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Rotating Machinery Predictive Maintenance Through Expert System

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Modern rotating machines such as turbomachines, either produce or absorb huge amount of power. Some of the common applications are: steam turbine-generator and gas turbine-compressor-generator trains produce power and machines, such as pumps, centrifugal compressors, motors, generators, machine tool spindles, etc., are being used in industrial applications. Condition-based maintenance of rotating machinery is a common practice where the machine's condition is monitored constantly, so that timely maintenance can be done. Since modern machines are complex and the amount of data to be interpreted is huge, we need precise and fast methods in order to arrive at the best recommendations to prevent catastrophic failure and to prolong the life of the equipment. In the present work using vibration characteristics of a rotor-bearing system, the condition of a rotating machinery (electrical rotor) is predicted using an off-line expert system. The analysis of the problem is carried out in an Object Oriented Programming (OOP) framework using the finite element method. The expert system which is also developed in an OOP paradigm gives the type of the malfunctions, suggestions and recommendations. The system is implemented in C++.

Keywords: Condition monitoring, Vibration, Finite element method, Expert system, Object oriented programming, Rotating machinery

1. INTRODUCTION

Condition monitoring programs offer an innovative means for modern rotating equipment to implement and schedule predictive maintenance, as opposed to relying on conventional preventive maintenance techniques. Vibration, wear and temperature are the three important condition monitoring techniques to predict the health of the rotating machinery. Sohre (1972) has discussed the analysis and prediction of the reliability of turbomachinery installations and influence of various components on overall reliability. He has also presented, the prediction and identification of faults in the turbomachinery. Renwick (1984) has presented a condition monitoring program for plant maintenance and diagnosis of heavy machinery problems in which he discussed the importance of machinery dynamics in the design of a rotating machinery. Zimmer *et al.* (1986) have

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reviewed predictive maintenance programs for the rotating machinery using vibration signature analysis. Recently, the literature has given emphasis on expert systems for condition-based maintenance. An expert system is an Artificial Intelligence system created to solve problems in a particular domain. Hill *et al.* (1988) have presented the expert system for health monitoring of a rotating machinery. They discussed the expert system components for condition monitoring and design considerations. Lewis *et al.* (1989) have described a rule-based expert system for fault identification and predictive maintenance of turbomachinery using vibration oriented diagnosis. Sakthivel and Kalyanaraman (1993) have presented a knowledge based expert systems (KBES) approach to an engineering data. Rao (1993) discussed the various aspects of condition monitoring based on the vibration signature analysis and development of an off-line expert system for fault diagnosis in rotating machines. Sarath Kumar and Prabhu (1997) presented a rule-based off-line expert system for condition monitoring of a rotating machinery. Bettig and Han (1998) have discussed the usage of rotordynamic models in predictive maintenance, in which variables characterizing the state of deterioration mechanism are trended to determine the rate of deterioration to predict the machine life or the maintenance period (the time period between major over haul shutdowns).

In finite element modeling of rotor-bearing systems, Gmür and Rodrigues (1991) proposed a C^0-compatible linearly tapered shaft element to model the rotor system. This element consists of four degrees of freedom per nodal point (i.e., two lateral displacements and two total cross-sectional rotations), which includes the effects of translational inertia, rotary inertia, gyroscopic moments, internal viscous and hysteretic damping, shear deformation and mass eccentricity. Singh *et al.* (1977) have made parametric studies on the performance characteristics (load-carrying capacity, oil flow, attitude angle, stiffness coefficients) of a hydrostatic oil journal bearing using the finite element method. Birembaut and Peigney (1980) predicted the dynamic characteristics of rotor supported on hydrodynamic bearings using FEM and validated the theoretically obtained results with the experimental results. Lund (1987) has reviewed the concept of calculating the spring and damping coefficients of journal bearings.

When a rotor is operated beyond a certain rotational speed, a very high and unstable vibration component with a fractional frequency of the rotor speed will be developed. This rotor speed is referred to as the instability threshold. Lund (1965) presented a theoretical analysis for predicting stability of a symmetrical, flexible rotor supported in journal bearings. Rao (1983) and Muszynska (1988) presented instability criteria of the rotor supported on journal bearings. Majumdar *et al.* (1988) investigated the stability of a rigid rotor in oil journal bearings including the effect of elastic distortion in the bearing liner.

