Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-45311

IMPROVING ROTORDYNAMIC PERFORMANCE OF AN AXIAL FLOW COMPRESSOR WITH TWO-LOBE BEARINGS USING NONLINEAR CONSTRAINT PARAMETER OPTIMIZATION

Henning Ressing
Engineering Compressors
MAN Diesel & Turbo SE
Oberhausen, Germany

Sebastian Kukla
Chair of Engineering Design
Ruhr-University Bochum
Bochum, Germany

ABSTRACT

Bearings are a key factor in achieving a good rotor dynamics performance for turbo machinery. Large compressors, steam and gas turbines for industrial applications are generally equipped with journal bearings either as tilting pad or multi-lobe bearing type. Here bearing parameters such as bearing geometry, bearing load or oil viscosity significantly alter bearing behavior and influence the rotor dynamics of the entire rotor-bearing system.

In order to find an optimal set of bearing parameters for a given rotor-bearing system a nonlinear parameter optimization approach is employed. The rotor-bearing system is parameterized using bearing width, clearance and preload as design variables, since they represent design parameters that can be modified without significantly influencing the rotor design as a whole. The set of design variables is further constraint to stay within feasible limits of bearing design. The objective function is defined as a quantitative measure of rotor dynamic performance evaluating the distance from required separation margins with respect to rotor critical speeds based on API 617 7th Ed. In order to compute the objective function based on the design variables the bearing code ALP3T, solving Reynolds equations for the bearing fluid film, is used to compute the required stiffness and damping coefficients as input to the rotor dynamics program. The rotor dynamics performance is then evaluated using the rotor dynamics code SR3 based on the transfer matrix method. Both programs have been developed by the University of Braunschweig and are de-facto industry standard within the German turbo machinery industry. The two programs are coupled and the nonlinear constraint optimization problem is solved using MATLAB’s optimization toolbox.

The feasibility of this method is discussed based on an example of an axial flow compressor using two-lobe bearings. It is shown that a significant improvement in rotor dynamic performance can be achieved when compared to previous bearing selections for similar compressor designs and that the approach is suitable for a real-life engineering environment.

NOMENCLATURE

AF	amplification factor
B/D	bearing width to diameter ratio
$\mathbf{C}$	bearing stiffness matrix
c	bearing stiffness coefficient
$\mathbf{D}$	bearing damping matrix
d	bearing damping coefficient
m	bearing preload
N	rotational speed
N_c	resonance frequency
N_{max}	maximum rotational speed
N_{min}	minimum rotational speed
N_{rated}	rated speed
R	bearing sleeve inner radius
r	shaft radius
SM	separation margin
s	distance of resonance point to API safety zone
u_x, u_y	rotor horizontal and vertical displacement
x	vector of design parameters
ΔR	bearing radial clearance vertical
ΔR_s	bearing radial clearance horizontal
ψ	bearing relative radial clearance
ϕ	objective function
ζ	modal damping ratio

1 INTRODUCTION

Journal bearings are widely used to support rotating machinery such as industrial gas turbines, steam turbines, centrifugal and axial flow compressors. Many of these machines are individually designed for their specific application. Late in the design process for such tailor-made machines much of the dynamics of the rotor-bearing system are already determined by earlier decisions. However, the bearing design is easily changed without significantly affecting other machine parts. Bearing design has an important impact on the system dynamics and thus offers an efficient way to optimize rotor dynamics performance late in the design process.

The influence of bearing parameters on rotor dynamics has been investigated extensively [1,2]. Some studies have tried to find an optimal set of parameters for journal bearings ignoring the influence of the rotor system [3-5]. However, as has been pointed out by many researchers [6-14], the system dynamics are influenced by both, bearing and rotor system, such that only the analysis of the combined rotor-bearing system can lead to a realistic system improvement. Optimization studies investigating the influence of bearing parameters on the rotor-bearing system have employed a variety of objective functions focusing on stability [6], unbalance response [7], rotor mass [8] or bearing power loss [9-11]. Some researchers have also used a multi-objective optimization approach combining rotor weight and resonance response [12] or rotor weight and bearing load [13]. Recent studies have also focused on introducing different optimization algorithms such as genetic [11,14] or particle swarm algorithms [9] rather than the classical direct search algorithms.

While most researchers have used simple rotor models with only few degrees of freedom, this study focuses on testing the practicality of the approach of automatic optimization of bearing parameters within a real-life engineering design process.

