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Dynamic Analysis of Rotor-Seal System Considering the Radial Growth Effect of the Seal

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Abstract. Effect of the increase in operating speed, temperature and pressure ratio on turbomachine components must be studied and understood seriously. One of the major challenges is to choose a right initial clearance for seals considering the seal growth as it may affect rotor system dynamic characteristics and leakage performance. In the present study the dynamic response of the rotor system has been calculated considering the seal growth effect at each operating speed. It is been analyzed by the response plots and the orbital plots. The results show that the dynamic response of the system is influenced by radial growth of seal at lower pressure ratio and higher rotational speed. At higher pressure ratio the radial growth of seal has insignificant effect on rotor stability. In addition to these, the present study also emphasizes on the effect of the seal on the overall rotor system by comparing two different rotor-seal system.

1. Introduction

Performance of a turbomachinery is undergoing continuous improvement in present days. The need for high power output and smaller size has increased the constraint on the design aspects of the machine components which is leading researchers to focus more on the optimization of each individual components. Increasing the efficiency has led to these components operating at higher speed, higher temperature and high pressures. Sealing in turbomachinery has been a challenge from past few decades and efficient sealing technology is being developed which will minimize leakages and increase the stability of the rotor. Extensive analytical, numerical and experimental studies have been made in this regard. Labyrinth seals are the most commonly used seal in the turbomachinery due to its simple structure and ease of manufacturing. It's the generic seal configuration based on which different seal configuration are designed [1]. The main design parameter when assembling the labyrinth seal into the machine is the initial clearance between the seal and the stator. The effect of rotation and temperature on the seal clearance and subsequent effect on the leakages have been thoroughly studied recently by Subramanian et al.[2].

One of the earliest works in labyrinth seal by Alford [3] studied about the disturbing forces due to circumferential variation of static pressure within the seal and forces due to rotor eccentricity due to change in blade tip clearance causing whirling of compressor and turbines. Wyssmann et al. [4] presented a theory based on ideal gas behavior of the fluid in labyrinth seal, prediction of seal rotor



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dynamic coefficients has been done for a centrifugal compressor labyrinth seal. The calculations are based on turbulent flow assumption and influence of seal geometry, operating condition and gas molecular weight has been analyzed.

An Iwastsubo-based solution for compressible flow in labyrinth seals for predicting the rotor dynamic coefficients has been obtained by Childs and Scharrer [5]. The model predicted the seal coefficients within 25% of the measured test data for a straight through teeth on stator seal. Dietzen and Nordmann [6] showed that it is possible to calculate dynamic coefficient of seals by finite difference techniques based on the Navier – Stokes equation on a turbulent model. The derived values were compared with those obtained by Childs[7] and found to be in good agreement with experimental data. Scharrer [8] developed a new two control volume method for predicting rotor dynamic coefficients in labyrinth seals. The model predicted both cross – coupled stiffness and the direct damping better than theory of Childs and Scharrer [5].

Consideration of seal forces to predict the non linear response of the rotor bearing system has been studied by few researchers in past. Hua et al. [9] established a non-linear model of the rotor-seal system using Muzynska's non linear seal forces. A direct integration scheme is used to solve the equations. The influence of seal was analyzed using bifurcation diagram and Poincare maps. Cheng at al. [10] investigated the nonlinear dynamic response of a rotor-seal -bearing system. Influence on parameters such as rotation speed, seal clearance and eccentricity of rotor were analyzed using Poincare maps, frequency spectra and bifurcation diagram. Shen and Zhao [11] studied the effect of seal force on stability of rotor-bearing- seal system theoretically and experimentally. Using numerical calculations, effects of different parameter including air pressure difference is investigated and analyzed with the help of bifurcation diagram , rotor orbits and waterfall diagram. Xin at al. [12] presented a transient CFD procedure to compute the non linear dynamic characteristics of the rotor-seal system. The effects of the rotational speed and pressure ratio on the vibration characteristic of the rotor-seal system were analyzed by the bifurcation theorem. They concluded that pressure ratio has effect on frequency response of the first critical speed.

Zhigang et al. [13] determined the effects of operating condition and geometrical parameters on the leakage performance and rotordynamic characteristics of labyrinth seal. Effects of pressure ratio, operating speed and inlet preswirl were investigated using a novel CFD method for a tooth on rotor labyrinth gas seal.

Zhang et al [14] investigated the parametric influence of rotational speed, eccentricity, foundation movement and inlet pressure on the nonlinear dynamic response of a rotor- bearing- foundation- labyrinth seal system. The results are analyzed using spectrum cascades, whirl orbits and Poincare' maps. Recently, Zhang et al [15] also investigated the effect of radial growth on rotor dynamic characteristics of labyrinth seal rotor system. They investigated for one pressure ratio and considered the rotational effect and temperature effect on the system stability.

