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Non Linear Dynamics of a Flexible Rotor Supported by Laminar Fluid Film Journal Bearing with Couple Stress Fluid

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ABSTRACT

The study presents a dynamic analysis of a rotor supported by two laminar flow model journal bearings and lubricated with couple stress fluid under nonlinear suspension. The dynamics of the rotor center and bearing center is studied. The dynamic equations are solved using the ODE-45 MATLAB method. The analysis methods employed in this study is inclusive of the dynamic trajectories of the rotor center and bearing center. The results show that the values of dimensionless parameters l^* strongly influence dynamic motions of bearing and rotor center. It is found that couple stress fluid improve the stability of the system when $l^* < 0.25$ even if the flow of this system is laminar. We also demonstrated that the dimensionless rotational speed ratios s and the dimensionless unbalance parameter b are also significant system parameters.

INTRODUCTION

Couple Stress Fluid is a mixture of Newtonian fluid & some additive (long chain organic compound) which is responsible for stability. Sometimes lubricating fluids of low viscosity are used as lubricants to simplify the equipment design or simplify the numerical simulation models. Hydrodynamic journal bearings using in rotor high-speed turbo machinery lubricated with liquid metals. Cheng-Ying (2008) nonlinear dynamics of a flexible rotor supported by turbulent journal bearings with couple stress fluid. This study presents a dynamic analysis of a rotor supported by two turbulent flow model journal bearings and lubricated with couple stress fluid under nonlinear suspension. The dynamics of the rotor center and bearing center is studied. Naschie and Chaos (1988) first proposed the concept of chaotic turbulence. He introduced the generalized bifurcation and temporal chaos in science and engineering and then proceeded to show that the localized buckling of elastic shells can be viewed globally as a form of special chaos. He also concluded that localized shell buckling can be interpreted as special turbulence of a thin elastic surface just like the turbulence can be interpreted as special temporal deterministic chaos of a fluid. Gardner and Ulschmid (1974). performed an experiment for a tilting-pad and a sleeve journal bearing. Hopf G. and Schuler (1989) an experiment for journal bearings lubricating with transition flows between laminar and turbulent flow regimes. Mittwollen and Glienicke (1990) proposed a global analysis method for a Reynolds equation based on the empirical turbulence coefficients defined by Constantinescu and they extended this global concept to the energy equation.

MATERIALS AND METHODS

The present research considers a flexible rotor supported by two couple stress fluid film journal bearings with foundation which behaves as nonlinear springs subjected to a periodic external excitation is studied using a ODE45 routine of MATLAB.

Fig.1 shows a flexible rotor supported horizontally by two identical couple stress fluid film journal bearings with nonlinear springs. $O_m$is the center of rotor gravity, O_1 is the geometric center of the bearing, O_2 is the geometric center of the rotor, O_3 and is the geometric center of the journal. Fig.1 shows the cross section of the fluid film journal bearing where (X, Y) is the fixed coordinate and (e, u) is the rotated coordinate, e being the offset of the journal center and φ being the attitude angle of the X-coordinate.

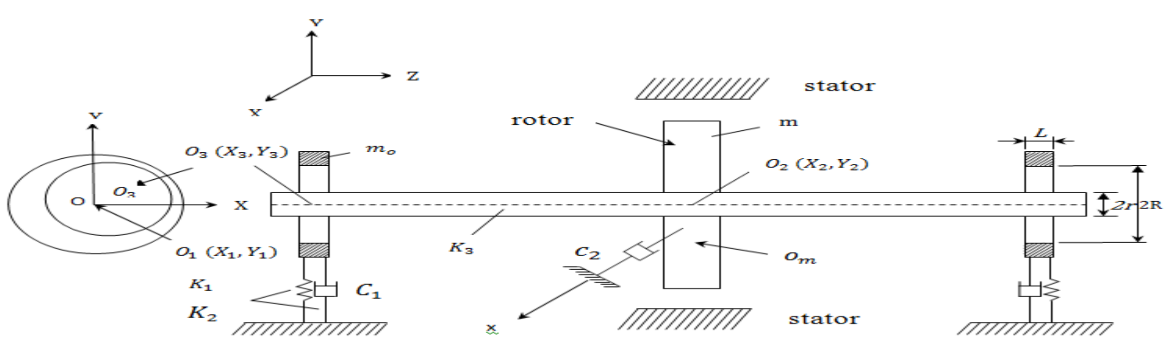


Fig. 1. Model of a flexible rotor supported on two non-linear suspensions

$$F_x = f_e \cos\emptyset + f_\emptyset \sin\emptyset = \frac{K_s(X_2 - X_3)}{2} \quad (1)$$

