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Stability Improvement for Upgraded Four Stage Centrifugal Compressor Rotor-Bearing System

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Abstract

As the plant capacity is increased, the equipment in the plant also needs to be upgraded. Replacing with a new machine is not always the right economical decision when alternative options for upgrading and retrofitting the exiting machine exist. Upgrading the machine not only increases the performance thus reducing the power consumption but it also increases the system reliability, reducing the downtime and thus helping the plant operate more efficiently. The compressor of interest for this paper was in operation and had degradation on the impellers due to fouling caused by process contaminants. The existing multi-stage centrifugal compressor was upgraded for increased inlet volume flow rate of 20% with the objective of retaining the same casing and rotor shaft. The compressor stages were redesigned and performance was improved in terms of reduced power consumption.

The redesigned four stage centrifugal compressor operating at design speed of ~8856 rpm may be prone to instabilities caused by aerodynamic cross-coupling and hydrodynamic bearing modification. The authors in the present study are exploring any instabilities in the rotor/bearing system that may be caused due to the upgrade of the compressor for higher volume flow rate. The bearings were redesigned to meet the new operational requirements. The stability analysis was performed considering the destabilizing effects of the aerodynamic excitations, based on which modifications were performed on the design to stabilize the multi-stage centrifugal compressor and to conform to satisfy the API Level I stability criteria. The modifications performed on the rotor /bearing system resulted in reduced vibration amplitudes for the required level of rotor unbalance conditions. These modifications also significantly eliminated the instability that was initially observed and the redesigned compressor not only met the requirements for the change in operating conditions but also fully complied with API 617 standards.

Keywords: Destabilizing forces, Rotor/bearing stability, bearing sensitivity study, API, Retrofit.

Nomenclature

API	American Petroleum Institute	*x amp*	*x*-direction amplitude
ISO	International Organization of Standardization	*y amp*	y-direction amplitude
		m	Preload
FEM	Finite element method	γ	Offset
FDM	Finite difference method	αLE	Orientation angle
CS	Critical speed	$d\varphi 1$	Axial groove
UCS	Undamped critical speed	*P*	Load

FB	Front bearing	*Hmin*	Minimum oil film thickness
RB	Rear bearing	n_o	Operating speed
AF	Amplification factor	*Nfr*	Power loss
MOFT	Minimum oil film thickness	*Fr*	Friction factor
L	Axial length	ΔT	Temperature rise
D	Journal diameter	C_b	Bearing clearance
L/D	Length-to-diameter ratio	C_p	Pad clearance

1 Introduction

With increase in plant capacity, there is a need to upgrade different equipment's used in the process. Centrifugal compressors are one of such components which can be upgraded by redesigning the flow path while retaining most of the major parts like casing and rotor therefore keeping costs relatively low. When the system is upgraded for increased pressure or mass flow rate, additional challenges arise to ensure the reliability of the system as well as to meet the applicable API standards. In the present work, a four-stage centrifugal compressor was retrofitted by redesigning the impellers for an increased volume flow rate [1]. The aerodynamic efficiency of the compressor has been improved and is reported in [1]. In addition to improving the specific power consumption of the compressor the mechanical losses also need to be reduced. Since there is an increase in the volume flow rate, the power consumption of the compressor has also increased necessitating studies on the rotor-bearing system. The rotordynamics studies were performed using AxSTREAM Rotordynamics™ at first for the existing bearing. As the rotor was found to be unstable with the existing bearings, it was decided that the bearings had to be redesigned. A sensitivity study is required not only to reduce the mechanical losses in the bearings, but also to make ensure that the designed system meets the API standards. An optimization of bearing design variables employed the power loss in bearings, stability criteria, and unbalance responses as a set of objective functions [2-6]. Recent studies with bearing optimization are performed using sensitivity analysis based on DOE method [2,3]. Since the retrofitted multi-stage centrifugal compressor rotor is to be used, there is little scope for modifications in the rotor and all studies are focused mainly on the bearing design [1]. To reduce frictional losses and power losses, the minimum film thickness needs to be increased [6] and at the same time the temperature of the oil film should be reduced. The critical speed separation margin is to be maintained as per the API standards. The present study first focuses on the existing elliptical bearing that was optimized to avoid high temperatures at the bearings and instability in the rotor-bearing system. Since the requirements were not satisfied even with the optimized elliptical bearings, tilting pad bearings were considered as the next alternative and optimized. To understand the effect of different parameters on the stability, a set of variables was chosen, based on which a sensitivity study plan was created. The sensitivity analysis was performed manually without considering an optimization algorithm.