The method of implementation whereby programs are organized as a cooperative collections of objects, in which each object represents an instance of some class is called Object Oriented Programming (OOP). In rotor dynamic analysis a physical object is a constituent of the rotor, representing its geometrical shape, material, etc. and also the linkages with other rotating components. The main features of OOP are data encapsulation, operator overloading and inheritance. Each object encapsulates (hides) its data attributes from other objects and only displays its behavior, which enhances the portability of the application codes. Operator overloading refers to multiple usage of the same function name, with differing argument characteristics. A class may have several derived classes that may inherit some of its attributes and behavior from the parent class. This enables efficient and reusability of codes (e.g., coupling of the shaft or bearing may inherit characters of the class shaft). In OOP, both data and subroutines are linked intrinsically which leads to clear thinking of program design, so that the errors are minimized. Allocation of less storage space, less computational time, allowing programs to be substantially altered, without the need to change existing code, are the advantages of the OOP compared to FORTRAN language. Forde *et al.*

(1990) have discussed the implementation of an object-oriented program for the numerical analysis of two-dimensional solid and structural mechanics, emphasizing on knowledge-based expert system. Zeglinski *et al.* (1994) have discussed the concepts of OOP for the finite element method using C++ language in a generalized matrix library and showed that the efficiency, flexibility and maintainability of the C++ program is superior to a comparable version written in a non-OOP language, such as FORTRAN. Bettig *et al.* (1997) described the concepts of OOP for predictive maintenance of a rotating machinery, in which trend variables (e.g. bearing rotordynamic coefficients) were trended to predict the machine life/maintenance period using a finite element model in a graphical framework.

An expert system for condition monitoring purposes consists of three key components. They are (i) knowledge base (ii) inference engine and (iii) data base. The architecture of the expert system is shown in Appendix A. The system comprises hierarchical levels of generic rules (surface knowledge) and generic analytical simulation models (deep models). For a rotating machinery some surface knowledge is required about vibration, bearings, lubricant, seals, coupling etc. coupled to a much deeper knowledge of vibration analysis (i.e., critical speeds, resonance, responses, stability, etc.), bearing analysis (static and dynamic characteristics), lubricant flow analysis, seal analysis, crack initiation/propagation etc. In order to predict the type of the malfunction exactly, the detailed models knowledge is used. In the present study the deep knowledge regarding vibration analysis and bearing analysis is used to predict the malfunctions in rotating machinery. Knowledge of the machine is stored in the form of IF-THEN rules. Therefore rule has the following form

$$
\begin{array}{ll}
\text{IF} & \langle \text{antecedent1} \rangle \\
\text{AND} & \langle \text{antecedent2} \rangle \\
\text{AND} & \langle \text{antecedent}n \rangle \\
\text{THEN} & \langle \text{consequent1} \rangle \\
\text{AND} & \langle \text{consequent2} \rangle \\
\text{AND} & \langle \text{consequent}m \rangle
\end{array}
$$

The antecedents 1 to n are known as premises, which are given facts, and the consequents 1 to m are known as conclusions, which are deduced facts. The inference engine analyses the base and actual data in the data base and detects the discrepancies by using rules. The data base stores base values (from data books, reference manuals, past experience of personnel and experts in particular field) and actual values (experimentally measured values, analytically calculated values etc.). The rule compiler converts the rules about a machine condition to a form expected by the inference engine. The report generator, creates a consolidated report about the health of the machine based on inferences. The system gives the type of malfunction, cause of the problem, recommendations and suggestions regarding the machine. Very little literature is available which relates to combining theoretical rotor dynamic analysis with expert system for predicting the malfunctions in a rotating machinery. In the present work a theoretical model to predict the dynamic characteristics of the rotor-bearing system using the finite element method and a rule-based off-line expert system, for fault diagnosis of a rotating machinery, are presented and discussed in an object oriented framework.