2 MODELING OF THE ROTOR-BEARING SYSTEM

Rotordynamics modeling was based on API 617 7th Ed. [15] which presents a de facto standard in the industrial turbomachinery industry. Dynamics analysis according to API 617 chapter 2.6 requires an undamped modal analysis in order to determine natural frequencies and mode shapes of the rotor-bearing system as well as a subsequent damped unbalance response analysis. Design criteria are the absence of lowly damped critical speeds, e.g. a natural frequency at resonance condition. According to API 617 a lowly damped critical speed is defined as having an amplification factor of 2.5 or greater. Depending on the damping ratio, a minimum separation margin of critical speed to speed range is required. While amplification factor and separation margin according to API 617 are derived from the damped unbalance response analysis, it is just as well possible to derive them directly from a damped modal analysis. Here, this method has been chosen for conducting the optimization problem, because it does not require any assumptions regarding relevant unbalance distributions to be applied to the rotor-bearing system.

API 617 requires further a stability analysis to ensure stable behavior of the rotor-bearing system, including the influence of fluid destabilizing forces. For practical purposes this analysis has been ignored for the case at hand, since it is MAN Diesel & Turbo experience that axial compressors featuring comparatively low final pressure levels do not show any unstable behavior caused by the process fluid.

2.1 ROTOR MODEL

The analysis is based on a 20 MW axial flow compressor shown in Figure 1. The compressor consists of sixteen axial stages, one radial end-stage, a balance piston and a flexible coupling at the discharge side end. For the rotor model only shaft mass and stiffness as well as additional lumped masses for impellers and couplings were considered. The journal bearings are two-lobe bearings with a diameter of 180 mm at the suction side and 200 mm at the discharge side. The support stiffness used was based on the API 617 recommendation of $8.75 \cdot 10^6$ N/mm in the horizontal and vertical direction. Damping is only considered for the oil film. The rotor model was solved using the rotor dynamics program SR3 [16] based on the transfer matrix method. SR3 has been developed by the University of Braunschweig and represents a de facto industry standard for rotor dynamics computation within Germany.

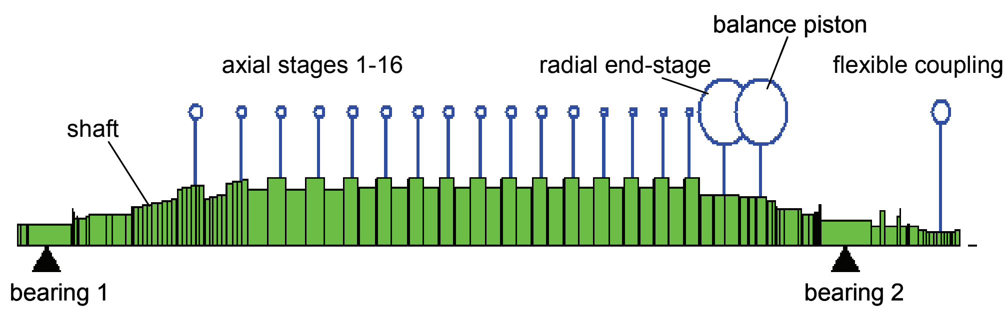

Figure 1. Rotor model showing shaft elements, additional masses for blades and impellers and bearing positions.

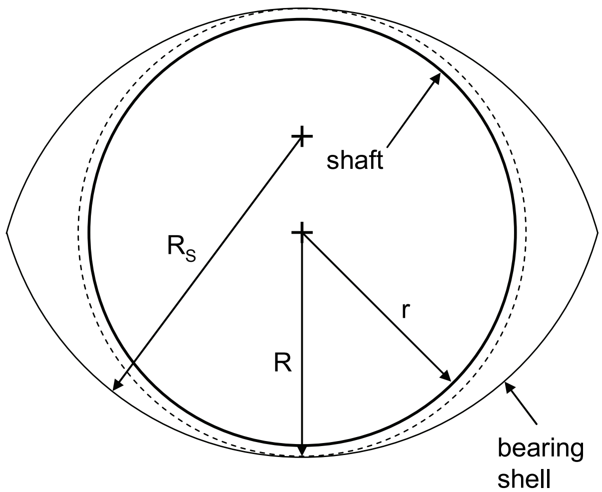

Figure 2. Geometry parameters of the two-lobe bearing model.

