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DESIGN OPTIMIZATION OF ROTOR-BEARING SYSTEMS

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ABSTRACT

This paper presents a brief study of the information from the published literature and author's works regarding rotor-bearing systems analysis with respect to optimization. The main goal of this work is to motivate and give an idea to designers who are willing to deal with optimization of rotor-bearing sytems. The results obtained and presented in this study are to provide a comparison with numerical optimum design methods such as gradient-based method, and to show the potential of genetic algorithms in optimization of rotor-bearing systems. Genetic algorithms have been used as optimization problem solving techniques. They are parameter search procedures based on the idea of natural selection and genetics. These robust methods have increasingly recognized and applied in many applications.

Key Words : Rotor, Bearing, Optimization, Genetic algorithms

ROTOR-RULMAN SİSTEMLERİNİN TASARIM OPTİMİZASYONU

ÖZET

Bu makale rotor-rulman sistemlerinin analizi ve optimizasyonunu göz önüne alarak yazarın çalışmaları ile birlikte literatürden kısa bir bilgi sunumu mahiyetinde bir çalışmadır. Çalışmayla, rotor-rulman sistemlerinin optimizasyonu alanında çalışmak isteyen tasarımcıya bir motivasyon ve fikir verme amaç edinilmiştir. Çalışmada genetik algoritmalar methoduyla elde edilen sonuçlar numeric metodla elde edilen sonuçlarla kıyas edilerek genetik algoritmaların kabiliyeti gösterilmiştir. Genetik algoritmalar tabii seleksiyon (seçim) tekniğini kullanarak tanımlanan sınırlar içinde tarama yapan ve genetik fikrine dayalı uygun araştırma teknikleridirler. Günümüzde genetik algoritmalar bir çok alanda kullanılmaktadır.

Anahtar Kelimeler : Rotor, Rulman, Optimizasyon, Genetik algoritmalar

1. INTRODUCTION

The vibration of rotors and rotor systems has been a concern of engineers and scientists for a more than a century. In 1869, Rankine (1869) published an article, " On the Centrifugal Force of Rotating Shafts", which is the earliest reference to vibrations of a rotating system. From this period of time, dynamical analysis of shaft was begun. Modern designs of rotor-bearing systems usually aim for increased power output and improvement in efficiency. The demanding requirements placed on modern rotating machines, such as turbines, electric motors, electrical generators, compressors, have introduced a need for higher speeds and lower vibration levels. In addition to demand for improved aerodynamic performance, the mechanical components must satisfy requirements for prediction and control of rotor response, balancing, and rotor-bearing stability. The successful design of high-speed rotor-bearing system needs to be analyzed to evaluate the problems and to identify optimum solutions to the problems. This paper provides an introduction and a review of the role of rotor dynamics in design of high speed rotating equipment.

The role of rotor dynamics in accurately predicting the dynamic characteristics of rotor-bearing systems has become increasingly important, as the emphasis in design of modern rotating machinery has been towards higher speeds and higher outputs. To remain competitive, it is necessary that rotor-bearing system be designed to operate as efficiently as possible at higher speeds and power levels. Figure 1 shows a rotor-bearing-foundation system model. The designers of the rotating machinery systems typically perform calculations to examine the lateral bending characteristics of rotor systems. These classified as synchronous response, stability, critical speed, and transient analysis. Each of these involves different assumptions on the form of the system stimuli and response, (Rouch et al., 1991). The mathematical modeling of a rotor-bearing system requires consideration of structural dynamics and fluid dynamics to describe the rotor and bearing behavior. Each of these aspects can be considered in varying level of detail in the model.