2. ANALYSIS

A true object oriented solution to an engineering problem arises from a thorough analysis of the problem domain and the development of an application that emulates the existence of objects. A rotor analysis using the finite element method consists of four different classes of objects. They are: shaft, material, matrices and eigenvalue. Class shaft consists of details of the number of elements, number of nodes, type of element, type of shape function, speed of rotor, number of point masses etc. Figure 1 corresponds to the C++ program in an OOP paradigm and it illustrates how the class shaft data has been accessed through member functions in the main program without passing the data directly through functions.

```
class shaft{//class declaration
protected:
int ne, nn, npt; //No. of elements, nodes, point
  masses
float len, dia, speed; //shaft length, diameter, speed
char eletyp[10], shafunc[10]; //element type, shape
  function
public:
void input1 ( ); //functions than can access the
  private or protected data
void printshaft ( );
};
//class method definitions follow
void shaft :: input1 ( ){
cout << "Enter ne:" << "npt:" << "speed:" << endl;
cin >> ne >> npt >> speed;
for (int i = 1; i <= ne; i++){
cout << "i:" << "len:" << "dia:" << endl;
cin >> i >> len >> dia;
}
cout << "Enter eletyp:" << "shafunc:" << endl;
cin >> eletyp >> shafunc;
nn = ne + 1; //In beam element}
void shaft :: printshaft ( ) {
cout << setw (3) << ne << setw (3) << nn << setw (3)
  << npt << endl;
cout << setw (5) << speed << setw (10) << eletyp <<
  setw (10) << shafunc << endl;
for (int i = 1; i <= ne; i++){
cout << setw (3) << i << setw (5) << len << setw (5)
  << dia << endl;}}
//MAIN PROGRAM
void main (void) {
shaft s1; //shaft object
s1.input1 ( ); //variables encapsulated within
  methods
s1.printshaft ( );
//remaining source code follows
..................}
```

FIGURE 1 Passing of data from class shaft to main program via member functions.

The program shown in Fig. 1 consists of class shaft, two member functions input1 ( ), printshaft ( ) and the main program. The class shaft consists of two access specifiers "protected" and "public". The data declared under protected were called as member data and the functions declared in public are called member functions or methods, because they are the members of the class. The data attributes declared under protected of the class shaft are encapsulated with in the object. The data declared under "protected" were only known to the class and its descendent classes in the hierarchy, whereas "public" data are known to all the users of the class. The two functions input1 and printshaft defined in the public of the class shaft is used to access the input from the class shaft and print the same. The data declared in "private" is known only to the class and not accessible to the other classes. Messages are the means of invoking the methods supported by the class of objects and they are the primary way in which objects communicate and interact with each other. Messages are generally accompanied by arguments. The scope resolution operator "::" is used to tie the function definition to the class shaft. In Fig. 1, input1 ( ) and printshaft ( ) are the two member functions tied up to the class shaft. Member function input1 ( ) is for giving the input details like number of elements, number of point masses, speed of rotation, length, diameter of the shaft element etc., in the finite element analysis of rotor. Member function printshaft ( ) to print the input details given in member function input1 ( ). The main program, consists of object shaft "s1". s1. input1 ( ) and s1.printshaft ( ) corresponds to class member access operator, which connects the object name and the member functions, so that, the member functions input1 and printshaft of the s1 object will be executed.

Class material consists of physical and geometrical properties (ρ, E, $\mathbf{G}$, A, I_p, I_a, etc.) of shaft and disks. Class bearing is inherited from the class shaft which consists of the number of bearings, member functions to calculate static and dynamic characteristics of the bearing etc. The inherited class bearing is shown in Fig. 2. In a similar way as the class shaft above, the two classes, material and bearing are framed to access the data to the main program. Class matrices consist of element stiffness matrices and global stiffness matrices (stiffness matrix, mass

```
class bearing: public shaft {
private:
float theta, z, **p,*h, L, U, eta;    //eta = η
float c, eps, W,Q,F;   //eps = ε
float mu,trise, **K,**C;    //**K,**C = stiffness
  & damping coefficients
public: //functions to calculate steady state &
  dynamic characteristics.
void calcsteady(float**p,float*h);
void calcdyna(float **K,float **C);};
```

FIGURE 2 Inherited bearing class.