2.2 BEARING MODELLING

The two-lobe bearing model used is shown in Figure 2. Both bearings consist of two pads with oil pockets between them. Input parameters are the ratio of bearing width to diameter B/D, the bearing clearance ψ defined by

$$\psi = \frac{R - r}{R}, \tag{1}$$

and the preload m where

$$m = 1 - \frac{\Delta R}{\Delta R_s} = 1 - \frac{R - r}{R_s - r}. \tag{2}$$

Here, R is the bearing sleeve inner radius, r the shaft radius and R_s the pad curvature radius. All other bearing parameters remained constant. The bearings were modeled using ALP3T [17] in order to compute the bearing stiffness and damping coefficients matrices **C** and **D**, respectively, which are dependent of the shaft rotational speed N:

$$\mathbf{C} = \mathbf{C}(N) = \begin{bmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{bmatrix} \tag{3}$$

$$\mathbf{D} = \mathbf{D}(N) = \begin{bmatrix} d_{xx} & d_{xy} \\ d_{yx} & d_{yy} \end{bmatrix} \tag{3}$$

It must be noted that for two-lobe bearings the coupling coefficients of the stiffness matrix c_{xy} and c_{yx} as well as the coupling coefficients of the damping matrix d_{xy} and d_{yx} are neither zero – as in the case of tilting-pad bearings – nor are the matrices **C** and **D** symmetric [1].

Figure 3 shows the results of the damped modal analysis of the rotor-bearing system for the bearing parameter set [$\psi_1 = 1.5$, $m_1 = 0.65$, $(B/D)_1 = 0.75$, $\psi_2 = 1.5$, $m_2 = 0.65$, $(B/D)_2 = 0.75$]. A plot of the mode shapes shows the significant modes with respect to the rotors operating speed range being the classical cylindrical and conical modes. However, next to their rigid-body character they also exhibit a significant amount of shaft bending, particularly the second and fourth mode.

Due to the anisotropic character of the two-lobe bearing each mode shape exists twice – once in the dominantly horizontal and once in the dominantly vertical direction. Hence, modes 1 to 4 are often referred to as 1 horizontal (1h), 1 vertical (1v), 2h and 2v. The difference in bearing stiffness is so great that the distance between the resonance frequencies of modes 1v (41.8 Hz) and 2h (61.5 Hz) is smaller than the distance between 2h and 2v (119.9 Hz).

Figure 4 shows a root plot of the damped eigenvalues, where modal damping is plotted versus resonance frequencies of the rotor-bearing model. At the same time, the safety zone according to API 617 around the rotors rotational speed range is shown. Eigenvalues at resonance within the speed range must exhibit a modal damping ratio of at least 0.2 in order to not be considered critical speeds, as is the case for mode 2 h. To the left and to the right of the rotors minimum and maximum continuous speed the required damping ratio is being reduced. Here, the limits are defined by API 617 as

$$SM_{N_{\min}} = 17\left(1 - \frac{1}{AF - 1.5}\right), \tag{5}$$

$$SM_{N_{\max}} = 10 + 17\left(1 - \frac{1}{AF - 1.5}\right), \tag{6}$$

where AF is the amplification factor defined by

$$AF = \frac{N_c}{N_2 - N_1}. \tag{7}$$

Here, N_c is the resonance frequency and N_1 and N_2 are the frequencies at 0.707 · peak amplitude to the left and right of the resonance point in a bode diagram. However, for low levels of damping and within the linear region the amplification factor can be converted into the modal damping ratio by