As seen from previous references most of the studies have been involved with finding the rotor dynamic coefficients of the labyrinth seals. Few of them have investigated on the dynamic response of the rotor- bearing – system using the computed rotor dynamic coefficients. Effect of radial growth has been considered quiet recently but has been investigated only for a single pressure ratio. Subramanian et al [16] have computed numerically the rotor dynamic coefficient of a rotating seal considering centrifugal growth for different pressure ratios and speeds. The present study is using these coefficients in the analytical model to investigate the dynamic response of the rotor- seal- bearing system. So, the impact of wide range of pressure ratio and rotational speed is understood on the system response and stability.

2. Analytical Model

2.1 Seal Configuration

For the present investigation a teeth-on-rotor straight labyrinth seal configuration has been chosen. This generic configuration has been commonly and actively used by several researchers. The physical

configuration of the seal configuration under consideration is shown in Figure. 1. It is a six – tooth straight- through labyrinth seal having teeth-on- rotor.

The seal parameters are taken from the study by Subramanian et al [14]. For this study we are using the rotor-dynamic coefficients calculated by Subramanian et al [14] for these seal parameters. The parameters are given in Table.1. The clearance between seal and the stator, C_r has been taken as 0.50 mm. This is the cold clearance or the clearance at ambient temperature.

2.2 Rotor - Seal - Bearing Model

To determine the effect of radial growth of seal on the response of the turbomachinery a model of rotor, seal and bearing is taken as shown in the Figure 2. In this model we will consider the fluid forces from the seals. The model is a simplified version of a labyrinth seal disc mounted on a shaft on the center location and supported on bearing at the two ends.

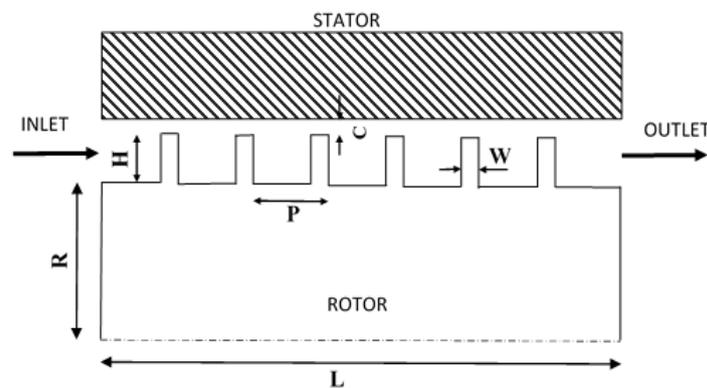


Figure 1. Seal Configuration

Table 1. Seal Parameter Values

Seal Parameter	Value
R	50 mm
P	12 mm
H	10.5 mm
W	2.5 mm
C_r	0.5 mm
L	100 mm

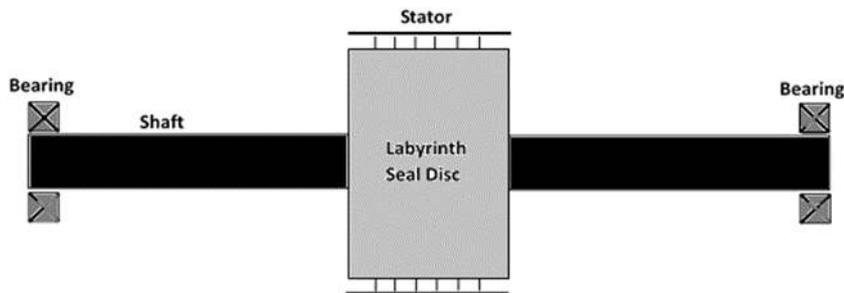


Figure 2. Rotor - Seal - Bearing Model

The equations of motion in differential form for above system can be written as following.

$$m_b \ddot{X}_b = -D_b \dot{X}_b - K_s (X_b - X_d) + F_{bx} + m_b e_b \omega^2 \cos \omega t \tag{1}$$

$$m_b \ddot{Y}_b = -D_b \dot{Y}_b - K_s (Y_b - Y_d) + F_{by} + m_b e_b \omega^2 \sin \omega t - m_b g \tag{2}$$

$$m_d \ddot{X}_d = -D_{seal} \dot{X}_d - K_s (X_d - X_b) + F_{sx} + m_d e_d \omega^2 \cos \omega t \tag{3}$$

$$m_d \ddot{Y}_d = -D_{seal} \dot{Y}_d - K_s (Y_d - Y_b) + F_{sy} + m_d e_d \omega^2 \sin \omega t - m_d g \tag{4}$$

X_b and Y_b are response of journal and X_d and Y_d are the response of the seal disc in X and Y directions respectively. F_{bx} and F_{by} are the oil film forces acting on the journal and F_{sx} and F_{sy} are the seal forces included in the equations. K_s is the stiffness of the shaft.