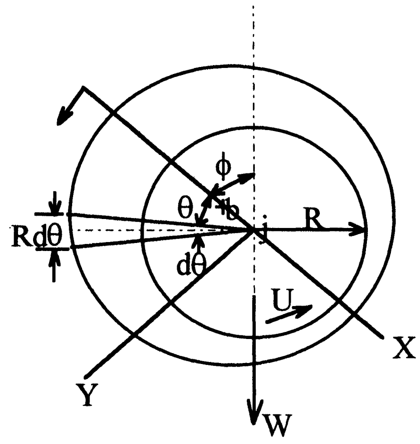

FIGURE 3 Journal bearing.

matrix, gyroscopic matrix, damping matrix and unbalance force matrix, etc.). Class eigenvalue is used for determining the critical speeds of the rotor. The different classes of objects for the expert system are Class input, Class rules and Class inference. Class input consists of details of actual and base values of machine. Class rules consists of the knowledge base rules of the rotating machinery, and Class inference for details of interpreting the data. The information transfer from one class to another class was achieved through messages.

2.1. Bearing Modeling

A schematic diagram of journal bearing is shown in Fig. 3. The two-dimensional Reynolds equation governing the flow of the lubricant in the clearance space of journal bearing under steady-state loading is

$$\frac{1}{R^2}\frac{\partial}{\partial\theta}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial z}\right)=\frac{\omega}{2}\frac{\partial h}{\partial\theta}. \quad (1)$$

By using the variation principles of calculus the above equation can be expressed in pressure functional as

$$J(p)=\int_{\theta=0}^{\theta=\pi+\alpha}\int_0^L\left[\frac{h^3}{24R^2\eta}\left(\frac{\partial p}{\partial\theta}\right)^2+\frac{h^3}{24\eta}\left(\frac{\partial p}{\partial z}\right)^2+\left(\frac{\omega}{2}\frac{\partial h}{\partial\theta}\right)p\right]\mathrm{d}\theta\,\mathrm{d}z. \quad (2)$$

The fluid film has been divided into triangular elements as in the reference Booker and Huebner (1972). Applying Reynolds boundary conditions $p=0$ at $\theta=0$ and at $z=0,L$; $p=(\partial p/\partial\theta)=0$ at $\pi+\alpha$.

The load carrying capacity, leakage of lubricant and frictional force are calculated by integrating the Eqs. (3)–(5) which are given in Cameron (1966).

$$W=\left[\left(\int_0^{\pi+\alpha}\int_0^L Rp\cos\theta\,\mathrm{d}\theta\,\mathrm{d}z\right)^2+\left(\int_0^{\pi+\alpha}\int_0^L Rp\sin\theta\,\mathrm{d}\theta\,\mathrm{d}z\right)^2\right]^{1/2}, \quad (3)$$

$$Q=-2R\left[\int_0^{2\pi}\frac{h^3}{12\eta}\left(\frac{\partial p}{\partial\theta}\right)\mathrm{d}\theta+\int_0^{2\pi}\frac{h^3}{12\eta}\left(\frac{\partial p}{\partial z}\right)\mathrm{d}\theta\right]+\int_0^L\frac{Uh}{2}\,\mathrm{d}z, \quad (4)$$

$$F=\int_0^{2\pi}\int_0^L\tau\,\mathrm{d}A=\int_0^{2\pi}\int_0^L\left[\frac{h}{2}\left(\frac{\partial p}{\partial\theta}\right)+\frac{\eta U}{h}R\right]\mathrm{d}\theta\,\mathrm{d}z, \quad (5)$$

$$\text{Coefficient of friction } (\mu)=F/W, \quad (6)$$

$$\text{Temperature rise } \Delta t=\frac{FU}{JmC_p} \text{ where } m=(\rho Q). \quad (7)$$

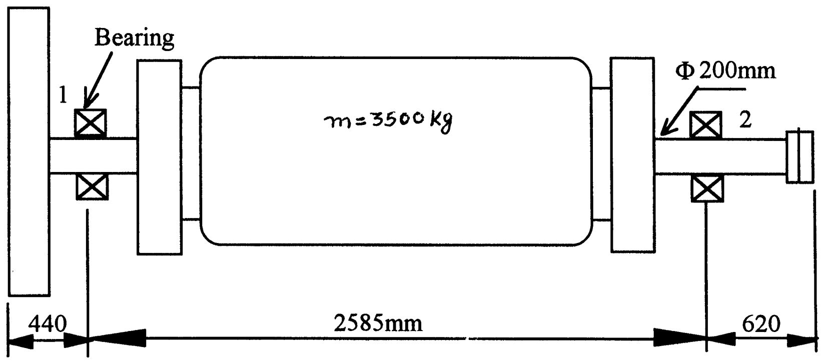

FIGURE 4 Electrical rotor.