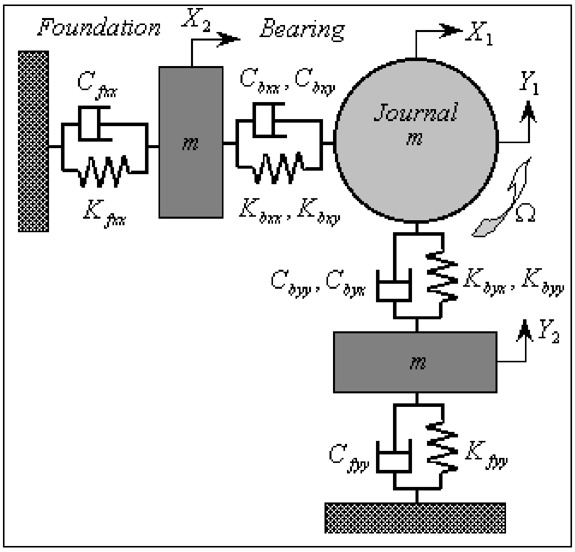

Figure 1. Rotor-bearing-foundation model

With this model, it is possible to optimize the design of the system. The optimization of designs by computer is an area in which research is rapidly growing. The conventional optimization techniques such as Non-linear Programming Method, Method of Feasible Directions, and Sequential Quadratic Programming Method have been used to conduct the optimum design of rotor-bearing system. These programming techniques provide a general approach for obtaining solution to both single and multi-objective design problems. These methods require at least the first-order derivatives of both the objective and constraint functions with respect to design variables.

Rajan et al. (1987) presented an automated design procedure for the optimal placement of undamped critical speeds of a rotor-bearing system. The desired design objective is treated as a non-linear programming problem that minimizes an objective function subject to constraints. The optimization program was designed to interact with the rotor dynamic analysis program to search for the feasible optimal design. An optimal design algorithm was developed by Shiau and Chang (1993) to minimize individually and simultaneously, the total weight of the shaft and the transmitted forces at the bearings, which play very important roles in a rotor-bearing system, under the constraints of critical speeds. The bearing stiffness, the cross-sectional area of the shaft, and the positions of bearings and disks were chosen as the design variables. The dynamic characteristics of rotor-bearing system also were determined by using the generalized polynomial expansion method. A notable work by Bhat et al. (1982) used an optimization technique to find the optimum dimensions of the plain cylindrical journal bearing and the viscosity of the oil to achieve minimum unbalance response. Barret et al. (1978) presented optimum support damping to minimize the unbalance response and maximize stability speed limit in the vicinity of the first critical speed of a rotating machinery. Chen (1987) and Chen et al. (1988) presented several optimal design procedures in the design of rotor-bearing systems using mathematical programming optimization techniques; Recursive Quadratic Programming and Feasible Direction Method. Roso (1997) applied the Method of Global Criterion to the optimization of rotor-bearing system design and found the method workable and predictable in its performance.

Recent advances in computer technology have allowed more complex systems to be optimized. Due to more efficient computers available today, a variety of new techniques and applications of optimization have been developed over the past years. Genetic algorithms are probably the best-known algorithms among those. Genetic algorithms are search procedures based on the mechanics of natural genetics, (Goldberg, 1989). The rapidly growing use of genetic algorithms in engineering problems is extended to include rotor-bearing system design optimization. The genetic algorithm is a non-traditional global search and optimization technique that provides attractive features for multi-objective engineering design optimization. Genetic algorithms have been shown to be capable of searching for optima in function spaces, which causes difficulties for gradient techniques (Saruhan et al., 2001; Saruhan et al., 2002). Roso (1997) developed and outlined the design criteria for analysis of the interaction between the rotor dynamic behavior and the performance of fluid-film bearings. Roso (1997) in his study concluded that the numerical optimization techniques involving multiple design variables may not always produce an absolute optimum solution, but rather a local optimum solution depending upon the mathematical characteristics of objective function. In this point of view, the author of this paper has provided use of Genetic Algorithms Method to handle these problems, (Saruhan, 2001). It is not the intention of this paper to provide broad information about genetic algorithms; interested reader may refer to (Goldberg, 1989; Mitchell, 1997) for in depth introduction of genetic algorithms.