D_b and D_{seal} are the damping coefficients of journal and seal disc respectively. m_b and m_d are the lumped masses located at the journal center and disc center respectively. e_b and e_d are the eccentricities of the journal and the disk. For the present study we will consider the supports as rigid, i.e $X_b = Y_b = 0$. So, our equations (1) to (4) will reduce to the following form :

$$m_d \ddot{X}_d = -D_{seal} \dot{X}_d - K_s(X_d) + F_{sx} + m_d e_d \omega^2 \cos \omega t \tag{5}$$

$$m_d \ddot{Y}_d = -D_{seal} \dot{Y}_d - K_s(Y_d) + F_{sy} + m_d e_d \omega^2 \sin \omega t - m_d g \tag{6}$$

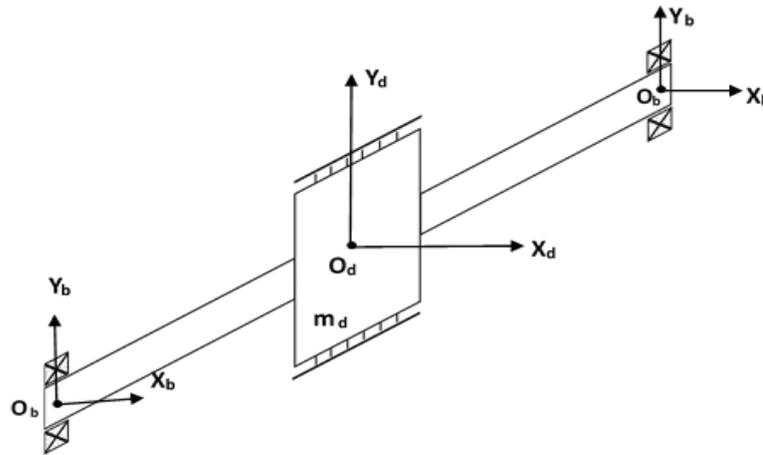


Figure 3. Model for Analytical Study

The geometrical parameters and operating condition of the rotor seal system used in the present numerical study has been given in Table 2.

Table 2. Seal material properties

Parameter/Properties	Value
Material	Inconel 718
E	200 GPa
γ	0.294
ρ	8221 kg/m ³
T_{ref}	21° C
PR	1.1, 1.5, 1.9, 2.5
e_d	50 μ m
ω	0 – 5000 rad/s

2.3 Seal Forces Calculation

For small circular whirling motion of a rotor about a centered position, the rotor-dynamic coefficients are commonly defined by a simple linearized force displacement model. A second order model equation is given below.

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \begin{bmatrix} M & m \\ -m & M \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} + \begin{bmatrix} C & c \\ -c & C \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} + \begin{bmatrix} K & k \\ k & K \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} \tag{7}$$

$$\frac{F_x}{e} = M\Omega^2 - c\Omega - K \tag{8}$$

$$\frac{F_y}{e} = -m\Omega^2 - C\Omega + k \tag{9}$$

Here F_x and F_y corresponds to reaction force components of seals along x and y direction respectively. M , m , C , c , K and k represent the direct inertia, cross coupled inertia, direct damping, cross coupled damping, direct stiffness and cross coupled stiffness of the seals. Using these rotor dynamic coefficients seal forces are computed which are then used in solving the differential equations. The e , eccentricity is taken as $50 \mu\text{m}$ which is 10% of the initial cold clearance. Synchronous whirling is assumed throughout the present study.

2.4 Rotor sizing

As the main aim is to understand the effect of the seal on the dynamic response of the system, the sizing of the rotor should be such that its response value does not exceed the cold clearance between the seal disc and stator. The speed of interest for the present study is from 0 – 5000 rad/s. Two Jeffcott rotor with different shaft length and shaft diameter as shown in Figure 4. have been studied. It has been modelled in ANSYS 17.0 and modal analysis has been done to identify the critical speeds and maximum displacement of the rotor disc center.

