GAS FOIL BEARINGS: LIMITS FOR HIGH-SPEED OPERATION

Luis San Andrés, Tae Ho Kim
Mechanical Engineering Department, Texas A&M University, College Station, Texas 77843-3123, USA

ABSTRACT
Commercial oil-free micro turbomachinery relies on gas foil bearings (GFBs) for reliable performance with improved efficiency. However, GFB modeling is still largely empirical, lacking experimental validation. An analysis of simple GFBs operating at large shaft speeds (infinite speed number) follows. The bearing ultimate load and stiffness coefficients are derived from simple algebraic equations for the gas film pressures at the equilibrium journal position and due to small amplitude journal motions, respectively. GFBs without a clearance or with assembly interference are easily modeled. The underlying elastic structure (bump foil strip) determines the ultimate load capacity of a GFB as well as its stiffnesses, along with the limiting journal displacement and structural deformation. Thus, an accurate estimation of the actual minimum film thickness is found prior to performing calculations with a complex computational model, even for the case of large loads that result in a journal eccentricity well exceeding the nominal clearance, if applicable. An initial assembly preload (interference between shaft and foil) increases the GFB static clearance or with assembly interference are easily modeled. The relationship between hydrodynamic pressure (P) and film thickness (h) follows from the limiting form of Reynolds equation at infinite speed number, i.e. very high journal speeds, Ω→∞ [11]:

\[ \frac{\partial (P_h)}{\partial \theta} = 0 \text{ or } (P_h)_{\theta=0} = P_a h_{\theta=0} \]  

where θ is the circumferential coordinate, P_a is ambient pressure, and

\[ h = \left(c - r_p\right) + e \cos(\theta) + e \sin(\theta) + \frac{1}{K_f} \left(P_c - P_a\right) \]  

c and r_p are the radial clearance and assembly preload, and e_x, y are the journal eccentricity components. \( K_f \) is the stiffness per unit area of the foil support structure. It is simple to show that a static load (W) applied through a pressure renders a pressure field and film thickness symmetric about this angle, and thus e = e_x and e_y = 0. Thus, a GFB has no cross-coupling since the journal eccentricity (e) is parallel to the load direction. Small amplitude journal motions about the equilibrium position, e_x = e_x + Δe_x and e_y = e_y + Δe_y, render changes in the film pressure, i.e. \( P = P_a + P_x + d_x + P_y + P_y \). From eqn. (1) follows that

\[ P_{h(\theta)} = \left[ P_a + c - r_p + e \cos(\theta) \right] \left( \frac{P_a - P_c}{K_f} \right) \]  

\[ P_{X(\theta)} = \left[ \frac{P_a - P_c}{K_f} \right] \left( h_{\theta=0} + \frac{1}{K_f} \right) ; \quad \frac{P_{Y(\theta)}}{P_a} = -\left( \frac{P_{h(\theta)}}{h_{\theta=0}} + \frac{P_{Y(\theta)}}{P_a} \right) \]  

The solution of quadratic eqn. (3) is

\[ P_{h(\theta)} = \frac{K_f}{2} \left[ A_1 + \left( A_2 + 4 A_3 \right) \right] \]  

where \( A_1 = e \cos(\theta) - P_c / K_f \), \( A_2 = (c - r_p + e) P_c / K_f \) as in [11], except for the included preload \( r_p \). Integration of the pressures, equilibrium and perturbed, on the bearing surface (LD) renders the reaction force and stiffness coefficients \( K_{X(\theta)}, K_{Y(\theta)} \), i.e.

\[ \{ W \} = \left[ K_{XX} \quad K_{XY} \right] \left[ K_{XY} \quad K_{YY} \right] \left[ \frac{\cos(\theta)}{\sin(\theta)} \right] d\theta \]  

As expected, \( K_{XY} = K_{X(\theta)} = 0 \). Incidentally, damping force coefficients are nil at infinite speed operation [14].

RESULTS AND DISCUSSION
An example of ultimate GFB force performance follows. The bearing length (L) and diameter (D) are 38 mm, c = 0.032 mm, with foil support stiffness \( K_f = 4.74 \text{ Gm}^2 \) [8], and \( P_r = 1.01 \text{ bar} \). Dimensionless load (W') and direct stiffnesses \( K_{XX}', K_{YY}' \) relate to \( P_a LD \) and \( P_a LD R \), respectively. Engineered GFBs must have \( W' > 1 \), i.e. specific pressure \( (W' L D) > P_a \). The clearance (c) is referential only. Figure 1 shows the journal eccentricity (e = c/2) versus load \( W' \) at increasing speed numbers, \( \Lambda = 6(\pi D^2 P_a)(R / R^2 \epsilon) \), as obtained from [7]. The journal displacements lie between the limit at \( \lambda = \infty \), and the structural deformation at null shaft speed, \( \Lambda = 0 \) [13]. Thus, the
simple formulae, eqn. (3-6), in conjunction with [13], facilitate the estimation of the actual GFB eccentricity at a finite shaft speed.

Figure 2 depicts the effect of an increasing preload \( r_p \) on the limit journal eccentricity and minimum film thickness at both null and infinite shaft speeds. \( r_p = c \) denotes an effective null clearance, and \( r_p = 2c \) a positive interference. \( c = (c-r_p) \) ensures a gas film with generation of hydrodynamic pressure. As \( W \) increases, the journal displacements approach the deflections of the support structure. The film thickness tends to similar magnitudes, irrespective of the preload. A GFB cannot exceed the elastic load limit from its support structure.

Figure 3 shows the stiffness, \( K'_{xx} \), at null and infinite speeds for three preloads (similar \( K'_{yy} \) not shown for brevity). As \( W \) increases, the GFB stiffness approaches its corresponding structural value, which is largest for the configuration with an assembly interference, \( r_p = 2c \), and smallest for one with a finite clearance, \( r_p = 0 \). Note that the structural stiffness is piece-wise linear depending on the contact area for a given load and assembly preload [13].

CONCLUSIONS

The paper presents a simple analysis to estimate the limiting journal eccentricity, minimum film thickness and stiffnesses of a GFB operating at infinite shaft speed. The predictions demonstrate the ultimate load of a GFB can not exceed that of its underlying elastic structure. The GFB combines in series the gas film and structural stiffnesses. The structural stiffness is much softer than the gas film stiffness, and thus it is the commanding one in the actual operation of GFBs supporting significant loads.

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REFERENCES