$$\zeta = \frac{1}{2\,AF}. \tag{8}$$

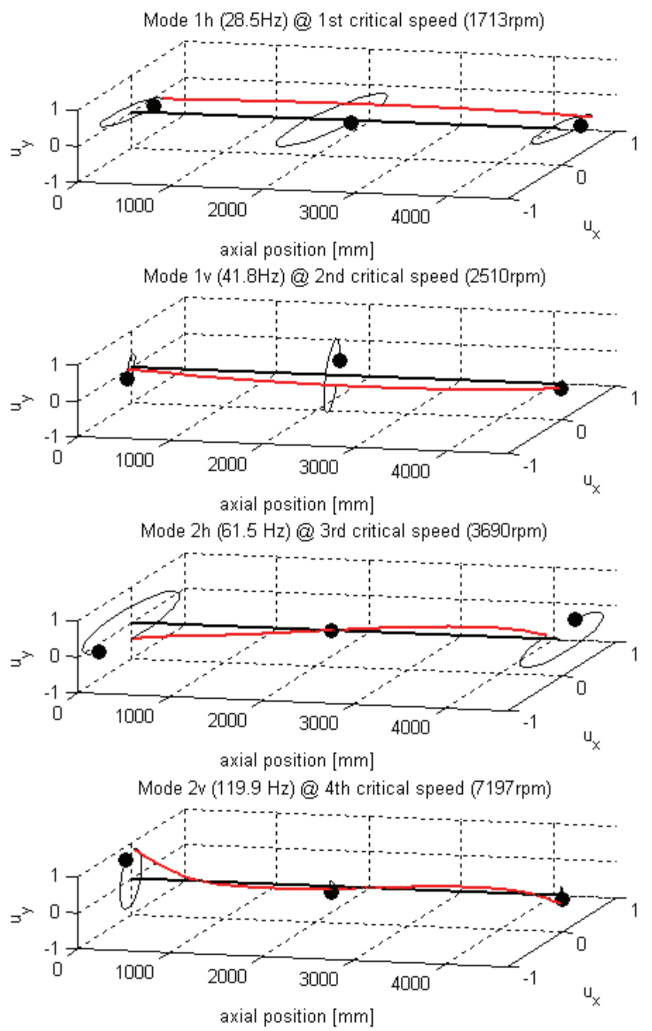

Figure 3. Rotor natural frequencies and 3D mode shapes at resonance speed.

Further, no resonance frequency is allowed to exhibit negative damping, since this would indicate an unstable behavior of the rotor-bearing system. Thus, in simple terms it can be stated that most rotor dynamics criteria of API 617 are fulfilled, if no resonance occurs within the API safety zone shown in Figure 4.

3 OPTIMIZATION APPROACH

In order to find an optimized set of bearing parameters from a rotor dynamics point of view, they should be chosen such that the rotor-bearing system exhibits a maximum distance between or – in order to state the problem as a minimization problem according to common convention – minimum closeness of its resonance points to the API safety zone of Figure 4, while keeping bearing parameters within feasible design limits. Thus, the problem can be stated as a constrained optimization problem

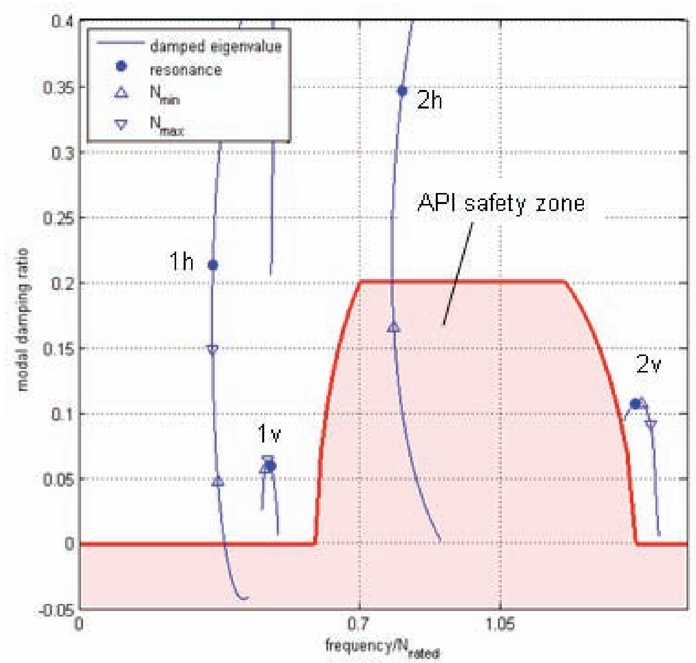

Figure 4. Damped eigenvalue plot of the rotor-bearing system exhibiting the API safety zone.

$$min\ \phi = f(x) \tag{9}$$

where x is the vector of design parameters

$$x = [\psi_1, m_1, (B/D)_1, \psi_2, m_2, (B/D)_2\ ] \tag{10}$$

subject to:
$$1.0 \leq \psi_{1,2} \leq 2.0$$
$$0.5 \leq m_{1,2} \leq 0.8 \tag{11}$$
$$0.5 \leq (B/D)_{1,2} \leq 1.0,$$

where the indices 1 and 2 denote the bearings.

The boundaries placed on the design variables represent design limits upon which the rotor dynamic model is valid based on experience. They also ensure that the bearing will not experience an excessive thermal load or mean bearing pressure.