2. ROTOR-BEARING SYSTEM

A typical rotating system is composed of various components, such as rotors, disks, support bearings and foundations. These massive and flexible components absorb and dissipate energy when subjected to disturbances, and produce a unique pattern of a variety of response, (Arora, 1987). The dynamic response of a rotor-bearing system can be approximated by the set of linear differential equations obtained from the finite element representation. The system parameters including the geometry of the system, coefficients of bearing, inertia properties of rigid disk, and the distribution of the mass and stiffness of rotating assemblies all of which have significant influence on the dynamic characteristics of the rotor-bearing system.

2. 1. Rotor

The representation of the shaft element as a series of beam elements is common in the application of the finite element method to rotor-bearing systems. Under a small deflection assumption, the linearized equation of motion for the shaft element can be used to model the shaft. For more detailed methodology, reader can refer a study by Rouch (1977). The general finite rotor element system of differential equation:

$$[M_t^r + M_r^r]\{\ddot{q}^r\} + \Omega[C_g^r]\{\dot{q}^r\} + [K^r]\{q^r\} = \{Q^r\} \qquad (1)$$

Where M_t^r, M_r^r, C_g^r, and K^r are the element translational mass matrix, the rotational inertia matrix, the gyroscopic matrix, and the element stiffness matrix respectively. Q^r is the vector of generalized forces. The displacement vectors q^r are the time dependent endpoint translations and rotations displacement of the finite element, which is shown in Figure 2.

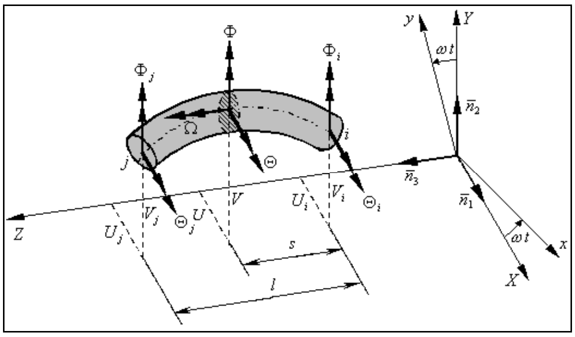

Figure 2. Typical finite rotor element and coordinate system

2. 2. Mass Elements

In many applications it is acceptable to consider the width of the disk to be negligible in comparison with overall rotor-bearing system, (Enrich, 1992). The disks are assumed to be concentrated, rigid body. Rigid disks are generally considered to be symmetric about the axis of rotation. The effect of the forces and moments applied by the disks on the shaft is added to the global dynamic stiffness matrix equation. The rigid disks are modeled as a four-degree of freedom rigid body with generalized coordinates defined as two translational (U,V) of the mass center in the (X,Y) directions and two rotations (Θ,Φ) of the plane of the disk about the (X,Y) axes. The rigid disk is usually required to be located at a node of the rotor element.

$$[M_t^d + M_r^d]\{\ddot{q}^d\} + \grave{U}[C_g^d]\{\dot{q}^d\} + [0]\{q^d\} = \{Q^d\} \qquad (2)$$

Where, $q^d = (U,V,\grave{E},\ddot{O})^T$ is the displacement vector of the finite rotor element station at which the disk is located and Q^d is the vector of generalized forces of the disk. M_t^d and M_r^t are translational mass matrix and rotational inertia matrices, respectively. The gyroscopic matrix, C_g^d, is a skew-symmetric matrix.

$$[M_t^d + M_r^d] = \begin{bmatrix} M & 0 & 0 & 0 \\ 0 & M & 0 & 0 \\ 0 & 0 & I_t & 0 \\ 0 & 0 & 0 & I_t \end{bmatrix}^{d} \qquad (3)$$

$$[C_g^d] = Ù\, I_p \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & -1 & 0 \end{bmatrix} = Ù \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & I_p \\ 0 & 0 & I_p & 0 \end{bmatrix} \qquad (4)$$

