NON LINEAR DYNAMICS OF A FLEXIBLE ROTOR SUPPORTED BY LAMINAR FLUID FILM JOURNAL BEARING WITH MICRO POLAR STRESS FLUID

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ABSTRACT

The examination introduces a powerful investigation of a rotor upheld by two laminar stream model diary course and greased up with couple pressure liquid under nonlinear suspension. The elements of the rotor focus and bearing focus is contemplated. The dynamic conditions are settled utilizing the ODE-45 MATLAB technique. The examination techniques utilized right now comprehensive of the dynamic directions of the rotor focus and bearing focus. The outcomes show that the estimations of dimensionless parameters l^* unequivocally impact dynamic movements of bearing and rotor focus. It is discovered that couple pressure liquid improve the solidness of the framework when l^*< 0.25 regardless of whether the progression of this framework is laminar. We likewise exhibited that the dimensionless rotational speed proportions s and the dimensionless unbalance parameter b are additionally noteworthy framework parameters.

Keyword: - *Non Linear Dynamics, Turbulant flow, Micro polar stress, and Flexible Rotar etc….*

INTRODUCTION

Couple Stress Fluid is a blend of Newtonian liquid and some added substance (long chain natural compound) which is answerable for dependability. At times greasing up liquids of low consistency are utilized as ointments to disentangle the gear structure or improve the numerical reenactment models. Hydrodynamic diary orientation utilizing in rotor fast turbo hardware greased up with fluid metals. Cheng-Ying Lo [1] Nonlinear elements of an adaptable rotor upheld by violent diary orientation with couple pressure liquid .This investigation displays a unique examination of a rotor bolstered by two fierce stream model diary heading and greased up with couple pressure liquid under nonlinear suspension. The elements of the rotor focus and bearing focus is examined. EI Naschie [2] first proposed the idea of confused disturbance. He presented the summed up bifurcation and transient confusion in science and building and afterward continued to show that the limited clasping of flexible shells can be seen all around as a type of unique disorder. He likewise inferred that confined shell clasping can be deciphered as exceptional disturbance of a slender versatile surface simply like the choppiness can be deciphered as uncommon worldly deterministic disorder of a liquid. Gardner and Ulschmid [3] played out an examination for a tilting-cushion and a sleeve diary bearing. Hopf G. What's more, Schuler [4]an try for diary direction greasing up with change streams among laminar and fierce stream systems. Glove and Glienicke [5] proposed a worldwide examination strategy for a Reynolds condition dependent on the observational choppiness coefficients characterized by Constantinescu and they stretched out this worldwide idea to the vitality condition. Hashimoto et al. [6, 7] inspected the impacts of wear on consistent state and dynamic attributes of the hypothetical and test strategies under working conditions.

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.