2 Rotordynamics analysis of baseline rotor/bearing system

The rotor geometry in Fig. 1 was modelled as beam elements with 47 nodes in AxSTREAM Rotordynamics™. The impellers were modeled as lumped mass-inertia elements with four degrees of freedom (two displacements and rotations in the lateral plane). The existing elliptical bearing was independently analyzed using AxSTREAM Bearing™ which is based on FDM and uses Elrod & Adams model [2,3] with a successive over-relaxation (SOR) algorithm. The front and rear bearing journals have diameters of 149 mm and 126 mm, respectively, with preload of 0.34, length of 80 mm and 68 mm, respectively, and vertical clearance of 0.11175 mm and 0.0945 mm, respectively. The bearing gravitation loads are 4107 N and 4067 N for the front and rear bearings, respectively. The lubricant used is ISO 32 with temperature-dependent properties. These bearings were designed to operate up to 15°C temperature rise. The inlet oil temperature is 50°C and the design bulk oil discharge temperature is 65°C. Since the strength of the Babbitt material reduces at elevated temperatures, the maximum temperature should not exceed 85°C. The results of the hydrodynamic and mechanical characteristics for the rated speed and other rotational speeds as applicable were also calculated and are presented in Table 3.

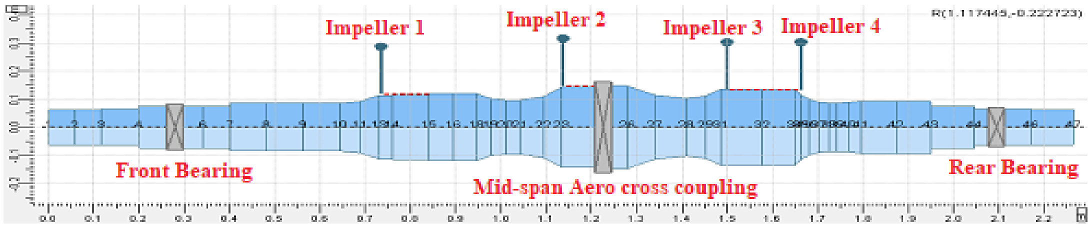


Fig. 1 Rotordynamics FE model

2.1 Synchronous Unbalance Response Analysis of baseline design

The rotordynamics analysis showed that the first three undamped critical speeds of the machine supported by flexible bearings are within the operating speed range. The accurate prediction of unbalance response for the rotor-bearing system is difficult because the distribution of unbalance mass along the rotor is usually unknown [7]. In order to excite a mode shape, a certain set of unbalance masses was attached to the rotor and the unbalance response was calculated in the operating regime. In this study, at 0.5586 kg mm of unbalance, the first bending mode on bearing stiffness x-axis direction was excited as calculated according to API 617/684 requirements [8] and applied at station 13 (see Fig. 2: case 1). The second bending mode on bearing stiffness x-axis direction was excited and applied at station 24 as shown in (see Fig. 2: case 2). Similarly, the first conical mode on bearing stiffness x-axis direction was excited by two equal unbalance masses shifted 180 degrees out of phase and applied at stations 1 and 47 (see Fig. 2: case 3). The conical mode is caused in the system due to lower bearing stiffness present in the y-axis direction.