Assembly of the matrices are:

$$\begin{bmatrix} M & 0 & 0 & 0 \\ 0 & M & 0 & 0 \\ 0 & 0 & I_t & 0 \\ 0 & 0 & 0 & I_t \end{bmatrix}^d \begin{Bmatrix} \ddot{U} \\ \ddot{V} \\ \ddot{È} \\ \ddot{Ö} \end{Bmatrix}^d + Ù\, I_p \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & -1 & 0 \end{bmatrix}^d \begin{Bmatrix} \dot{U} \\ \dot{V} \\ \dot{È} \\ \dot{Ö} \end{Bmatrix}^d + \qquad (5)$$

$$\begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} U \\ V \\ È \\ Ö \end{Bmatrix}^d = Q^d$$

2. 3. Linear Bearing

In the rotor-bearing system, fluid-film bearings are frequently used because of their low wear and damping properties. Techniques for modeling bearings can be classified as linear or non-linear; linearized approaches usually require restrictions on the range of applicability, (Rouch, 1977). When load pushing down on the journal, it occupies an eccentric position. The displacement of loaded journal is not in the same direction of the acting load. Therefore, the journal reaction force exhibits both vertical and horizontal components. Thus, the fluid-film acts like a non-isotropic spring and dashpot with cross coupling between vertical and horizontal directions as shown in Figure 3. The most commonly used model for small perturbations of the journal from the static equilibrium position is that of eight anisotropic linear coefficients; each anisotropic bearing is described by four stiffness and four damping coefficients. The linear part of the spring and damping forces of such a bearing has the following form:

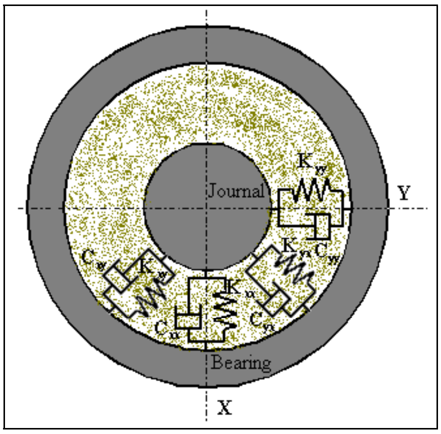

Figure 3. Modeling of fluid-film bearing

$$[C^b]\{\dot{q}^b\} + [K^b]\,[q^b] = \{Q^b\} \qquad (6)$$

or

$$\begin{bmatrix} C_{UU} & C_{UV} \\ C_{UV} & C_{VV} \end{bmatrix}^b \begin{Bmatrix} \dot{U} \\ \dot{V} \end{Bmatrix}^b + \begin{bmatrix} K_{UU} & K_{UV} \\ K_{UV} & K_{VV} \end{bmatrix}^b \begin{Bmatrix} U \\ V \end{Bmatrix}^b = \{Q^b\} \qquad (7)$$

Where $[C^b]$ is the damping matrix, $[K^b]$ is the bearing stiffness matrix, and $\{Q^b\}$ is the vector of generalized forces of the bearings. Fluid inertia effects are assumed to be negligible and translational and rotational motions are also decoupled.

2. 4. Foundation

The structure beyond the bearings, which is called the foundation, can have a significant effect on the response of the rotor-bearing system. A foundation is generally represented by stiffness and damping coefficients. The influences of foundation stiffness and damping on rotor-bearing system vibration characteristics cannot be ignored in most cases. The addition of foundation flexibility to the rotor-bearing system computation tends to lower the frequency of the first critical speed and increase the amplification factor, (Rouch et al., 1991). The system of governing equation of motions of each foundation level connecting node i to node j can be written as:

$$[M^f]\{\ddot{q}^f\} + [C^f]\{\dot{q}^f\} + [K^f]\{q^f\} = \{Q^f\} \qquad (8)$$

or

$$\begin{bmatrix} M_i^U & 0 & 0 & 0 \\ 0 & M_i^V & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}^f \begin{Bmatrix} \ddot{U}_i \\ \ddot{V}_i \\ \ddot{U}_j \\ \ddot{V}_j \end{Bmatrix}^f + \begin{bmatrix} C_{ij}^U & 0 & -C_{ij}^U & 0 \\ -C_{ij}^U & 0 & C_{ij}^U & 0 \\ 0 & C_{ij}^V & 0 & -C_{ij}^V \\ 0 & -C_{ij}^V\,0 & 0 & C_{ij}^V \end{bmatrix}^f \begin{Bmatrix} \dot{U}_i \\ \dot{V}_i \\ \dot{U}_j \\ \dot{V}_j \end{Bmatrix}^f$$