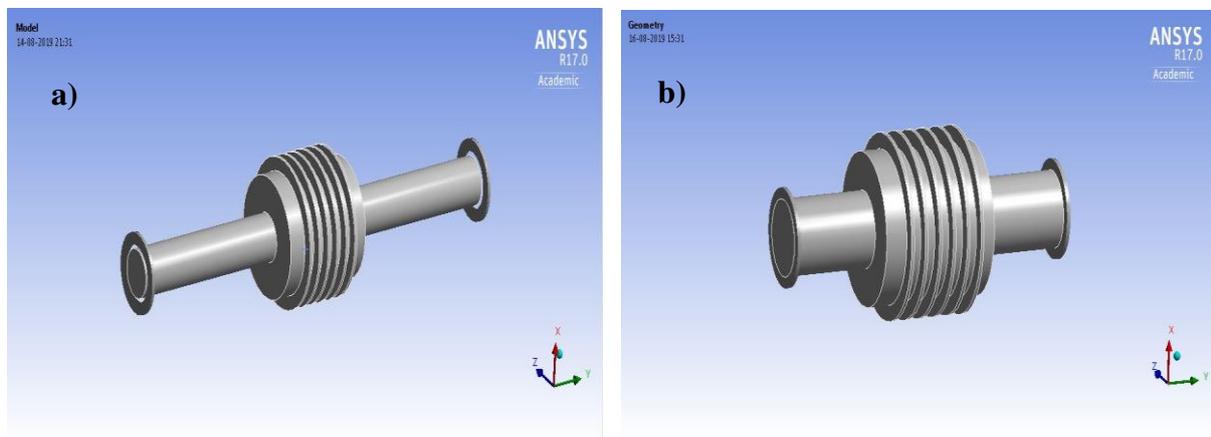


Figure 4. ANSYS Model for (a) Configuration I and (b) Configuration II

Rotor configuration I shown in Figure 4a. has shaft length of 450 mm and shaft diameter of 40 mm. The configuration has one critical speed of about 1500 rad/s between 0 to 5000 rad/s and the maximum displacement of disc is 0.019 mm at that critical speed which is less than the cold clearance of 0.5 mm. Thus, making sure that the disc does not rub the stator when passing through the critical speed.

Similarly, for the rotor configuration II shown in Figure 4b. the shaft length is 250 mm and shaft diameter is 40 mm. This configuration has no critical speed between 0 to 5000 rad/s. These two configurations are considered as to make sure the effect of seal is prominently seen in the dynamic response.

3. Results

3.1 Response of Rotor Configuration I

Rotational speed plays an important role in controlling the turbomachinery. The differential equation for the motion has been solved for the Jeffcott rotor system for the range of 0 – 5000 rad/s considering the seal forces without seal centrifugal growth and seal forces with seal centrifugal growth (CG) for four different pressure ratios respectively. The system is operating at ambient temperature of 21°C for the present study.

For the configuration I rotor, the response v/s speed plots for PR = 1.1 to 2.5 are shown in Figure 5a. to 5d. respectively. For PR = 1.1, in Figure 5a. the rotor response without considering CG rises gradually at low speed and a peak is observed at the first critical speed of around 1410 rad/s. As speed approaches near 3000 rad/s we can observe the response rising drastically and between 3000 – 3500 rad/s it reaches a maximum value of about 0.1 mm. Further increase in the speed i.e. after 3500 rad/s

the response gradually decreases. So, at this low-pressure ratio we can expect the rotor to undergo instability at speed range of 3000 – 3500 rad/s. But the max value of the response in this range is less than 0.1 which is much lesser than the cold clearance value of 0.5 mm.

When we consider the seal CG and observe the response at PR = 1.1, the response curve follows the similar trend as the response without CG till the speed of 2500 rad/s, after which the response increases drastically and the response value reaches the maximum of about 0.45 mm between range of 2700 – 4000 rad/s and after that gradually decreases. The response in this range is higher than the corresponding response values when CG is not considered. The maximum response value of 0.45 is almost nearing the cold clearance value of 0.50 mm which indicates that there is possibility of rotor – stator rub. Also, the instability range of speed has increased to 2700 – 4000 rad/s.

For higher PR of 1.5, 1.9 and 2.5, as shown in Figure 5b. to 5d. it can be observed that the response increases as speed is increased, reaches a maximum at critical speed of 1410 rad/s and then gradually decreases. This trend is same for response without CG and response with CG. There is no noticeable instability region in this range of speed. So, it can be deduced that the effect of seal forces is negligible at higher pressure ratios.

The variation in the response values at critical speed has been shown in Figure 6. The response value follows a decreasing trend with increase in pressure ratio. At PR = 1.1 the critical speed response value is almost same for response without CG and response with CG. At PR = 1.5 the critical speed response value for response without CG is less than response with CG. But as we increase the PR to 1.9 there is slight increase in the critical speed response value without CG, but the critical speed response value with CG decreases and becomes less than the response value without CG. And at PR = 2.5 the response value for both case although decreasing again becomes almost equal.