Figure 5 shows a flow chart of the optimization procedure. A starting guess of the vector of design variables is combined with the set of bearing parameters which remain constant in order to create a bearing model in ALP3T. This model is solved and the bearing stiffness and damping coefficients are obtained and used as input to the rotor dynamics model. From this the rotor dynamics solution is obtained and the relevant modes and resonance frequencies are evaluated. The rotor dynamics objective function ϕ is then defined as the minimum distance s between the resonance points of the damped eigenvalue plot to the API safety zone

$$\phi = min(s_i),\ \text{for } i = 1, \ldots, \text{number of resonance points.} \tag{12}$$

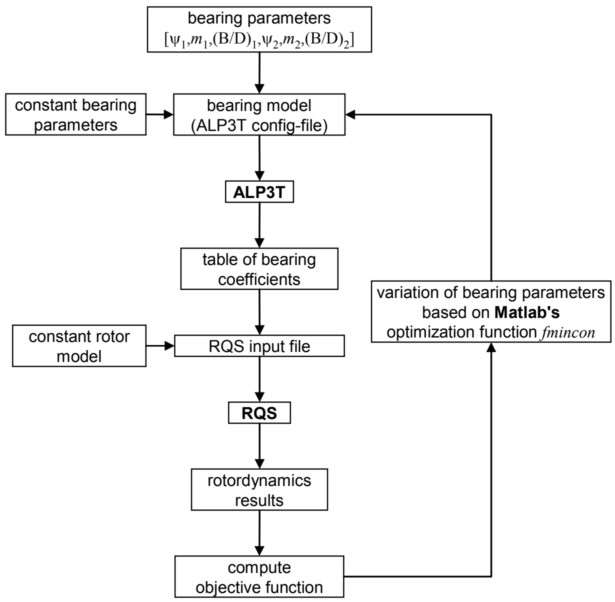

Figure 5. Optimization flow chart.

Here, s was evaluated numerically, by discretizing the API safety zone boundary into 30000 small segments.

Resonance points outside the safety zone are considered to have a negative distance, points within the safety zone are calculated to have a positive distance to the closest safety zone boundary.

The optimization loop is controlled by Matlab's *fmincon* function for constrained optimization problems using a gradient based algorithm [18]. The gradient is computed using a simple two point finite difference approach. For numerical reasons the objective function was evaluated using normalized values of resonance frequency and modal damping. Convergence criteria were set to:

$$\begin{aligned} \Delta\psi_{1,2} &\leq 0.001 \\ \Delta m_{1,2} &\leq 0.001 \\ \Delta(B/D)_{1,2} &\leq 0.001 \\ \Delta\phi &\leq 0.0001 \end{aligned} \qquad (13)$$

where Δ marks the change between two consecutive optimization iterations.

4 RESULTS

Initially a parameter study was conducted investigating the influence of input parameters ψ, m and B/D on the rotor dynamic performance. For this the bearing parameters for bearing 1 and 2 were set to be identical thus reducing the number of design variables to three. Results for various values of B/D are shown in Figure 6. It can be seen that all design variables have a significant influence on rotor dynamic performance. Best rotordynamic performance is achieved using the minimum preload. However, the best choice of bearing clearance varies with the ratio of width to diameter. For low B/D a small clearance seems beneficial, for high B/D a higher clearance achieves better results.

