of IS-2 helicopter.

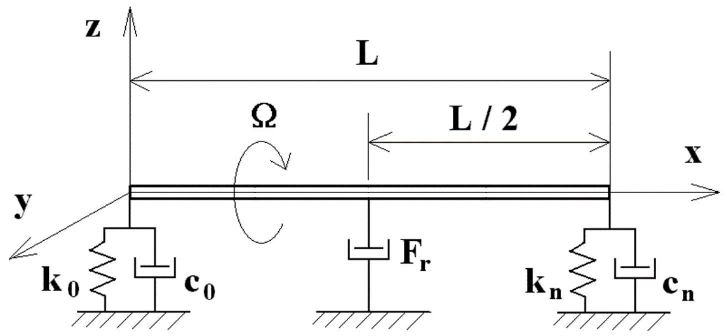

Fig. 1. Physical model of the primary shaft equipped with a dry friction damper

Primary shaft parameters:
- length, L = 3.32 [m],
- density, ρ = 2700 [kg/m^3],
- cross-section area, A = 0.00022 [m^2],
- Young modulus, $E = 0.7\cdot10^{11}$ [Pa],
- moment of inertia (geometrical), $I = 59.3\cdot10^{-9}$ [m^4],
- support stiffness, $k_0 = k_n = 10^6$ [N/m],
- support damping, $c_0 = c_n = 0$ [Ns/m],
- external (viscous) damping coefficient, c_t = 0.5 [Ns/m^2],
- eccentricity (constant along the shaft), e = 0.001 [m].

For such a shaft (without damper) the 1-st critical speed Ω_0 is, due to: low moment of inertia, low damping and high supports stiffness (relative to the stiffness of the shaft), practically equal to the 1-st frequency of free, undamped vibrations of analogous beam on non-deformable supports – ω_0.

3. The concept of the damper

The concept of the damper [7, 8] was developed at the Institute of Aviation, for cooperation with the primary shaft, described before. Structural sketch of the damper is shown in Fig. 2.

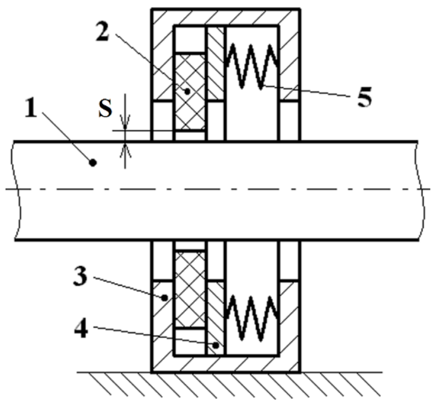

Fig. 2. The dry friction damper structural sketch: 1 – shaft, 2 – movable disc, 3 – damper body, 4 – pressure plate, 5 – pressure spring

The key assumptions for the damper were following:

1) the damper uses only dry friction and its characteristic is a gap *S* (radial clearance) in the movable disc (Fig. 2) selected so that the outside of the resonance region shaft rotates freely, without any contact with the damper,
2) the damper cooperates with a supercritical shaft of the 1-st order (its operating speed is located between the 1-st and the 2-nd critical speed), which has the form of a cylindrical beam, pivotally supported at the ends, and the damper is located in the middle of the shaft (Fig. 1).

4. The mathematical model of the damper

It was assumed that the damper disc performs the plane motion, and its mass is negligible. The disc movement is described in the fixed, left-handed coordinate system *0xyz* (Fig. 1, Fig. 3).