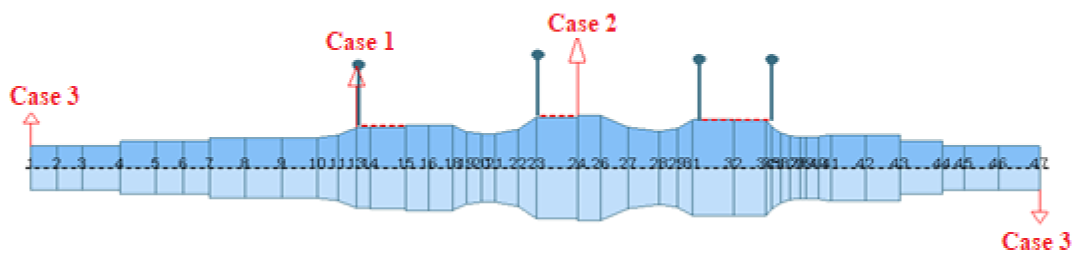


Fig. 2 Unbalances to excite mode shape cases

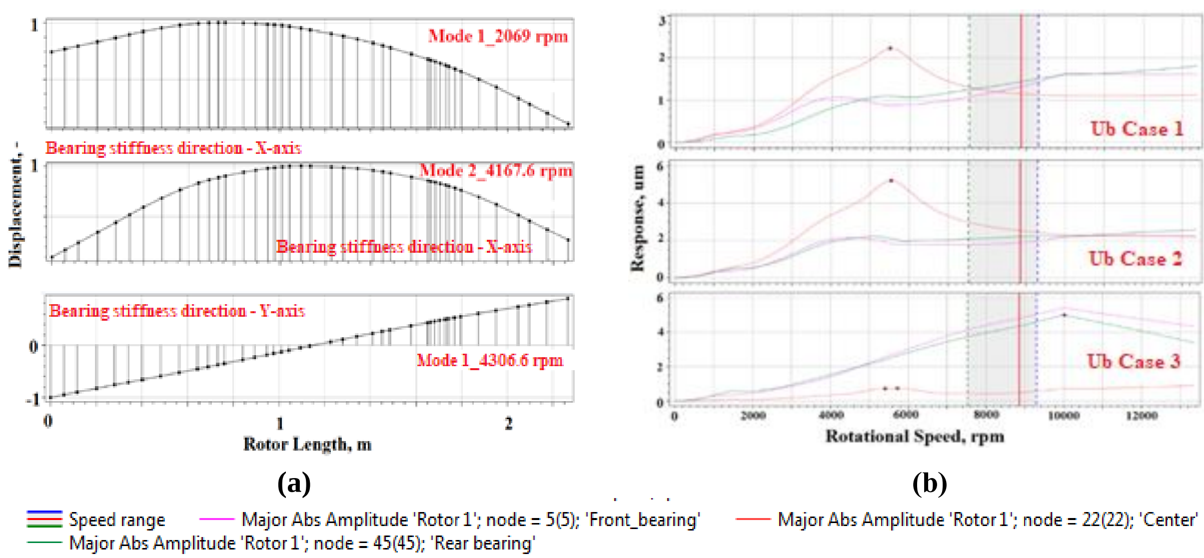


Fig. 3 (a) Undamped critical speed mode shapes and (b) Frequency response amplitude (peak-to-peak, µm) for initial existing rotor/bearing system

The damped unbalance responses were evaluated for speeds starting from 0 rpm to 13367 rpm in steps of 300 rpm using corresponding speed-dependent bearing coefficients. The rated speed was 8856 rpm. For the maximum amplitudes of the major elliptical whirling orbits, upper limit constraints were mentioned and the unbalance was calculated for the first case at the front and rear bearing locations (1.26 µm and 1.39 µm, respectively) and shaft mid span location (1.14 µm). For the second unbalance case, the response amplitude at the front and rear bearing locations (1.72 µm and 2.06 µm) and shaft mid span location (2.52 µm) were calculated. Similarly, the third unbalance case response amplitude at the front and rear bearing locations (4.66 µm and 4.25 µm, respectively) and shaft mid span locations (0.48 µm) were calculated. Baseline unbalance results are presented in Fig. 3b and their corresponding undamped critical speed mode shapes (eigenvectors) are shown in Fig. 3a.

2.2.1 Stability analysis of baseline design

A necessary condition for the rotor-bearing system is to be stable within the operational speed range which is mentioned in the last paragraph. Indeed, an unstable system would generate high level of vibrations which may severely damage the equipment. If sufficient damping is available to counter the destabilizing forces generated by seals and aerodynamic cross-coupling, the rotor may become stable even though there exist some instabilities in the rotor-bearing system. The cumulative summation of cross-couplings values on each stage are calculated and added at the rotor mid-span for between bearing rotors like the current one [8]. However, if damping is insufficient, the instability in a rotor will cause the rotor-bearing system to be unstable [9]. The stability of the rotor-bearing system is checked by using the "logarithmic decrement" (defined as the natural log of the ratio of two successive vibration amplitudes) stability criterion of API 684 [8]. As a design criterion, it is desirable to have a log decrement greater than or equal to +0.1 [8] for a stable behavior to exist. The stability results are listed in Table 4 for the first four forward and backward whirl modes. The results show that the second mode is at 61.34 Hz with a log decrement of -0.43 resulting in an unstable rotor. This requires some modification in the existing design which is described in the following section. The first, third and fourth modes are very stable thus satisfying the API requirements.

The mode shapes (eigenvectors) for the second mode are illustrated in Fig. 4.

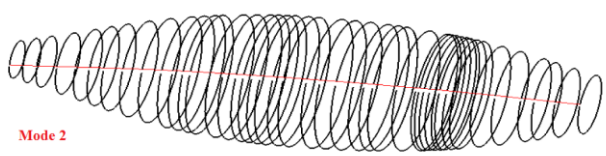


Fig. 4 3D mode shapes for initial existing rotor/bearing system design, second damped natural frequencies at 8856 rpm.