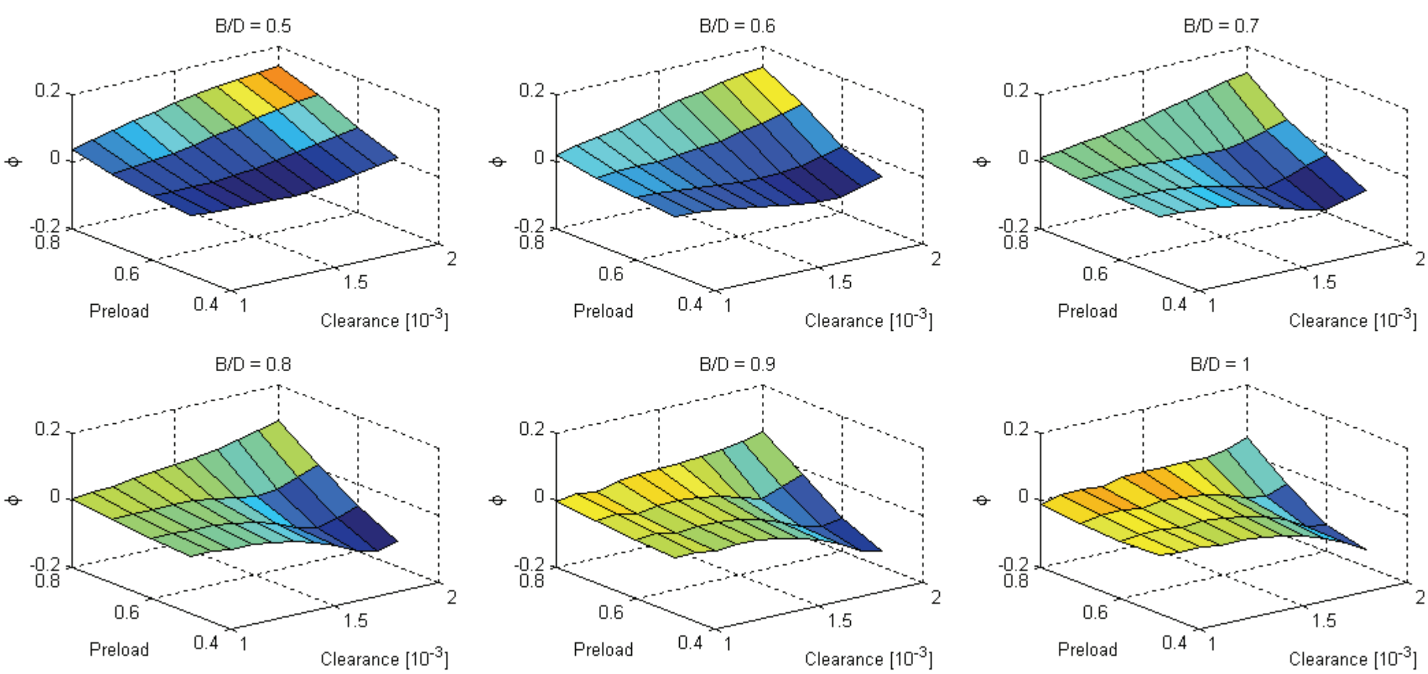

Figure 6. Surface plot of the objective function under the assumption of identical bearing parameters for various values of *B/D.*

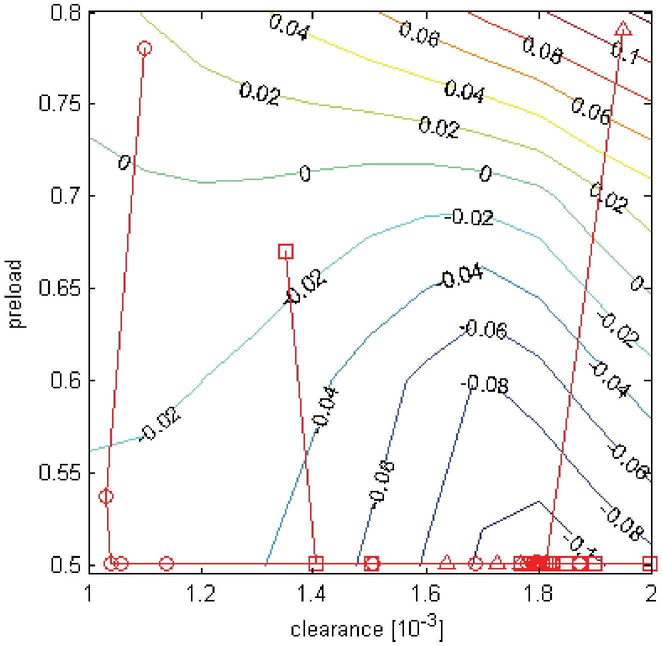

Figure 7. Convergence path for three different starting points.