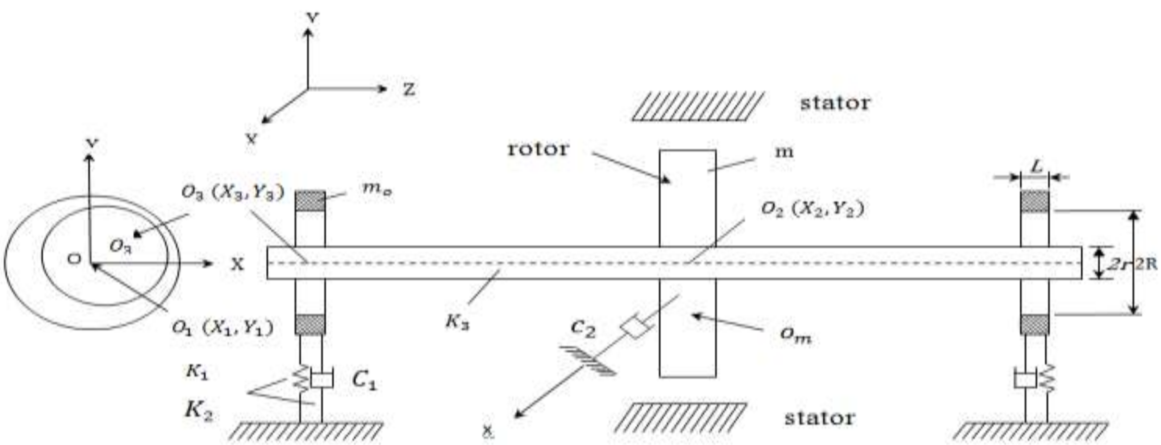


Fig. 1 Model of a flexible rotor supported on two non-linear suspensions

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

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

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

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

$m_0\ddot{X}_1 + c_1\dot{X}_1 + k_1X_1 + k_2X_1^3 = F_x$... (5)

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

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

Thus Reynolds equation can be rewritten as.

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

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

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

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

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

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

$$G_z\&G_\theta=1\text{ , because it is laminar.}$$

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

$$f_\phi=-\frac{\mu L^3R}{2c^2}\int_0^{\pi}\left\{\frac{\left[(\omega-2\dot{\phi})\varepsilon\sin\theta-2\dot{\varepsilon}\cos\theta\right]\sin\theta}{\left[(1+\varepsilon\cos\theta)^3-12(l^*)^2(1+\varepsilon\cos\theta)+24(l^*)^3\tanh\left(\frac{1+\varepsilon\cos\theta}{2l^*}\right)\right]}\right\}d\theta(14)$$

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

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

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

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

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

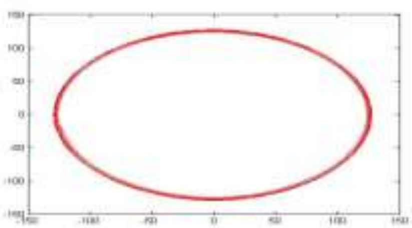
φ Attitude angle

RESULTS AND DISCUSSION

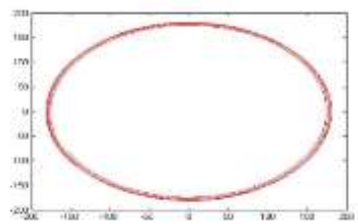
This work has explored the effects of varying the main system parameters of a flexible rotor supported by laminar journal bearings and lubricated with couple stress fluid under nonlinear suspension. The results show that the values of dimensionless parameters l* strongly influence dynamic motions of 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

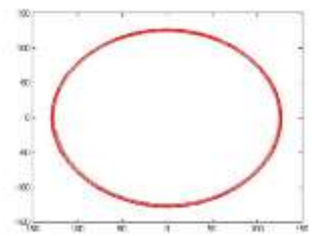
Journal diameter(D)	0.0254m
Length of bearing(L)	0.0127m
Mass of rotor (2m)	5.4523 kg
Radial clearance(C)	50.8µm
Lubricant	ISO 32



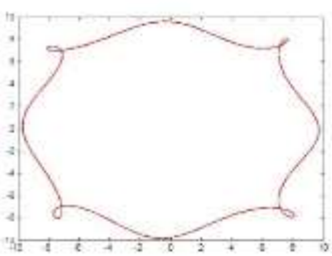
L=0.05, S=1.000
Fig.1



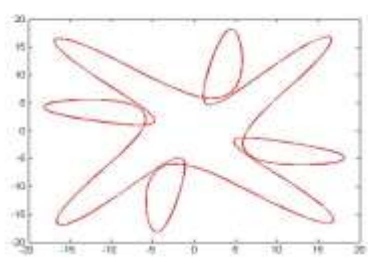
L=0.05, S=5.500
Fig.2



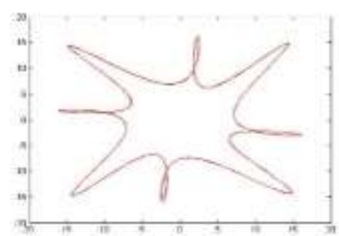
L=0.05, S=7.000
Fig.3



L=0.10, S=2.500
Fig.4



L=0.10, S=4.500
Fig.5



L=0.10, S=5.500
Fig.6

Figure 1,2&3 Shows More Stability Than 4,5&6.

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