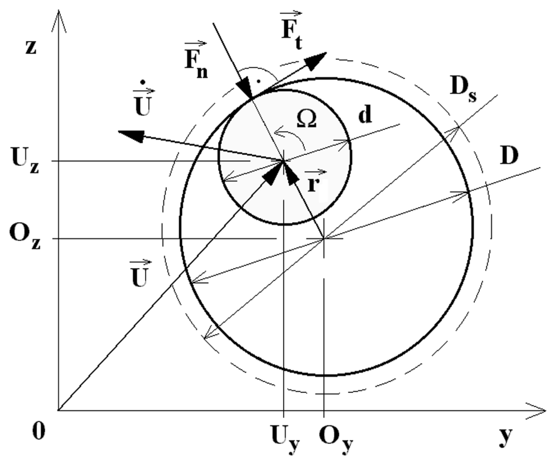

Fig. 3. The shaft-damper interaction model and its designation system

An essential element of the damper is movable disc, which has a central hole of *D* diameter (through which the shaft of *d* diameter passes). By varying the pressure springs (Fig. 2) force, specified friction against the side faces of the disc is set, giving the damper resistance force F_r. On the shaft, which is in contact with the damper disc, operates resultant force *F*, which has two components: a normal F_n and tangential F_t. The impact forces between the shaft and the damper and basic designations used to describe the model of the shaft-damper interaction, are shown in Fig. 3. Vectors *U* and *O* are position vectors respectively for the shaft centre and the centre of the damper disc in fixed coordinate system *0xyz*, while the vector *r* is the position vector for the shaft centre in a moving coordinate system associated with the centre of the damper disc (this system performs, along with the disc, plane motion in the system *0xyz*). In order to facilitate the determination of the impact force *F* between the shaft and the disc, an elastic zone of D_s diameter was introduced into the disc. In the zone, the force *F* varies linearly from zero to F_r (Fig. 4).

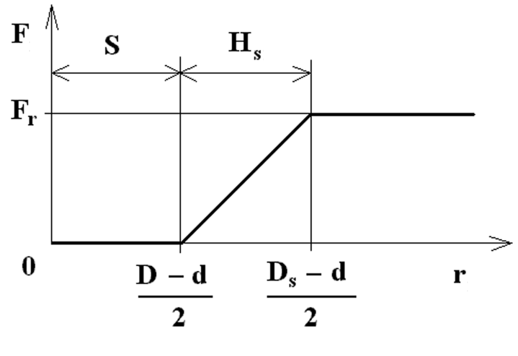

Fig. 4. The force of shaft-damper interaction F(r)

The introduction of this zone allows avoiding answering the question: "what is actually the force of shaft-damper interaction when they are in contact?" – its presence makes this force "self-developing". This is very important because knowledge of the damper resistant force F_r is not sufficient to determine the impact of the shaft-damper force F, which can be shown through a thought experiment: the fact that the damper resistance is infinite does not mean that at the shaft, being in contact with the damper, acts infinite force (it's absurd). The elastic zone is not only very convenient "trick" for easy calculation, but it also has a physical sense – it can be linked with the disc own elasticity and elasticity of the whole damper mounting. The thickness of the elastic zone is very important in numerical modelling of the shaft vibrations – it cannot be too small and should be selected in pair with integration step for equations of motion – the shaft should need at least a few steps to traverse this zone, otherwise it may lead to its "overshooting", and, consequently, to the wrong "work" of the model. The numerical model of the damper assumes that the shaft, after exceeding the elastic zone, moves the damper disc by the amount of that excess.

Dry friction damper parameters:

– resistant force, F_r [N],
– the gap, $S = (D - d)/2$ [m],
– thickness of the elastic zone, $H_s = (D_s - D)/2$ [m],
– shaft-damper friction coefficient, μ [1],

where:

d – diameter of the shaft [m],
D – diameter of the disc hole [m],
D_s – diameter of the elastic zone [m].

The following concatenations were used to determine the vectors F, F_n, F_t:

$$\vec{F} = \vec{F}_n + \vec{F}_t, \tag{1}$$

$$F_t = \mu \cdot F_n, \tag{2}$$

$$F_n(r) = \frac{F(r)}{\sqrt{1+\mu^2}}, \tag{3}$$

$$\vec{F}_n = -F_n(r) \cdot \frac{\vec{r}}{|\vec{r}|}, \tag{4}$$

$$\vec{F}_t \cdot \dot{\vec{U}} < 0. \tag{5}$$

5. Dimensionless parameters of the shaft-damper system

In order to develop dimensionless damper parameters a "substitute shaft" introduced. The primary shaft, with constant parameters (but not the only one), can be replaced with a substitute shaft in which mass, eccentricity and damping are concentrated in the middle of the shaft (Fig. 5).