$$+ \begin{bmatrix} K_{ij}^U & 0 & -K_{ij}^U & 0 \\ -K_{ij}^U & 0 & K_{ij}^U & 0 \\ 0 & K_{ij}^V & 0 & -K_{ij}^V \\ 0 & -K_{ij}^V\,0 & 0 & K_{ij}^V \end{bmatrix}^f \begin{Bmatrix} U_i \\ V_i \\ U_j \\ V_j \end{Bmatrix}^f = Q^f \qquad (9)$$

The general dynamic equations of motion for the entire rotor-bearing-foundation system are obtained by assembling the appropriate components. Equation (10) gives the assembled equations of motion for the complete system. This equation describes the general motion of the entire physical system.

$$[M]\{\ddot{q}\} + ([C] + Ù\,[C_g])\{\dot{q}\} + [K]\{q\} = \{Q\} \qquad (10)$$

For fixed frame coordinates, where $[M]$ is generally known as the mass/inertia matrix which is a positive definite real symmetric matrix. $[C]$ is the damping matrix. It is a real non-symmetric and sparsely populated. $[C_g]$ is called the gyroscopic matrix, which is real skew-symmetric matrix. $[K]$ is a real non-symmetric due to non-conservative bearing properties and foundations. $\{q\}$ is the system displacement vector in a fixed reference frame. $\{Q\}$ is the vector of generalized forces.

3. ROTOR-BEARING SYSTEM ANALYSIS TYPES

Modern high-speed rotor-bearing systems are complex. With increasing performance criteria, the design process of these systems usually requires the integration of the design and analysis. All rotor-bearing system is supported by one or more bearings, which play a vital in determining the behavior of the rotating system under action of both static and dynamic loads. The analyses that are typically performed on rotor bearing systems are critical speed, synchronous response, stability, and transient analysis. In the following, these analyses will be introduced briefly.

3. 1. Critical Speed

A critical speed is defined as the frequency at which the rotational frequency of the shaft equals the vibration frequency. Critical speeds are dynamic properties of the rotor-bearing system. Each natural frequency of a rotor system has a particular mode shape. At a critical speed, the harmonic force from centrifugal unbalance excites the corresponding mode of the system, which causes the rotor to whirl in its supports in this mode shape, in synchronism with the rotor speed. A rotor system must be designed to operate without excessive vibration throughout its range of operating speed. Rotor unbalance, where the geometric center of rotation differs from the mass centroid, induces an unbalance force to cause synchronous vibration. Many machines operate above the first critical speed (first rotor system natural frequency). Therefore, they must be able to pass through one or more critical during start-up without excessive vibration, (Rouch et al., 1991). The primary source of system damping is usually the bearing. Sufficient bearing damping can allow passing through a critical speed or operation near a critical speed. It is also possible for rotors to be unstable due to poor bearing selection or due to aerodynamic effects, which can produce the equivalent of negative damping. Rotor systems must be designed so that instability does not occur in the operating range. The maximum operating speed is limited by the maximum speed at which the shaft, bearings, and components can be designed for safe operation. Damping is sometimes ignored in this analysis, providing an " undamped " critical speed for performing design purpose.