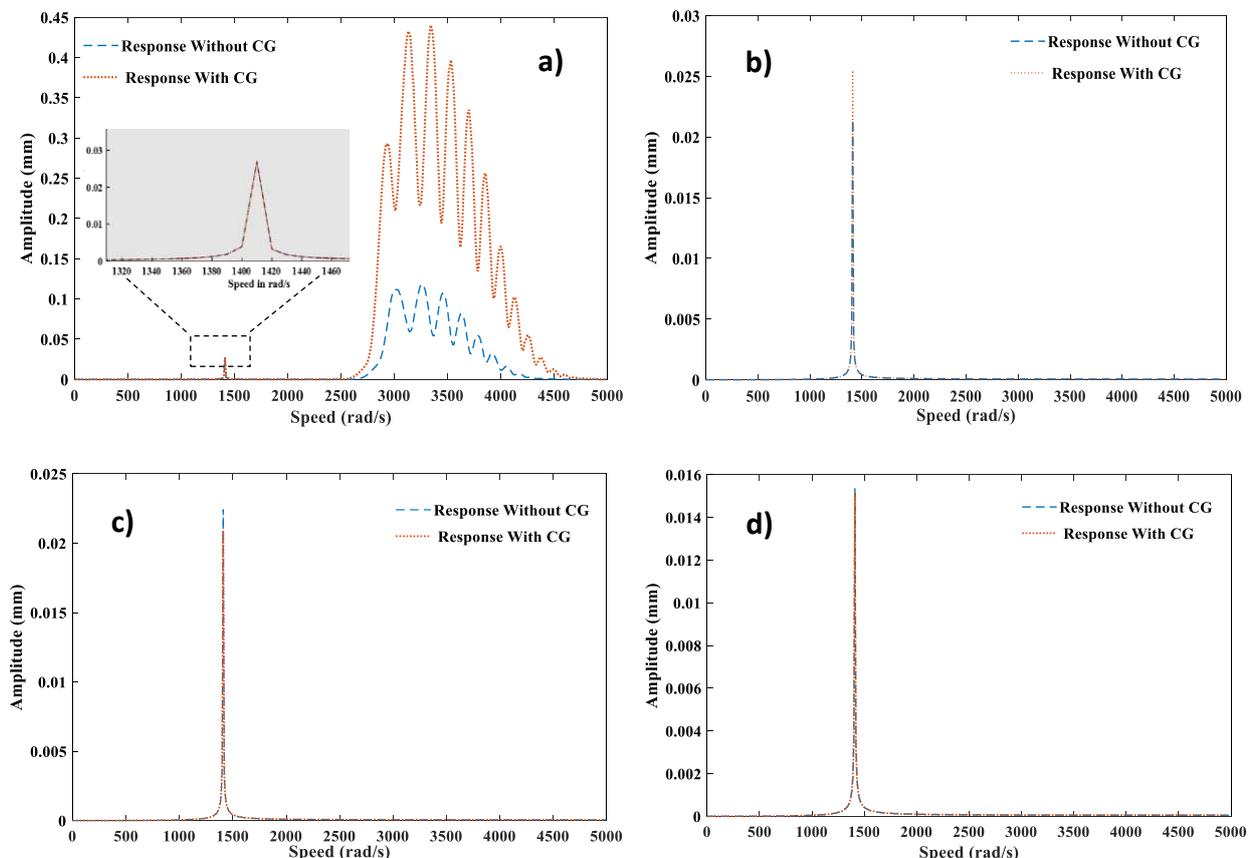


Figure 5. Response v/s speed plot without CG and with CG for (a) PR = 1.1, (b) PR = 1.5, (c) PR = 1.9 and (d) PR = 2.5

3.2 Response for rotor configuration II

For configuration II rotor – seal – bearing system the response v/s speed plot for PR = 1.1 to 2.5 are shown in Figure 7a. to 7d. respectively. As already discussed in rotor sizing section, there is no critical speed for this configuration in the speed range of interest. For PR = 1.1, as seen in Figure 7a. the response without CG increases gradually with speed and starts increasing drastically after the speed of 2500 rad/s and response value reaches a maximum of 0.0025 mm between 3000 – 4000 rad/s and decreases. It can be observed that the value of maximum response is much lesser than the cold clearance value of 0.5 mm.

The corresponding response with CG follows the similar trend for PR = 1.1. The response increases after 2500 rad/s and reaches a maximum value of 0.011 mm between 3000 – 4000 rad/s. Response starts to decrease after 4500 rad/s. In both the condition we can see that there is a range of speed where rotor demonstrating instability. But the value of response is very much low then the cold clearance value.