$$F_y = f_e \sin\emptyset + f_\emptyset \cos\emptyset = \frac{K_s(Y_2 - Y_3)}{2} \quad (2)$$

$$m\ddot{X}_2 + c_2\dot{X}_2 + k_s(X_2 - X_3) = m\rho\omega^2 \cos\emptyset \tag{3}$$

$$m\ddot{Y}_2 + c_2\dot{Y}_2 + k_s(Y_2 - Y_3) = m\rho\omega^2 \sin\emptyset - mg \tag{4}$$

$$m_0\ddot{X}_1 + c_1\dot{X}_1 + k_1X_1 + k_2X_1^3 = \tag{5}$$

$$m_0\ddot{Y}_1 + c_1\dot{Y}_1 + k_1Y_1 + k_2Y_1^3 = -m_0g + F_y \tag{6}$$

$$\frac{1}{R^2}\frac{\partial}{\partial\theta}\left(\xi(h,l)G_\theta\frac{\partial p}{\partial\theta}\right) + \frac{\partial}{\partial z}\left(\xi(h,l)G_z\frac{\partial p}{\partial z}\right) = \frac{\mu U}{2R}\frac{\partial h}{\partial\theta} + \mu\frac{\partial h}{\partial t} \tag{7}$$

Thus Reynolds equation can be rewritten as.

$$\frac{1}{R^2}\frac{\partial}{\partial\theta}\left(\xi(h,l)G_\theta\frac{\partial p}{\partial\theta}\right) + \frac{\partial}{\partial z}\left(\xi(h,l)G_z\frac{\partial p}{\partial z}\right) = -6\mu\omega c\varepsilon\sin\theta + 12\mu(c\varepsilon\dot{\emptyset}\sin\theta) \tag{8}$$

$$\frac{\partial^2 p}{\partial z^2} = \frac{-6\mu\omega c\varepsilon\sin\theta + 12\mu(c\dot{\varepsilon}\cos\theta + c\varepsilon\dot{\emptyset}\sin\theta)}{\xi(h,l)G_z} \tag{9}$$

$$B.C.\begin{cases}\dfrac{\partial p}{\partial z} = 0, & z = o\\ p = 0, & z = \pm\dfrac{L}{2}\end{cases}$$

$$P = -\frac{3\mu c}{\xi(h,l)G_z}\left[(\omega - 2\dot{\emptyset})\varepsilon\sin\theta - 2\dot{\varepsilon}\cos\theta\right]\left(Z^2 - \frac{L^2}{4}\right) \tag{10}$$

$$f_r = \int_0^{\pi}\int_{-\frac{L}{2}}^{\frac{L}{2}} pR\cos\theta\, dz\, d\theta \tag{11}$$

$$f_t = \int_0^{\pi}\int_{-\frac{L}{2}}^{\frac{L}{2}} pR\sin\theta\, dz\, d\theta \tag{12}$$

$G_z \& G_\theta = 1$, *because it is laminar.*

$$f_e = -\frac{\mu L^3 R}{2c^2}\int_0^{\pi}\left\{\frac{\left[(\omega - 2\dot{\emptyset})\varepsilon\sin\theta - 2\dot{\varepsilon}\cos\theta\right]\cos\theta}{\left[(1+\varepsilon\cos\theta)^3 - 12(l^*)^2(1+\varepsilon\cos\theta) + 24(l^*)^3\tanh\left(\frac{1+\varepsilon\cos\theta}{2l^*}\right)\right]}\right\}d\theta \tag{13}$$

$$f_\emptyset = -\frac{\mu L^3 R}{2c^2}\int_0^{\pi}\left\{\frac{\left[(\omega - 2\dot{\emptyset})\varepsilon\sin\theta - 2\dot{\varepsilon}\cos\theta\right]\sin\theta}{\left[(1+\varepsilon\cos\theta)^3 - 12(l^*)^2(1+\varepsilon\cos\theta) + 24(l^*)^3\tanh\left(\frac{1+\varepsilon\cos\theta}{2l^*}\right)\right]}\right\}d\theta$$