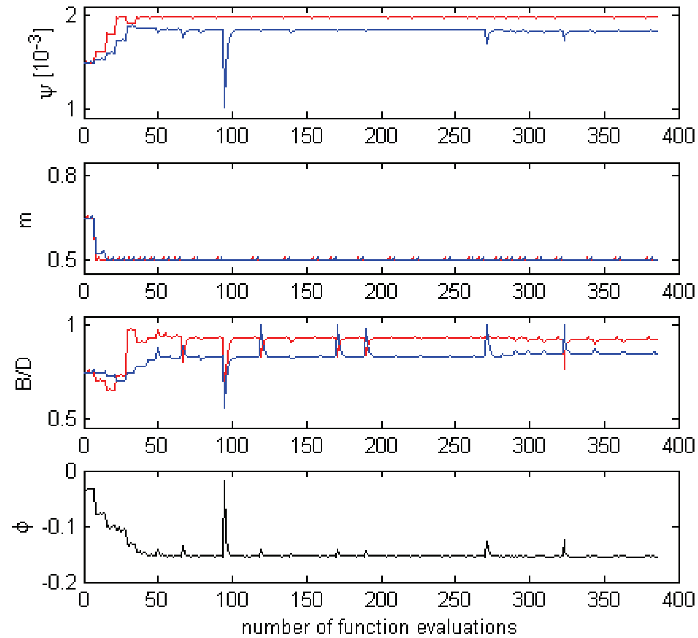

Figure 8. Convergence history for the full optimization using all six degrees of freedom.

The optimization routine employed was verified based on the results of the parameter study. Figure 7 shows the search path of the optimization algorithm for a constant value of $B/D = 0.7$. Convergence is achieved within 20 function evaluations on average. Further the global optimum is found from various starting points proving the suitability of the chosen optimization algorithm for this particular case. Similar results can be achieved, if B/D for both bearings is not kept constant. The global optimum is found repeatedly at x = [2.0 0.5 0.94] for various initial starting points. Convergence is achieved using 30 function evaluations on average.

Based on the results of the preliminary study the optimization run with the full set of six independent parameters was started. Figure 8 shows the history of the design parameters and the development of the objective function value. Convergence was achieved after 387 function evaluations. The found optimum point x_{opt} = [1.99 0.50 0.93 1.83 0.50 0.86] was the same for various starting points. Further, a sensitivity study, listed in Table 1, was conducted around the found optimum yielding no further improvement to the optimization problem. Therefore, and based on the results of the parameter study, it must be assumed that the found optimum represents the global optimum of rotordynamic performance. The result achieves a significant improvement over the original design point as is shown in Figure 9. The minimum distance of a resonance point to the API safety zone (mode 2v) has been increased by 300 %. In fact, the damping of mode 2v for the optimized design lies above 0.2 such that it would not even be considered as critical speed according to API 617. This only leaves mode 1v as a critical speed to be considered where the optimized design also leads to a significant increase in damping and thus a much improved characteristic with respect to rotor instability as well as run-up and shut-down behavior.

Table 1. Objective function gradient for a change in design parameters.

Parameter variation	+0.05	-0.05
ψ_1	N/A	0.142
m_1	0.106	N/A
$(B/D)_1$	0.103	0.039
ψ_2	0.018	0.045
m_2	0.027	N/A
$(B/D)_2$	0.034	0.004

5 CONCLUSIONS

The results show that nonlinear parameter optimization can be used to select an improved set of bearing parameters in order to optimize rotordynamic performance at a late point in the design phase. The found optimum showed a signifcant improvement in rotor dynamic performance increasing the minimum distance of the resonance point to the API safety

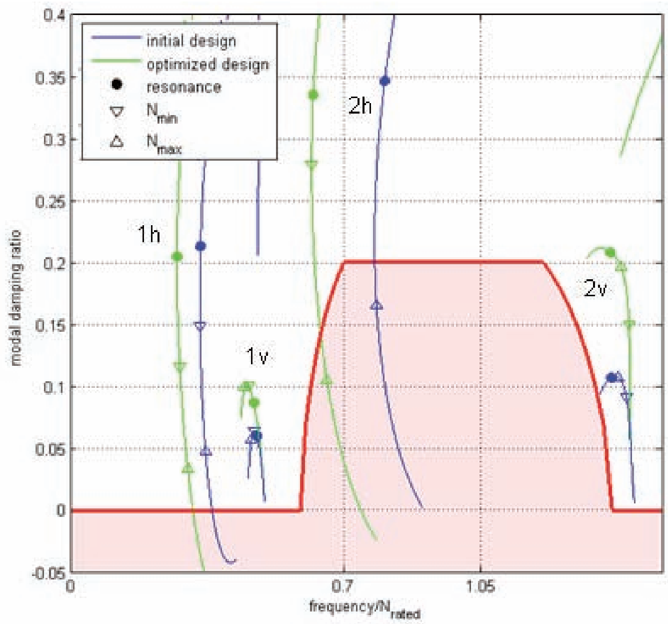

Figure 9. Comparison of rotor dynamic performance for the original and optimized rotor-bearing design.

zone by more than 300 % compared to the original design. Further, convergence to the global optimum was achieved from various starting points, proving the suitibility of this method. The number of function evaluations necessary to achieve the final result is quite high. However, a major portion of the optimization improvements is already achieved after less than 50 function evaluations, such that these benefits can be realized in less than a day making the approach suitable for a real-world engineering environment.