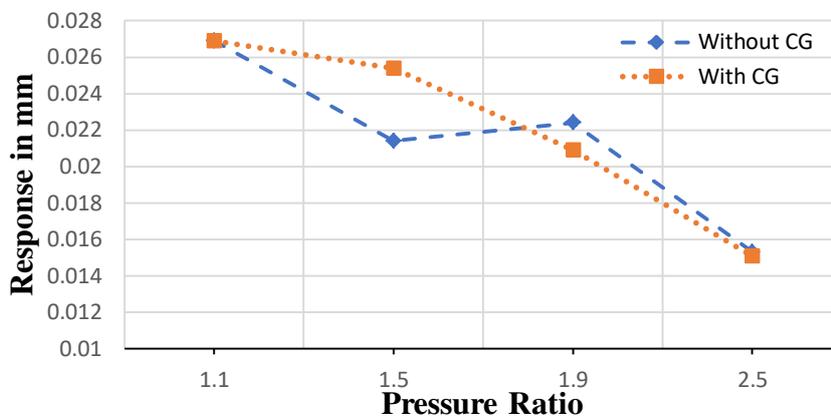
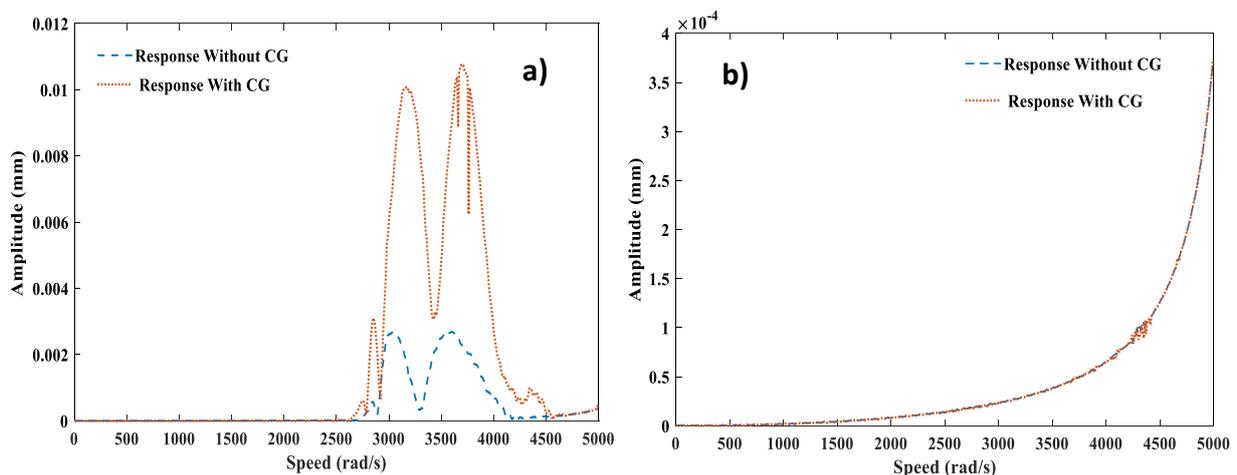


Figure 6. Variation of critical speed response values with pressure ratio for Configuration I rotor



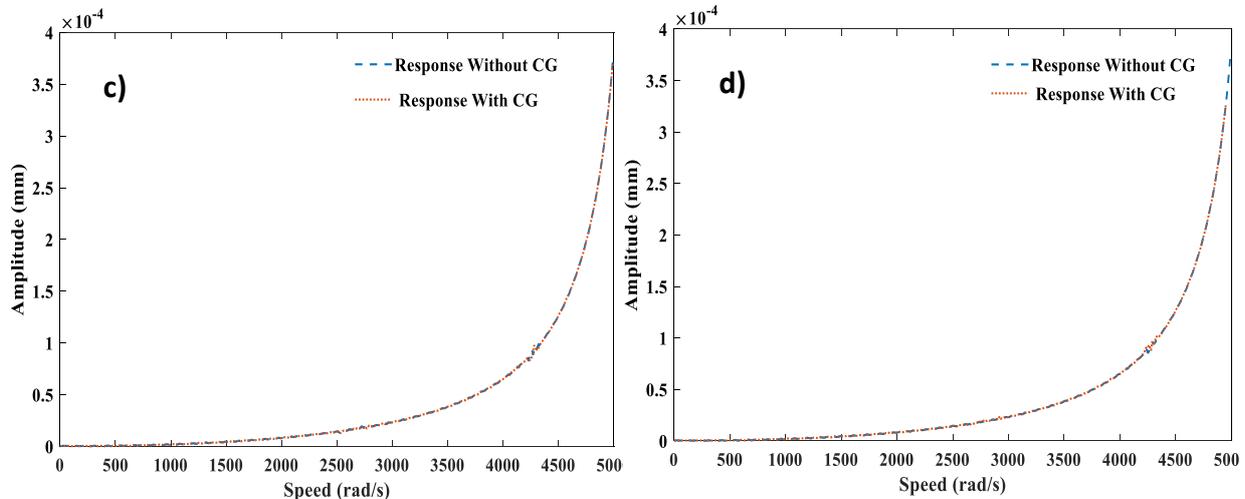


Figure 7. Response v/s speed plot without CG and with CG for (a) PR = 1.1, (b) PR = 1.5, (c) PR = 1.9 and (d) PR = 2.5

For higher pressure ratios of 1.5, 1.9 and 2.5 the response plots as shown in Figure 7b. to 7d. respectively,