Dynamic characteristics: Stiffness and damping coefficients of the journal bearing are calculated using the perturbation analysis presented by Lund (1987).

2.2. Rotor-Bearing System Modeling

The electrical rotor-bearing system taken for analysis is shown in Fig. 4. The schematic diagram of mathematical equivalent model is shown in Fig. 5. The rotor has been descritized into 50 beam elements. Equations of motion for a linearly damped rotor-bearing system proposed by Gmür and Rodrigues (1991) can be expressed in the matrix form as follows:

$$M\ddot{q} + (G + \eta 1 K)\dot{q} + (\eta_1 K + \eta_2 C)q = r \quad (8)$$

where r represents the $4N^e$ dimensional nodal unbalance force vector. The matrices M and K are symmetric, whereas G and C are skew-symmetric. Since the equations of motion are written in discrete form, the effects of thin rigid disks and short journal bearings can be incorporated directly into the formulation by adding adequate nodal contributions.

3. SOLUTION PROCEDURE

The problem has been formulated by using the finite element method. A computer code was developed in C++ in an object oriented paradigm. The formulated matrix equations are solved by Gauss elimination algorithm to obtain the bearing pressures in the clearance space of the bearing. Hessenberg algorithm is used to obtain the eigen values of the rotor bearing system. The stability limit of the rotor bearing system was obtained by plotting the values of growth factor versus rotational speed. The expert system code has been developed in C++ in an object-oriented framework. Figure 6 shows the flow chart for rotor dynamic analysis and its linkage to the expert system shell.

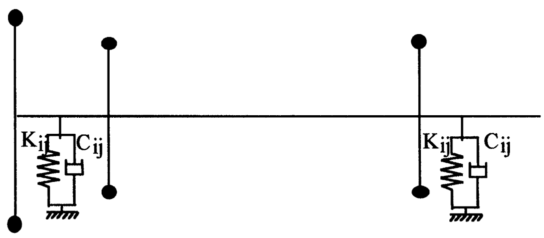

FIGURE 5 Mathematical equivalent model of rotor-bearing system.

4. RESULTS AND DISCUSSION

For a given rotor bearing system the program calculates the static and dynamic characteristics of the journal bearing and critical speeds, unbalance response, stability speed limit of the rotor. These values are taken as input to the actual value data

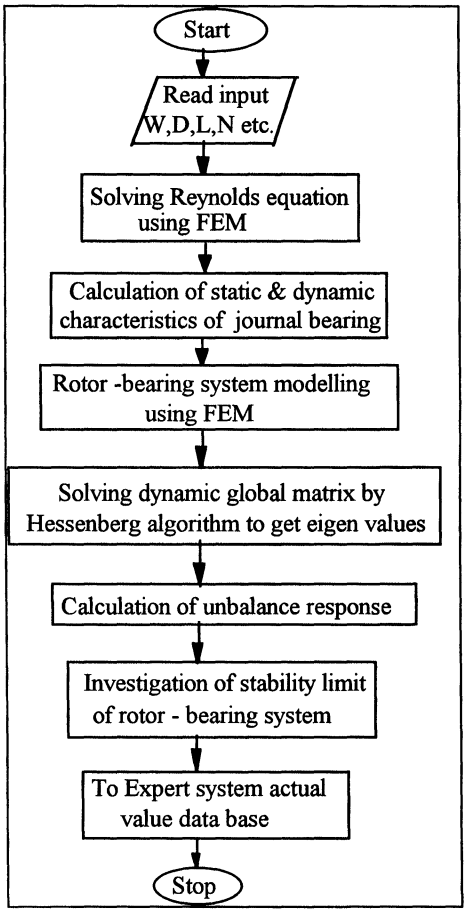

FIGURE 6 Flow chart showing rotor dynamic analysis and linkage to expert system shell.