3 Bearing Sensitivity Study:

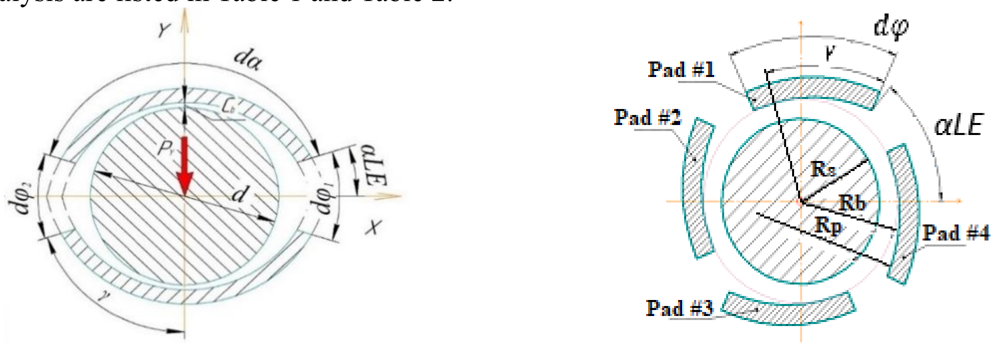
To overcome the vibration and reliability challenges, the rotor-bearing system needs to be modified such that the system becomes stable. Additionally, it is also desired to reduce the power losses so that the overall system efficiency can be improved. It was decided to perform a sensitivity study by varying selected geometrical parameters of the bearing without any changes to the rotor [2,3]. The main design variables used for the sensitivity analysis are L/D ratio (bearing axial length to journal diameter), pad offset factor, pad preload factor, orientation angle, angular groove extent and bearing radial clearance. The pad axial length to journal diameter ratio has an effect on fluid-induced instability as well as on load capacity. When the ratio is increased, the load carrying capacity increases [10,11]. With increasing bearing pad offset factor, the pad pivot position moves from the center of the pad toward the trailing edge. It is a common practice in the industry to provide journal pads with preload. The relationship between the two clearances C_b and C_p, namely bearing radial clearance and bearing pad clearance, is defined as preload. C_p is the difference of the radius of curvature of the pad, r_p, and journal radius, r_s. The preload factor can be expressed as [12].

$$m = (1 - C_b/C_p)$$

where C_p and C_b can be computed as:

$$C_p = r_p - r_s; \qquad C_b = r_b - r_s$$

The sensitivity analysis was performed using the following objective functions - minimum film thickness, maximum oil film temperature rise, power loss, friction factor, etc. Two different bearings were used in the present study: one elliptical bearing and the other a tilting pad bearing as shown in Fig. 5. The variables for sensitivity analysis are listed in Table 1 and Table 2.



(a) **(b)**

Fig. 5: Bearing schemes: (a) Elliptical and (b) Tilting pad Bearings

Table 1 Ranges for the design variables of the elliptical bearings

		Front Bearing		*Rear Bearing*	
Parameters		*Lower Limit*	*Upper Limit*	*Lower Limit*	*Upper Limit*
Bearing Length, mm	*L*	74.5	96.85	63	81.9
Length to Diameter ratio	*L/D*	0.5	0.65	0.5	0.65
Bearing Clearance, mm	C_b	0.11175	0.2682	0.0945	0.2268
Preload	*m*	0.34	0.56	0.34	0.56
Offset	γ	0.5		0.5	
Orientation, deg	αLE	9	25	9	25
Axial Groove, deg	$d\varphi 1$	45	50	45	50

Table 2 Ranges for the design variables of the tilting pad bearings

		Front Bearing		*Rear Bearing*	
Parameters		*Lower Limit*	*Upper Limit*	*Lower Limit*	*Upper Limit*
Bearing Length, mm	*L*	80	149	68	126
Length to Diameter ratio	*L/D*	0.54	1	0.54	1
Bearing Clearance, mm	C_b	0.0745	0.149	0.063	0.126
Preload	*m*	0.2857	0.375	0.2857	0.375
offset	γ	0.55	0.65	0.55	0.65

3.1.1 Results of Bearing Design Sensitivity Study:

The sensitivity analysis ranges for the selected design variables and the corresponding operating characteristics are shown in Table 1 and Table 2 for each bearing type. The value of m is taken as ranging from 0.25 to 0.6 [5]. The load and bearing temperature increase as the ellipticity ratio increases. However, it is observed here that the maximum averaged oil film temperature is 58.54°C which is lower than the acceptable limit of 65^0C. The effects of the two clearances in the elliptical bearing on its steady-state characteristics are shown in Fig. 6 a, b, c. Fig. 6 shows the results for a 149 mm diameter front bearing. The main controlling influence on the power loss is the bearing. When the clearance increases, it leads to a reduction in the power loss as shown in Fig. 6a. An increase in bearing clearance also tends to decrease maximum temperature, Fig. 6c. Minimum film thickness increases with increasing clearance (see Fig. 6b), the effect being a function of the actual values of journal load and operating speed. Therefore, an increase in the bearing clearance will also produce an increase in the maximum MOFT. This is due to the formation of a large supporting oil wedge which is mandatory for stable hydrodynamic lubrication [4]. Similarly, with minimum clearance there is more friction which results in more power loss. Whereas the maximum clearance has better results of reduction in power loss but it can cause instability if the clearances are too large.