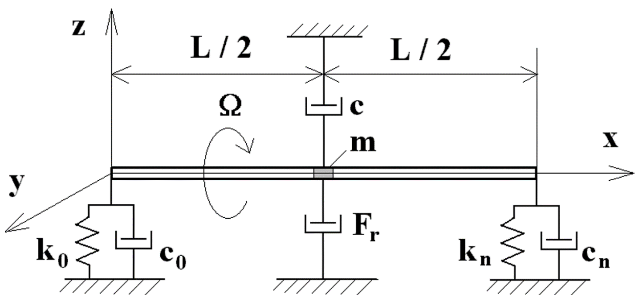

Fig. 5. Physical model of the substitute shaft equipped with a dry friction damper

In a limited range of rotational speed (up to about $3\Omega_0$, regardless of the eccentricity distribution in the primary shaft) the substitute shaft can be dynamically very similar to the primary shaft. It was assumed that the 1-st critical speed of Ω_0 (and the 1-st frequency of own vibrations ω_0) are the same for the primary and the substitute shaft, the same are also all of the beam parameters (length, Young's modulus etc.). Similar dynamic deflections of the two shafts in the relevant speed range are provided by appropriately chosen: substitute eccentricity and substitute external damping.

The stiffness of the substitute shaft in its midpoint can be obtained with the use of the known formula for homogeneous beam (supported in non-deformable supports):

$$k = \frac{48EI}{L^3}. \tag{6}$$

For known Ω_0, substitute (concentrated) mass can be calculated as follows:

$$\Omega_0 = \sqrt{\frac{k}{m}} \rightarrow m = \frac{k}{\Omega_0^2}. \tag{7}$$

Substitute eccentricity ε depends on the eccentricity distribution along the primary shaft, but when it is constant ($e(x) = e = \text{const}$) we obtain:

$$\varepsilon = 1.45 \cdot e. \tag{8}$$

Substitute external damping c depends on the damping distribution along the primary shaft, but when it is constant ($c_t(x) = c_t = \text{const}$) we can obtain it from simple relation:

$$c = c_t \cdot L \frac{m}{\rho \cdot A \cdot L} = c_t \cdot \frac{m}{\rho \cdot A}. \tag{9}$$

Now we can define the dimensionless parameters of the shaft-damper system (shaft speed, shaft deflection, damper resistant force, damper gap and elastic zone thickness):

$$\beta = \frac{\Omega}{\Omega_0}, u = \frac{U}{\varepsilon}, \gamma = \frac{c}{m\Omega_0}, f_r = \frac{F_r}{m\Omega_0^2\varepsilon}, s = \frac{S}{\varepsilon}, h_s = \frac{H_s}{\varepsilon}. \tag{10}$$

6. Results of dynamics simulations of the shaft-damper system

In the simulations a FEM model of the primary shaft was used – the model was described previously [2, 3, 5, 7], so only brief description here: the shaft has been modelled by means of 4 beam elements of equal length, which deformations are described by the respective Hermite polynomials. The force coming from the damper was introduced to the shaft model as extra edge forces for the corresponding finite elements. The results of the calculations are displacements of the shaft nodes (finite element ends) in time – especially, the most interesting displacements of the central node. The results allowed dividing main dimensionless parameters of the damper (f_r and s) into: "large" – greater than 1, and "small" – less than 1. A complete picture of characteristic behavior of supercritical shaft equipped with a "gapped" dry friction damper is shown in Fig. 6.