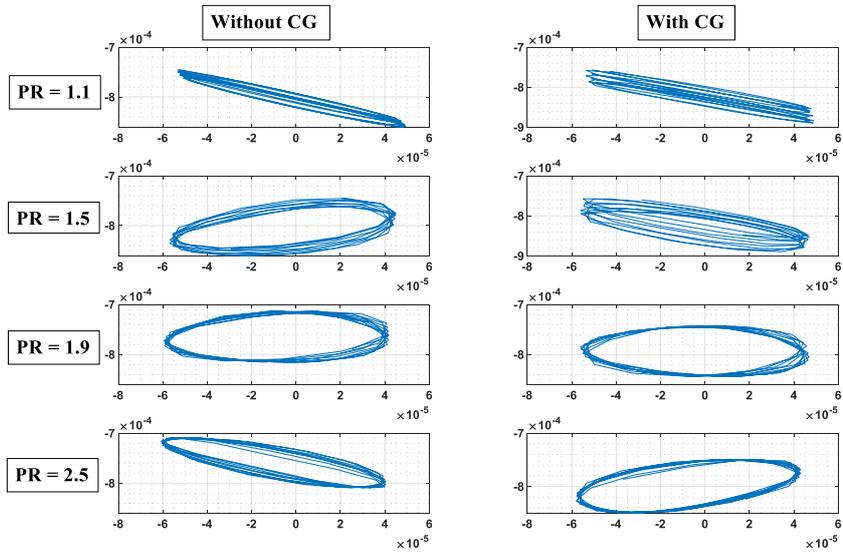
the response with CG and without CG almost follows the same trend. The response values are of the order 10^{-4} mm which are much lower. Therefore, the short rotor configuration is not affected by the seal forces due to centrifugal growth at high pressure ratios. At low pressure ratio of 1.1, visible change in response can be seen with and without CG.

3.3 Orbital Plots

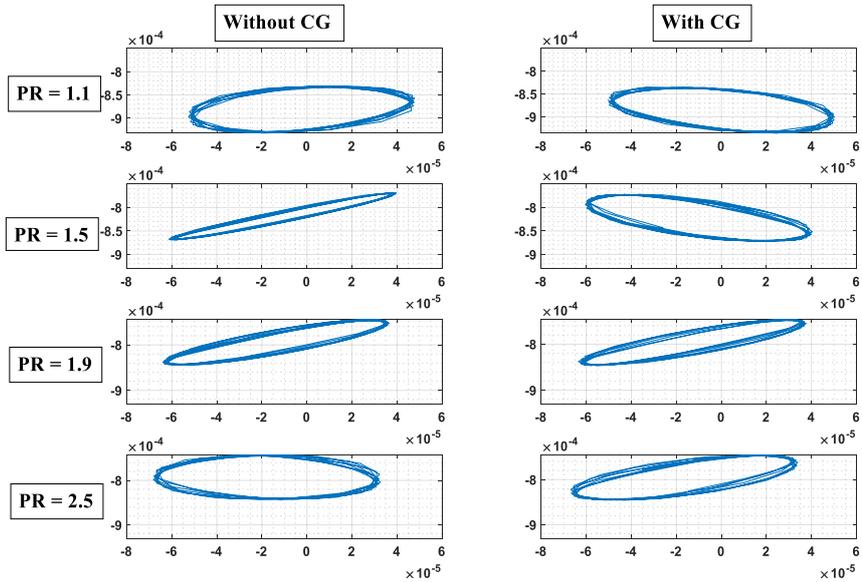
Between the two rotor configuration, the configuration I has exhibited influence of seal forces when centrifugal growth is considered. In order to understand the rotor behavior due to these effects, orbital plots can be helpful. Orbital plots for the center of the seal disc are shown in Figure 8a. to 8d. for four different rotational speed of 1000 rad/s, 2000 rad/s, 3000 rad/s and 4000 rad/s respectively. And for each rotational speed orbital plots have been displayed for four different pressure ratios of 1.1, 1.5, 1.9 and 2.5 without considering CG and with considering CG respectively. The change in orbitals plots can be easily identified with different operating parameters.

From Figure 8a and 8b, orbital plots for 1000 rad/s and 2000 rad/s we can observe that as pressure ratio is increasing the shape of orbit is becoming more stable. This is more prominent for the 1000 rad/s speed. At higher speed of 3000 rad/s the effect of CG is still more visible. In Figure 8c. the orbital plot for low PR = 1.1 is blown up when CG is considered. As we increase the PR the orbital plot is becoming more and more stable. Similarly, at even higher speed of 4000 rad/s as shown in Figure 8d. the orbital speed at low PR = 1.1 is haphazard when CG is considered. And as PR increases the orbit shape becomes clearer and more stable.