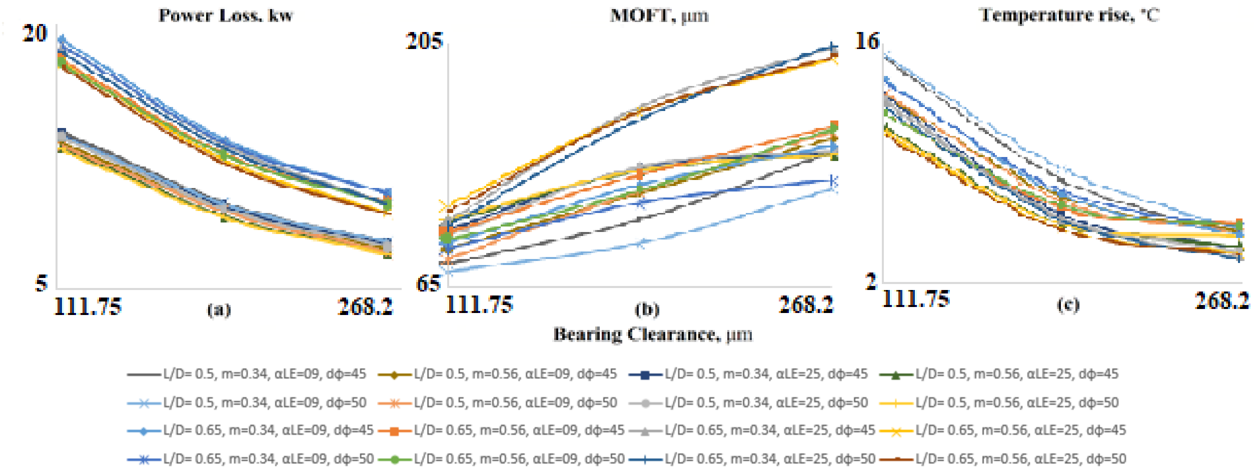


Fig. 6 Results of (a) Power loss [kw], (b) Minimum oil film thickness [μm], (c) Temperature rise [°C] vs bearing clearances for redesigned elliptical front bearing

The effects of the tilting pad bearing on its steady-state characteristics are shown in Fig. 7a, b, c. Results shown when the clearance increases, it leads to a reduction in the power loss. Decreasing the L/D ratios also leads to a reduction in the power loss (Fig. 7a). An increase in bearing clearance tends to decrease the maximum temperature and there is a minimum influence for different L/D ratios as shown in Fig. 7c. Minimum film thickness increases with increasing clearance (see Fig. 7b), the effects being a function of the actual values of journal load and operating speed. The redesigned elliptical front and rear bearings have diameters of 149 mm and 126 mm, with a preload of 0.56 and 0.34, length of 74.5 mm and 63 mm, respectively, and clearance of 0.19 mm and 0.2268 mm, respectively. The newly designed tilting pad front and rear bearings have diameters of 149 mm and 126 mm, with a preload of 0.375, length of 111.75 mm and 95 mm, respectively, and clearance of 0.149 mm and 0.126 mm, respectively.

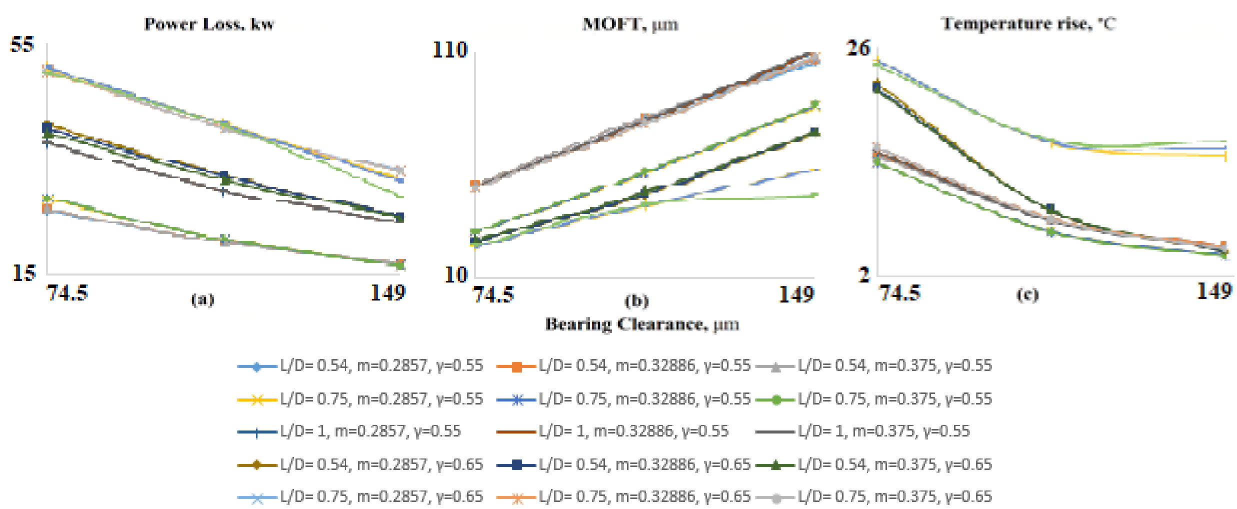


Fig. 7 Results of (a) Power loss [kw], (b) Minimum oil film thickness [μm], (c) Temperature rise [°C] vs bearing clearances for tilting pad front bearing