3. 2. Synchronous Response

Synchronous response involves solution for the motion of the rotor due to a specific unbalance force distribution at a given speed. A series of synchronous response solution across the operating speed range yields the location of damped critical speeds as peaks in the rotor response, (Rouch et al., 1991). The most simple rotor model is called a Jeffcott, (Jeffcott, 1919), rotor model, which is named after the dynamist who first used the rotor model to analyze the response of high-speed rotor system to rotor unbalance. The Jeffcotts's model of rotor is consists of the central mass carried on shaft supported by rigid bearings at each end as seen in Figure 4. Unbalance vibrations result from an unbalance force or unbalance moment. An unbalance force arises from eccentricity of the mass center as can be seen in Figure 4. Jeffcott introduced a damping force proportional to the velocity of the lateral motion. This made Jeffcott's model more realistic in the sense of rotor dynamic behavior. With this model, Jeffcott was able to explain the effect of unbalance when the rotating speed is near the natural frequency of the rotor. The differential equations of motion for Jeffcott's rotor model are:

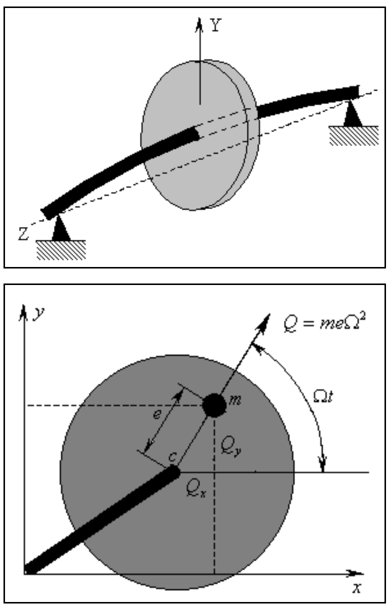

Figure 4. Jeffcott rotor on rigid bearing support and the mass eccentricity of rotor disk

$$m\ddot{x} + c\dot{x} + kx = Q_x \ ; \ Q_x = me\Omega^2 \ Cos\Omega t \qquad (11)$$

$$m\ddot{y} + c\dot{y} + ky = Q_y \ ; \ Q_y = me\Omega^2 \ Sin\Omega t \qquad (12)$$

Where m, c, and k are the rotor system modal mass, damping, and stiffness, respectively and Ω is rotating speed.

3. 3. Stability

Stability is related to the solution of the damped eigenvalue problem for the rotor system. The real part of eigenvalue is called the growth factor, and must be negative for the system to be stable. The imaginary part of eigenvalue is the damped critical speed. Stability analysis is necessary because of the effect of fluid forces in the rotor system. Incorrect bearing selection or presence of aerodynamic effects can produce the equivalent of negative damping, and give an unstable system. If a rotor is unstable at a given speed, any perturbation will cause the vibration amplitude to grow rather than decay. A typical stability analysis that includes fluid-film bearings and the destabilizing interaction with process fluid-flow forces is customarily summarized graphically by plotting the stability parameter (namely the growth factor or logarithmic decrement) versus an increasing value of the destabilizing parameter. Logarithmic decrement is defined as the natural logarithm of the ratio of any two successive amplitudes as shown in Figure 5.

$$\text{Logarithmic Decriment} = \delta = \ln \frac{A_1}{A_2} \qquad (13)$$

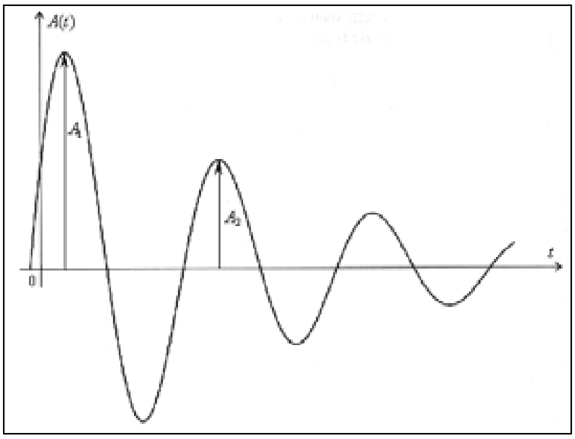

Figure 5. Free vibration of damped system.