(14)

$$x_1'' + \frac{2\xi_1}{s_1}x_1' + \frac{1}{s_1^2}x_1 + \frac{\alpha}{s^2}x_1^3 - \frac{1}{2c_{0_m}s^2}(x_2 - x_1 - \varepsilon\cos\phi) \tag{15}$$

$$y_1'' + \frac{2\xi_1}{s_1} y_1' + \frac{1}{s_1^2} y_1 + \frac{\alpha}{s^2} y_1^3 - \frac{1}{2c_{0_m} s^2}(y_2 - y_1 - \varepsilon \sin\phi) + \frac{f}{s^2} = 0 \quad (16)$$

$$x_2'' + \frac{2\xi_2}{s} x_2' + \frac{1}{s^2}(x_2 - x_1 - \varepsilon \cos\phi) = \beta \cos\phi \quad (17)$$

$$y_2'' + \frac{2\xi_2}{s} y_2' + \frac{1}{s^2}(y_2 - y_1 - \varepsilon \sin\phi) = \beta \sin\phi - \frac{f}{s^2} \quad (18)$$

Where,

c_1 Damping coefficient of supported Structure

c_2 Viscous damping of the rotor

F_x , F_y Component of fluid film force in X and Y direction

k_1, k_2 Stiffness of the spring which support the bearing housings

f_e, f_φ Components of the fluid film force in radial and tangential direction

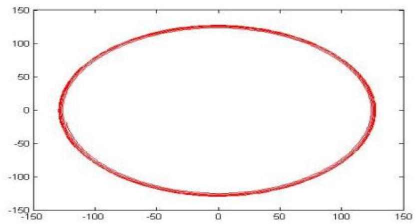
φ Attitude angle

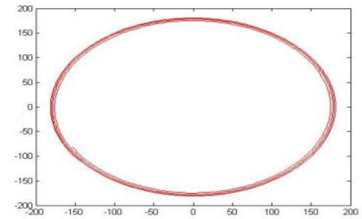
4. RESULTS AND DISCUSSION

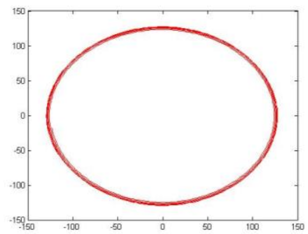
This work has explored the effects of varying the main system parameters of a flexible rotor supported by laminar journal bearings and lubricated with couple stress fluid under nonlinear suspension. The results show that the values of dimensionless parameters l* strongly influence dynamic motions of bearing and rotor center. It is found that couple stress fluid improve the stability of the system when l* < 0.25 even if the flow of this system is turbulent. We also demonstrated that the dimensionless unbalance parameter b is also a significant system parameter. It is well known that rotor-bearing system operating in a state of periodic motion exhibit broad band vibrations with comparatively large vibrational amplitudes, enhancing the probability of fatigue failure. Simulation results show that bearing center displacement may give rise to undesirable nonsynchronous vibrations.

Table 1 Specification of the rotor-bearing system

Journal diameter(D)	0.0254m
Length of bearing(L)	0.0127m
Mass of rotor (2m)	5.4523 kg
Radial clearance(C)	50.8µm
Lubricant	ISO 32

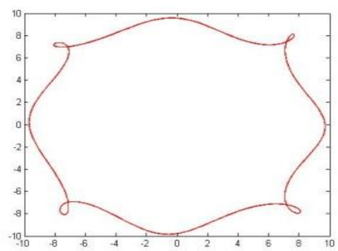


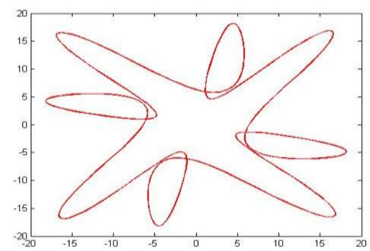


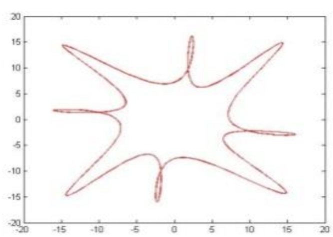


L=0.05, S=1.000 L=0.05, S=5.500 L=0.05, S=7.000

Fig.1 Fig.2 Fig.3







L=0.10, S=2.500 L=0.10, S=4.500 L=0.10, S=5.500

Fig.4 Fig.5 Fig.6

Figure 1,2&3 shows more stability than 4,5&6.

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