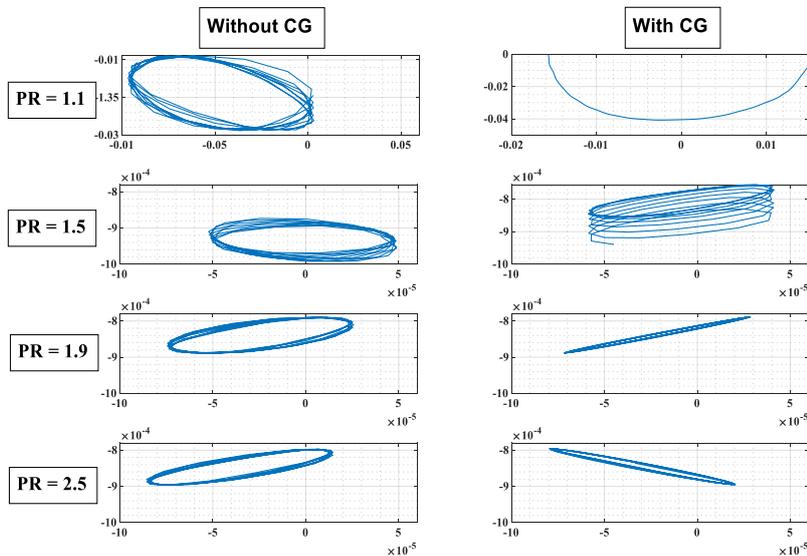
a)



b)



c)



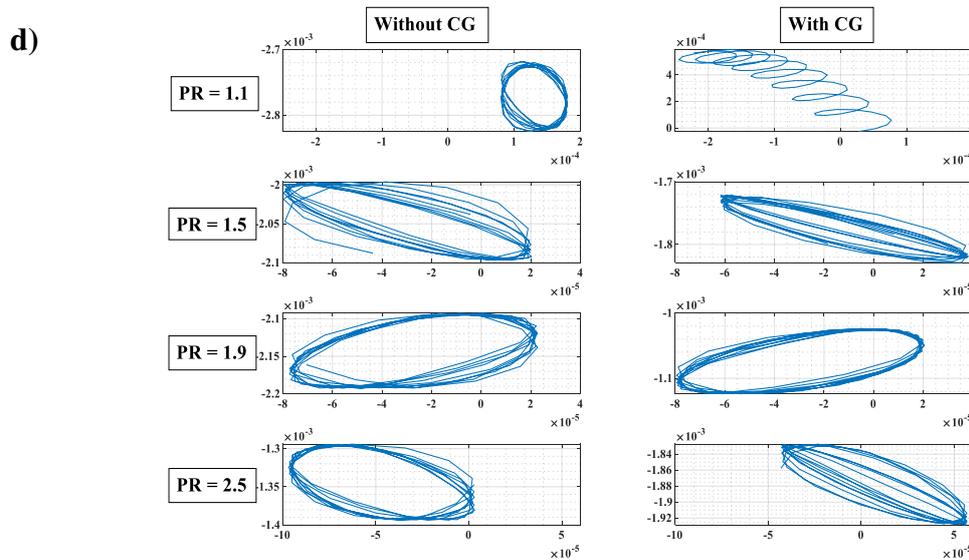


Figure 8. Orbital Plots for Configuration I rotor for different pressure ratio and speed (a) 1000 rad/s, (b) 2000 rad/s, (c) 3000 rad/s and (d) 4000 rad/s

4. Conclusion

The radial growth of seal due to rotation has been considered and the change in these seal forces due to the centrifugal growth has been incorporated in the analytical model. The differential equation for the motion for the center of disc has been solved for two different configuration of Jeffcott rotor with rigid bearing assumptions. The dynamic response and orbital plot have been presented and analyzed for different operating parameters – rotational speed and pressure ratio. Following conclusion can be drawn from this analytical study:

- Centrifugal growth of the seal influences the rotor only at very high speed and low-pressure ratios. At higher pressure ratios the influence of seal forces due to CG is negligible.
- The response plots clearly show the instability region of the rotor. At low pressure ratio there is more possibility of rotor stator rub when seal growth is taken into consideration for the given initial clearance. The effect of seal growth reduces as higher-pressure ratios are considered.
- The rotor size plays an important role when considering the seal design. For a much short rotor, the effect of seal growth vanishes at high pressure ratios. Also, at low pressure the effect is negligible when compared to values to cold clearances.
- The orbital plot clearly shows the rotor disc behavior in the instability speed range. At low pressure ratio the orbits are random and as pressure ratio is increased the orbit shape becomes stable.

The outcome of present analytical study emphasizes on the consideration of using seal forces due to centrifugal growth in finding the overall dynamic response of the rotor. The changes in rotor dynamic coefficients have been combined in an analytical model to have a much better understanding at the system level influence of the seals.

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