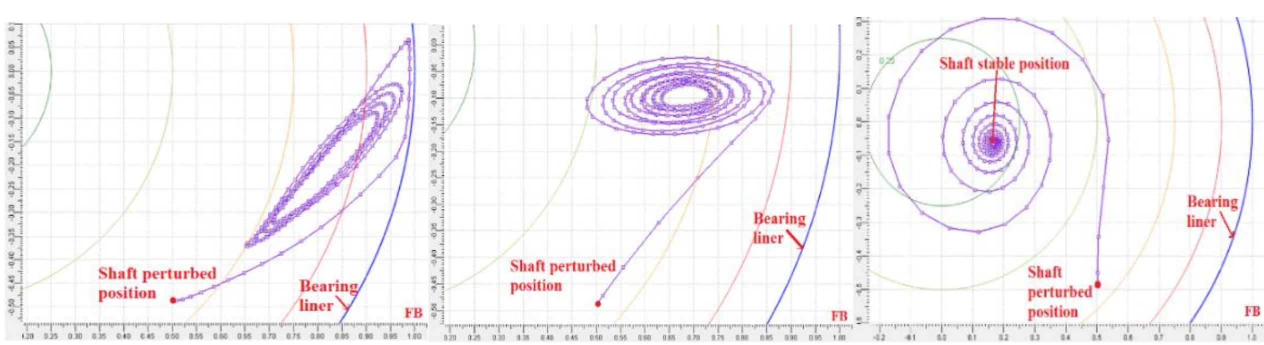
The redesign of elliptical bearing increases the front and rear bearing minimum oil film thickness (MOFT) by 45 percent and 34 percent, respectively. Power losses are reduced by 70 percent for both bearings. In addition, temperature rises are also reduced by 84 percent compared to the original elliptical bearings which is quite significant. The newly designed tilting pad bearing decreases the front bearing MOFT by 18 percent and increases the rear bearing MOFT by 5 percent. Despite decreasing the corresponding minimum oil film thickness value also there are enough load carrying capacity present in the present retrofitted system. Power losses are reduced by 24 and 28 percent at the front and rear bearings, respectively. In addition, temperature rises are significantly reduced: about 85 percent compared to the original elliptical bearings. The undamped critical speeds in the operating range are improved with these bearings. Some of the undamped critical speeds which exist in the original and redesigned bearings, due to their axisymmetric stiffness and damping characteristics. For the redesigned elliptical bearing case, the first undamped critical speed decreases from 2069 rpm to 1769 rpm and the second undamped critical speed decreases from 4167.6 rpm to 3580.4 rpm. These are suppressed by the new tilting pad design where the first mode frequency changes from 1769 rpm to 4154.6 rpm. The second and third mode frequencies move away from the 8856 rpm operating speed range (-15% to +5% of rated speed). Bearing performance characteristics, such as the maximum averaged oil film temperature across the film thickness T_{max}, Minimum oil film thickness H_{min}, Power loss N_{fr}, Friction coefficient F_r and Undamped critical speeds for the selected bearing model can be seen in Table 3.

Table 3 Bearing Sensitivity analysis results

		Initial Existing design		*Redesigned Elliptical design*		*Tilting pad Bearing*	
		FB	*RB*	*FB*	*RB*	*FB*	*RB*
MOFT	*Hmin (μm)*	90.76	72.24	131.79	96.92	74.30	75.72
Power Loss	*Nfr (kw)*	32.26	19.37	10.08	5.45	24.68	13.92
Friction loss	*Fr, -*	0.113	0.081	0.035	0.023	0.086	0.058
Temp Rise	*ΔT, °C*	27.93	28.82	4.84	4.16	4.56	4.01
1st UCS, X-axis	*rpm*	2069		1769		4154	
2nd UCS, X-axis	*rpm*	4167		3580		10468	
3rd UCS,_Y-axis	*rpm*	4306		3651		-	

The impact of the bearings on the vibrational behavior of the rotor system also becomes more important. For rotors operating below the first critical speed the bearing stiffness can play a large part in determining the vibration levels resulting from critical speed, balance, or aerodynamic cross-coupled forces.

Bearing stability analysis results are presented in Fig. 8a. It shows the results of shaft center motion trajectories as a result of perturbation in dimensionless coordinates for the initial bearing for an unbalanced rotor mass. At the rotating speed of N_{op}=8856 rpm, the shaft follows an elliptical trajectory within the limit cycle with a violent whirling motion making contact with the bearing surface, which may result in a failure of the rotor/bearing system. Based on the initial existing bearing instability and thermal conditions the redesign of bearing was carried out to get better performance and improve the bearing stability.



(a) (b) (c)

Fig. 8 Bearing stability for front bearings (a) Initial existing elliptical bearing, (b) Redesigned elliptical bearing, (c) Tilting pad bearing.