With the increasing rotational speed of the shaft: gradually the shaft deflection in the midpoint reaches a size of s (at β_1 shaft comes into the contact with the damper) and becomes fixed at a certain time, then at speed β_2 shaft has "enough power" to move the damper disc, at $\beta_3 > 1$ amplitude of the shaft reaches a maximum (resonance), then shaft deflection decreases, damper disc is "centred" and at β_4 shaft comes out from working with the damper – it is followed by a "jump down", further the dimensionless amplitude of vibrations is heading asymptotically to 1.

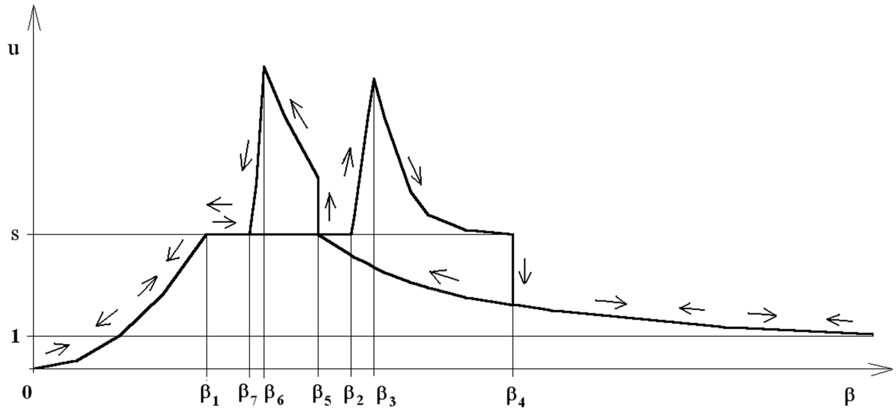

Fig. 6. The illustrative process of dimensionless displacements in the midpoint of the shaft equipped with a dry friction damper of a "big" resistance (fr > 1) and a "big" gap (s > 1) – the arrows indicate changes in shaft speed β

When we reduce the speed of rotation: the shaft comes into contact with the damper disc at β_5 and it is immediately followed by a "jump up", then the amplitude of the shaft vibration reaches a maximum at β_6 <1 (resonance), later followed by "centring" of the disc at β_7, then the shaft vibrates with the amplitude of *s*, and finally, at β_1 the shaft "gently" loses contact with the damper.

The characteristic "jumps" (up and down) shown here are not connected with the dry friction damper presence – they are characteristic for the supercritical shaft with an intermediate support (can be elastic or otherwise), which has a gap [1, 4].

How effective can the use of well-chosen friction damper be is illustrated by the results obtained in simulations of dynamic deflection in the midpoint of the primary shaft without a damper (Fig. 7) and with the damper (Fig. 8). As it can be seen, the dry friction damper almost completely "turned off" resonant vibrations of the shaft. In both cases, the angular acceleration of the shaft is constant and the same: $d\Omega/dt = 10$ rad/s^2. In the simulations, in general, the effect of the shaft weight was omitted, but it was verified that to take this effect into account, displacements should be referred to the line of free deflection of the non-rotating shaft, instead of to the *0x* axis (according to theory).

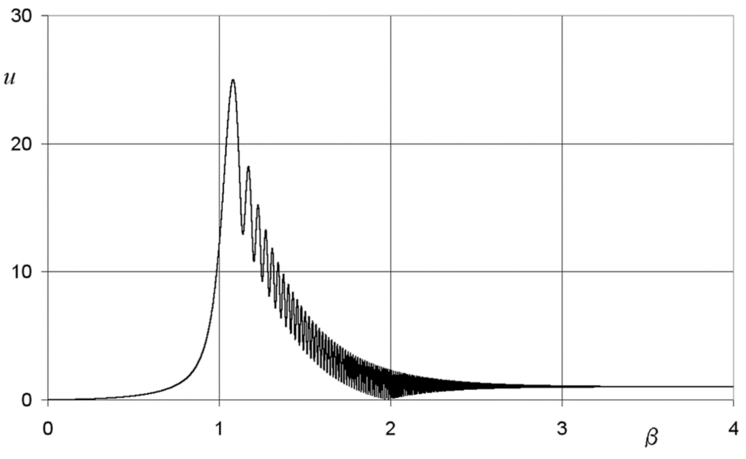

Fig. 7. The course of u(β) in the point x/L = 0.5 for the primary shaft without a damper