The results from the bearing parameter optimization were implemented on the described rotor and the axial compressor performed well during its mechanical running test.

REFERENCES

[1] Childs, D., *Turbomachinery Rotordynamics*, Wiley-Interscience, New York, 1993.

[2] Gasch, R., Nordmann, R., and Pfützner, H., *Rotordynamik,* Springer-Verlag, Berlin, 2006.

[3] Boedo, S. and Eshkabilov, S.L., "Optimal Shape Design of Steadily Loaded Journal Bearings Using Genetic Algorithms," *Tribology Transactions*, **46**, pp. 134-143.

[4] Wang, N.Z., Ho, C.L., and Cha, K.C., "Engineering Optimum Design of Fluid Film Lubricated Bearings," *Tribology Transactions*, **43**, pp. 377-386.

[5] Hashimoto, H., "Optimum Design of High-Speed, Short Journal Bearings by Mathematical Programming," Tribology Transactions, **40**, pp. 283-293.

[6] Matsuda, K., Kijimoto, S., and Kanemitsu Y., "Stability-Optimized Clearance Configuration of Fluid-Film Bearings," *ASME Journal of Tribology*, **129**, pp. 106-111, 2007.

[7] Cole, M.O.T., Wongratanaphisan, T., Keogh, P.S., "On LMI-based optimization of vibration and stability in rotor system design," *ASME Journal of Engineering for Gas Turbines and Power*, **128**, pp. 677-684, 2006.

[8] Chen, T.-Y., Wang, B.P., "Optimum Design of Rotor-Bearing Systems With Eigenvalue Constraints," *ASME of Engineering for Gas Turbines and Power*, **115**, pp. 256-260, 1993.

[9] Untaroiu, C.D., Untaroiu, A., Younan, A., and Allaire, P.E., Automatic Design Optimization of Rotors Supported on Tilting Pad Bearings," GT2009-59159, *Proceedings of ASME Turbo Expo 2009*, Orlando, USA, 2009.

[10] Angantyr, A. and Aidanpää, J.-O., "Optimization of a Rotor-Bearing System with an Evolutionary Algorithm," ISROMAC10-2004-060, The 10th International Symposium on Transport Phenomena and Dynamics of Rotating Machinery, Honolulu, USA, 2004.

[11] Angantyr, A. and Aidanpää, J.-O., "Constraint Optimization of Gas Turbine Tiltin Pad Bearing Designs," *Journal of Engineering for Gas Turbine and Power*, **128**, pp. 873-878, 2006.

[12] Choi, B.K. and Yang, B.S., "Optimum Design for Rotor-Bearing Systems Using Advanced Genetic Algorithms," *ASME Journal of Vibrations and Acoustics*, **123**, pp. 298-401, 2001.

[13] Choi, B.K. and Yang, B.S., "Optimum Shape Design of Rotor Shafts Using Genetic Algorithm," *Journal of Vibration and Control*, **6**, pp. 207-222, 2000.

[14] Shiau, T.-N., Kang, C.-H., and Liu, D.-S., "Interval Optimization of rotor-bearing systems with dynamic behaviour constraints using interval genetic algorithms," *Structural Multidisciplinary Optimization*, **36**, pp. 623-631, 2008.

[15] API Standard 617 7th Ed., *Axial and Centrifugal Compressors and Expander-compressors for Petroleum, Chemical and Gas Industry Services,* American Petroleum Institute, 2002.

[16] Forschungsvereinigung Antriebstechnik, "Programm SR3 zur Berechnung der selbst- und unwuchterregten Querschwingungen gleitgelagerter (und wälzgelagerter) Rotor mit Zusatzeinflüssen," *FVA-Heft 273*, 1990.

[17] Forschungsvereinigung Verbrennungskraftmschinen, "Robuste Lagerungen – Programmdokumentation zu ALP3T, Version 4.2," *Vorhaben Nr. 662*, 2004.

[18] Mathworks, *MATLAB 2007b*, 2007.