Fig. 8b shows the results of the redesigned elliptical bearing for an unbalanced rotor mass. At N_{op}=8856 rpm, the shaft follows an elliptical orbital path with a decreasing radius. The results show that the shaft features are acceptable and within stable limits. However, these conditions do not satisfy the API recommendations as discussed in the following section. This is why the new tilting pad design was carried out to meet these requirements. Finally, Fig. 8c shows the results of the new tilting pad bearing for an unbalanced rotor mass. At N_{op}=8856 rpm, the shaft follows a circular orbital path with a decreasing radius. The results show that the shaft is more stable than the previous bearing design. This condition satisfies the API requirements for the rotor/bearing system.

3.1.2 Synchronous Unbalance Response Analysis

The same unbalance criteria which were previously described in the original design configurations were used. The redesigned elliptical bearings improve the unbalance response amplitudes at both bearings and reduce the amplitudes at the shaft center. This amplitude reduction can reduce the chances of developing a hard rub which could set off the sensitive second mode as previously described in the stability analysis of the original rotor bearing system. In the redesigned elliptical bearings, increased amount of unbalance response (vibrations) did not cause any of the two bearings peak-to-peak (p-p) amplitudes to exceed 75% of the minimum design diametral running clearances throughout the machine. Unbalance results for the redesigned model are presented in Fig. 9a.

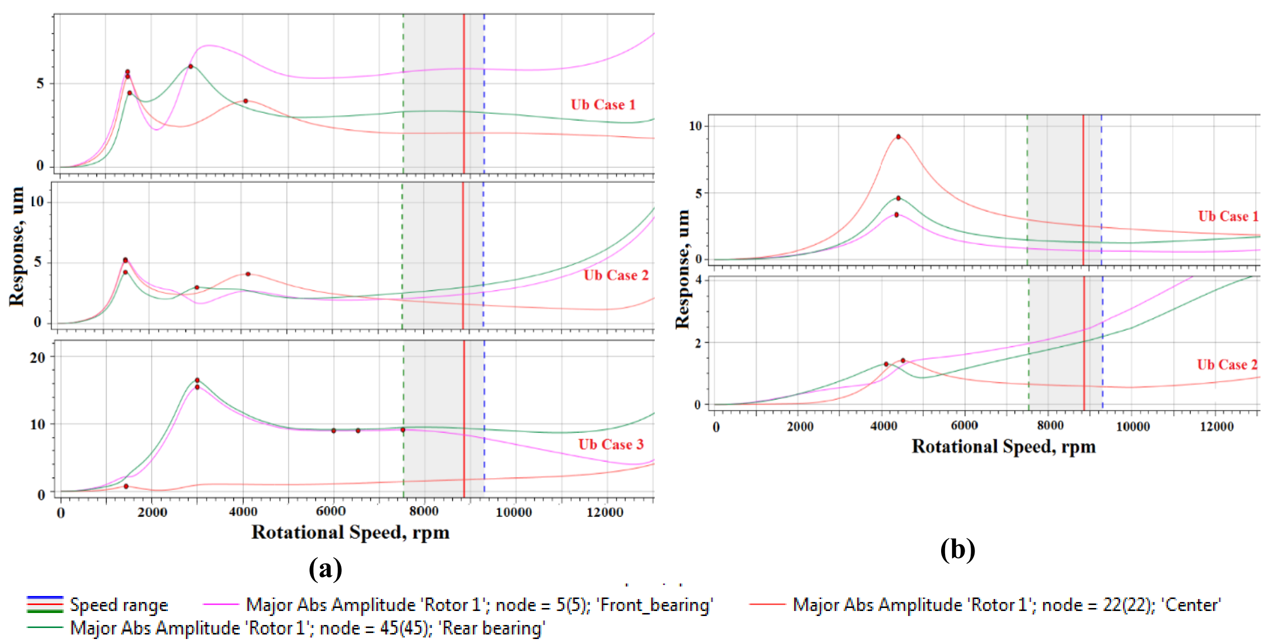


Fig. 9 Frequency response amplitude (peak-to-peak, um) for (a) Redesigned elliptical rotor/bearing system, (b) Tilting pad rotor/bearing system

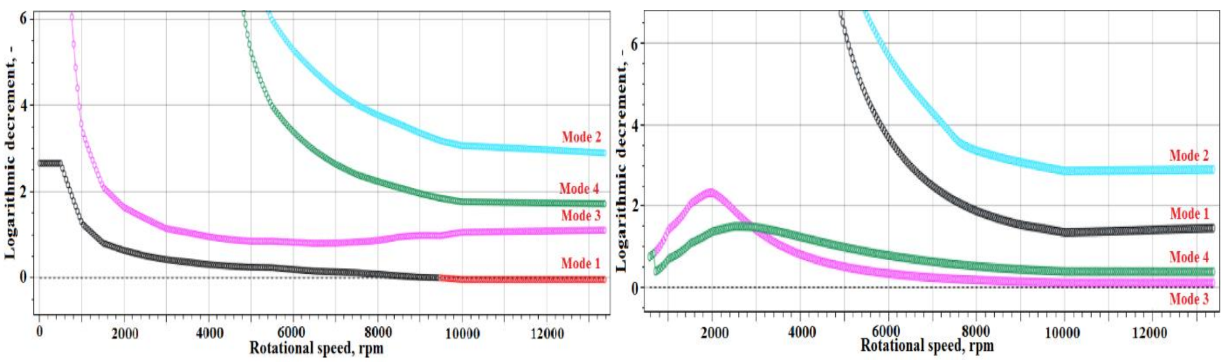
The new tilting pad rotor/bearing system follows the same unbalance criteria used previously. Fig. 2 describes the unbalance cases and their locations. The "worst case" unbalance condition consists of a 63 µm diametral running clearance. The new bearings improve the unbalance response amplitudes at both bearings and reduce them at the shaft center, as shown in Fig. 11. With the new tilting pad bearings, the increased amount of unbalance response did not cause either bearings peak-to-peak (p-p) amplitude to exceed 75% of the minimum design diametral running clearances throughout the machine.