3. 4. Transient Analysis

In most rotor-bearing system analyses, the system response is obtained in the form of analysis mentioned above. Sometimes, information of the instantaneous behavior of rotors is essential, especially for these regions close to critical speed and instability threshold speed. This often require that the time varying whirl orbits of such a rotor system can be calculated, (Subbiah and Rieger, 1988). Transient rotor dynamics solutions are more specialized analysis of time dependent or non-linear rotor-bearing system characteristics. Thus, this analysis calculates the forced response as a function of time. It also can represent the start-up and shutdown processes of rotating equipment, as well as the response to general time varying loading conditions (Buckles, 1995).

4. ROTOR-BEARING SYSTEM OPTIMIZATION

The use of genetic algorithms in the optimum design of a typical high-speed overhung centrifugal compressor, shown in Figure 6, is given as an example for design optimization of rotor-bearing system. The main focus of this example is to give a development of the bearing configurations that optimize stability along the other criteria such as frequency separation (placement of critical speeds). The system consists of a large disk (impeller) at the left-end side of the shaft, with the shaft supported in hydrodynamic fixed-lobe bearings at the station number I and II. The bearing optimized in this study is the one located at the station number I.

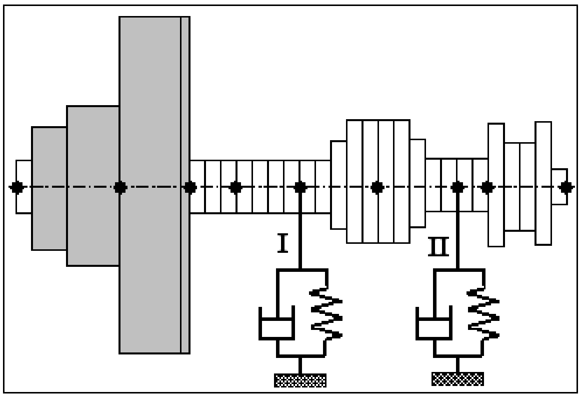

Figure 6. Finite element model configuration of the rotor-bearing system

Comparison of the best overall solution found by numerical optimization, (Roso, 1997), and that from the genetic algorithm technique, (Saruhan, 2001), is given in Table 1. As can be seen from these results, the genetic algorithm was able to obtain better results than those obtained by numerical optimization. Logarithmic decrement ended wıth

0.672 while numerical method with 0.666. This significant outcome satisfies the imposed specification and allows the rotor to maintain stability. Interested readers can refer to the study by Saruhan (2001) for more details.

Table 1. Comparison of the Best Overall Solution Found for Stability Objective and Design Criteria By Numerical and Genetic Algorithm Optimization

Objective Function		Numerical Optimization	Genetic Algorithm
Logarithmic Decrement		0.666	0.672
Frequency Separation (rpm)	1st Mode	20959	21696
	2nd Mode	80179	80174

5. CONCLUSIONS

With the demanding requirements for higher speeds and power output in rotating machinery, rotor-bearing systems optimization remains an active research area. Because of the complexity of rotor-bearing system analysis and time consuming nature of process an optimization procedure need to be employed that the design would be time-efficient and find the satisfactory design parameters to meet particular performans requirements. Many numerical optimization methods have been developed and used for design optimization of rotor-bearing systems. Most of these optimization methods make use of gradients to search feasible design parameters to achieve optimal objective functions. The development of faster computers has allowed implementation of more robust and efficient optimization methods. Genetic algorithms are one of these methods. They use objective function information instead of derivatives as in traditional methods. This work shows the efficacy of genetic algorithm optimization techniques and gives an idea to designers who are willing to deal with optimization of rotor-bearing sytems.

6. NOTATIONS

e	:	Radial location of unbalance mass (eccentricity)
k	:	Stiffness
m	:	Mass
Q	:	Generalized force
Q_x, Q_y	:	Force components
q	:	Generalized displacement vector
$\dot{q}$	:	Generalized velocity
$\ddot{q}$	:	Generalized acceleration
t	:	Time
U, V	:	Translational displacements in the X and Y-axis direction
Θ, Φ	:	Rotational displacement about X and Y-axis
Ω	:	Rotating speed
ω	:	Angular velocity vector about the principal axes
ω	:	Whirl speed
	:	

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