The mathematical analysis of the problems as well as numerical simulations [7] showed that the damper is, in practice, sufficiently characterized by only 2 main parameters: the resistance and the gap (elastic zone only facilitates calculations and a moderate friction between the shaft and the damper disc e.g. μ < 0.2 , has hardly no effect on the shaft behaviour). They also allowed to formulate 3 practical conditions for the damper properly chosen for the supercritical shaft of nominal, dimensionless rotational speed β_{nom}:

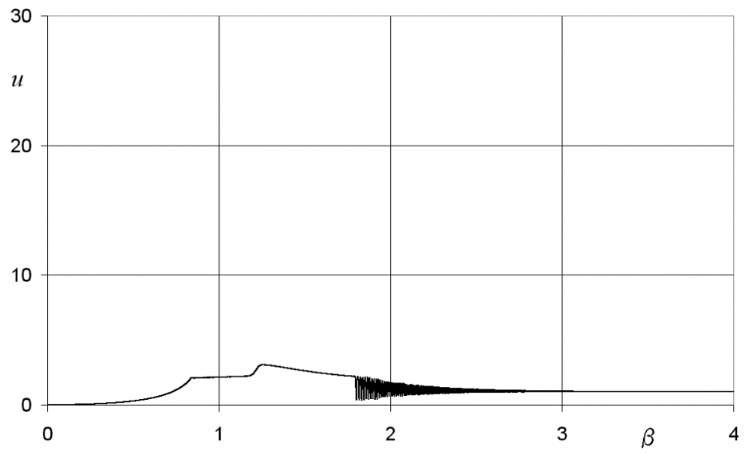

Fig. 8. The course of u(β) in the point x/L = 0.5 for the primary shaft with the dry friction damper of following parameters: s = 2.07, f_r = 2.024, h_s = 0.138 (F_r = 16 N, S = 0.003 m, H_s = 0.0002 m), μ = 0

$$s > 1 \text{ (necessary for the shaft freely rotating after leaving the damper),} \quad (11)$$

$$f_r > 1 \text{ (for effectively reducing the amplitude of the shaft vibrations in resonance),} \quad (12)$$

$$\beta_4 \approx \sqrt{\frac{s+f_r}{s-1}} < \min(\beta_{nom}, 2) \text{ (allows the shaft to leave the damper before } \beta_{nom}\text{),} \quad (13)$$

and if the nominal speed of rotation of the shaft $\beta_{nom} > 2$, this condition amounts to:

$$\beta_4 < 2 \Leftrightarrow f_r < 3s - 4. \quad (14)$$

These conditions (for the case: $\beta_{nom} > 2$) are shown graphically in Fig. 9 where s_{max} is the maximum allowable gap due to the damper construction and allowable shaft deflection, and the shaded area indicates where to look for the appropriate damper parameters. The graph highlights also the point (2, 2) which, according to the author, is close to optimum.

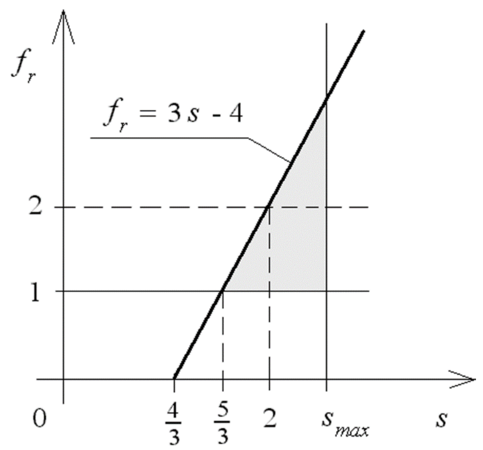

Fig. 9. Recommended parameters of the dry friction damper when β_{nom} > 2

7. Practical guideline for the selection of the dry friction damper parameters

For a simple shaft similar to the primary shaft described above (long cylindrical beam, pivotally supported at the ends in relatively rigid supports) which operating speed is above $2\Omega_0$ (it is recommended to set operating speed in the range of $2\Omega_0$ - $3\Omega_0$), the selection of the proper damper parameters can be made in a few simple steps:

1) determine the shaft stiffness k from (6),

2) determine the substitute mass of the shaft *m* from (7),
3) estimate the shaft eccentricity *e* (it is caused mainly by straightness deviation, so take maximum value from the manufacturer specification, or, ideally, rotate your supported shaft slowly and measure displacements in the middle) – it's the hardest step,
4) determine the substitute eccentricity ε from (8),
5) take f_r and *s* from Fig. 9 – for example f_r = 1.9 and *s* = 2.1 (just in case take f_r a little bit smaller and *s* a little bit great then highlighted value of 2),
6) having: m, ε, f_r, s calculate dimensional parameters of the damper, according to (10):

$$F_r = f_r \cdot m \cdot \omega_0^2 \cdot \varepsilon, \tag{15}$$

$$S = s \cdot \varepsilon. \tag{16}$$

Conclusions

The conclusions are summarized in a few points below:

1) The dry friction damper with a gap, properly sized, very effectively reduces supercritical shaft resonance vibrations and allows it to rotate freely, without contact with the damper, at nominal rotating speed.
2) The damper is very simple and inexpensive and can be easily build, even by amateurs.
3) The damper has 2 main parameters: the resistant force F_r and the gap *S*.
4) For the supercritical shaft of simple design, similar to the primary shaft described before (which can also be easily build by amateurs), proper parameters of the damper can be easily found with the guidance contained in the article (at least initially – then they can be adjusted in the trials).
5) As the contact between the shaft and the damper is limited in time (resonance region), friction wear should not be large for well-chosen materials.
6) The damper disk wear, in the form of the increase of the gap, does not adversely affect the operation of the damper and causes only comparable increase in amplitude of the shaft vibrations.
7) We should be aware when mounting the damper that the "flexible" shaft hangs under its own weight – in the "neutral" position the damper disc should be more or less concentric to the shaft.
8) The supercritical shaft can be a great solution when we need do provide the mechanical power for a long distance and we don't mind about shafts overhang and relatively high amplitude of shaft vibrations (in order of millimetres). Such a shaft can be successfully used without balancing it at all or with static balancing only (dynamic balancing of a long, flexible shaft could be very hard) – if the shaft is relatively straight (e.g. a well-manufactured tube or rode) it should be good enough.

References

[1] Bernay, B., *Aircraft engine responce due to Fan unbalance to the presence of "consumed" gaps in the engine during the phase of windmilling*, ICAS 2000 papers.

[2] Dżygadło, Z., Perkowski, W., *Nonlinear dynamic model for flexural vibrations analysis of a supercritical helicopter's drive shaft*, ICAS 2000 papers.

[3] Dżygadło, Z., and others, *Zespoły wirnikowe silników turbinowych*, WKiŁ 1982.

[4] Ehrich, F. F., O'Connor, J. J., *Stator whirl with rotors in bearing clearence*, Journal of Engineering for Industry, 1967.

[5] Gryboś, R., *Dynamika maszyn wirnikowych*, PWN, 1994.

[6] Kruszewski, J. and others, *Metoda sztywnych elementów skończonych*, Arkady, 1975.

[7] Perkowski, W., *Analiza dynamiki wału pracującego w warunkach nadkrytycznych, do napędu śmigła ogonowego ultralekkiego śmigłowca*, praca doktorska, thesis, WAT, 2001.

[8] Patent: PL 196 167 B1.