The unbalance response amplitude results of the tilting pad rotor/bearing system are compared with the redesigned elliptical rotor/bearing system. The front bearing response amplitude is decreased by 2.6 percent. In addition, the amplitude response increases by 1.5 percent at the shaft center and reduces by 3.1 percent at the rear bearing location. This increase at the shaft center location represents a change from 0 to 10 µm p-p. This is acceptable considering that the new tilting pad bearing has a diametric clearance of 63µm. The new tilting pad model unbalance results are presented in Fig. 9b.

3.1.3 Stability analysis

The stability analysis was performed considering the destabilizing effects of the aerodynamic excitations, based on which modifications were applied to the design in order to stabilize the multi-stage centrifugal compressor and to conform to the API Level I stability criteria. The stability results are listed in Table 4 for the first four whirl modes. Particularly noted are the low values of the "log decrements" for the modes with frequencies near the targeted operating speed of 8856 rpm. Results reveal that the redesigned elliptical bearings of the second, third and fourth modes are very stable. The first mode is at 29.66 Hz with a log decrement of +0.02 is only marginally stable. The second mode is at 102.27 Hz with a log decrement of +0.97. In addition, the third mode is at 52.32 Hz with a log decrement of +1.99. Finally, the fourth mode is at 255.75 Hz with a log decrement of +3.43. In the redesigned elliptical bearings, the rotor stability condition is slightly improved compared to the original elliptical bearings. The second mode log decrement increases from -0.43 to +0.97. However, these redesigned log decrements are still lower than the desired 0.1 threshold.

In the new "tilting pad" bearings, the rotor stability condition is very much improved compared to the redesigned elliptical bearing configurations. The first mode log decrement increases from +0.02 to +1.58. Finally, these new tilting pad bearings log decrements satisfy the API recommendations. Comparison of log decrements for the redesigned elliptical bearing and new tilting pad bearing is shown in Fig. 10.



(a) **(b)**

Fig. 10 Logarithmic decrement for (a) Redesigned elliptical and (b) Tilting pad rotor/bearing system

Table 4 Logarithmic decrement comparison results

	Mode Number	*Damped Natural frequency, Hz*	*Log dec*	*Stability condition*
Initial existing bearing	1	87.22	1.88	Very Stable
	2	***61.34***	***-0.43***	***Unstable***
	3	90.83	0.68	Stable
	4	143.81	4.94	Very Stable
Redesign Elliptical Bearing	1	***29.66***	***0.02***	***Marginally Stable***
	2	102.27	0.97	Stable
	3	52.32	1.99	Very Stable
	4	255.75	3.43	Very Stable
Tilting pad Bearing	1	238.22	1.58	Very Stable
	2	438.96	3.11	Very Stable
	3	80.95	0.15	Stable
	4	81.72	0.45	Stable

4 Conclusions

The analysis results have indicated that the redesigned elliptical bearing improves the rotor stability condition, compared to that with the original elliptical bearings. However, the redesigned log decrements are still lower than desired. It is predicted that these improvements would significantly diminish the chances of experiencing a rub due to excessive vibration amplitudes. However, the redesigned bearings do not satisfy the API recommendations. The potential for this problem to occur could be significantly reduced by increasing the log decrement and shifting the whirl frequency further away from the operating speed. In order to meet these goals, the design of new "tilting pad" bearings were carried out. The analysis has shown that the solution of turbomachinery problems can be addressed from different levels. Specifically, the new tilting pad bearings significantly improve the rotor/bearing system unbalance response amplitudes, compared to the response with the original bearings and the redesigned elliptical bearings. This amplitude reduction can reduce the chances of a hard rub, as discussed in earlier stages of this study. The worst case unbalance condition considers a 63 μm diametrical clearance. With the redesigned bearings, this amount of unbalance did not cause the front bearing peak-to-peak (p-p) amplitude to exceed its 75% diametric clearance safety limit. It is predicted that this improvement would significantly diminish the chances of experiencing a rub due to excessive vibration amplitude. These modifications also significantly eliminated the instability that was initially observed and the redesigned compressor met not only the change in operating conditions but also fully complied with API 617/684 standards.

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