base. The rotor shown in Fig. 4 is a rigid rotor supported on oil lubricated journal bearings. Total mass of the rotor is 3500 kg. The bearing details used for the analysis are $D = 0.2$ m, $L/D = 0.675$, $\eta = 0.025$ Pa s, $\varepsilon = 0.48$, $c = 160\,\mu$m, $N = 1500$ rpm. The steady state characteristics results of the journal bearing obtained from the equations (3–7) are $W = 17.44$ kN, $Q = 7.17 \times 10^{-4}\,\mathrm{m^3/s}$, $F = 251$ N, $\mu = 0.014$, $\Delta t = 5.18$°C. The coefficient of friction due to aerodynamic forces and unbalance whirling is assumed to be negligible. When the journal is perturbed by a small displacement and a small velocity, the new journal forces can be expressed by a Taylor series expansion, from which one can establish the stiffness and damping coefficients of

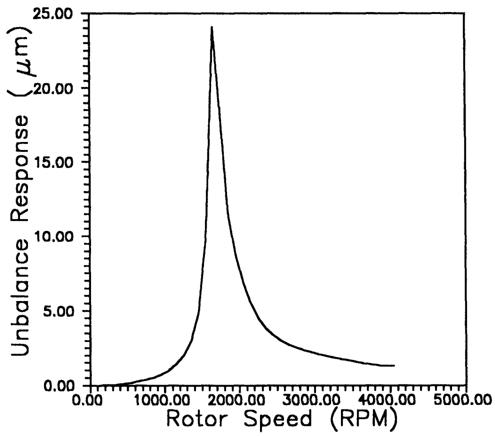

FIGURE 7 Rotational speed vs response.

the bearing. The values obtained are $K_{11} = 1.88 \times 10^8$, $K_{12} = 6.83 \times 10^7$, $K_{21} = 9.07 \times 10^6$, $K_{22} = 2.36 \times 10^8$ N/m, $C_{11} = 2.33 \times 10^5$, $C_{12} = 2.10 \times 10^5$, $C_{21} = 1.66 \times 10^5$, $C_{22} = 4.48 \times 10^5$ N s/m.

The static performance characteristic equations (3)–(7) and dynamic characteristics have compared with the software package REYNOLDS developed by the second author for bearing modeling using the finite element method in FORTRAN language. The results obtained by using the above software package for the same input values are $W = 17.50$ kN, $Q = 7.25 \times 10^{-4}\,\mathrm{m^3/s}$, $F = 260$ N, $\mu = 0.0148$, $\Delta t = 3.13$°C, $K_{11} = 1.92 \times 10^8$, $K_{12} = 6.73 \times 10^7$, $K_{21} = 8.97 \times 10^6$, $K_{22} = 2.48 \times 10^8$ N/m, $C_{11} = 2.13 \times 10^5$, $C_{12} = 1.83 \times 10^5$, $C_{21} = 1.34 \times 10^5$, $C_{22} = 4.24 \times 10^5$ N s/m. The former stiffness and damping coefficients values are used in the calculation of critical speeds, unbalance response and the stability of the rotor bearing system. The operating speed of the rotor is 1500 rpm. The rotor system is stable and free from subsynchronous vibration at operating speed. The unbalance eccentricity of the rotor is taken as 0.01 mm. Figure 7 shows the unbalance response of the rotor at the left end bearing point. The peak of the curve corresponds to first critical speed of the rotor, which is equal to 1780.2 rpm. The critical speed of the rotor is compared with the software package ROTODYNA developed by the second author for rotor-bearing modeling using the finite element method in FORTRAN. The calculated value of first critical speed of the rotor using ROTODYNA is 1772.8 rpm. The results obtained through OOP program were compared with the two software packages and found to be in good agreement. The amplitude of vibration at critical speed is 24.0 μm. From the ISO 2372 vibration severity chart for stable operation of the rotor, the vibration displacement peak-to-peak at the bearing cap at 1500 cpm is 10 μm, which is taken as the base value displacement of the rotor. These values can be directly fed to the data base. The available knowledge is implemented in the form of a set of rules. Some of the rules are shown in Appendix B, which will predict the type of malfunction of the rotating machinery through the inference engine.

The system uses data to match with the antecedents of every rule. If any rule matches, then that rule is said to be fired or triggered and the corresponding inference is added to the report generator. Suggestions and recommendations will be displayed by the system. For example, the actual vibration amplitude level at bearing point 1 at frequency 1*RPM is 24 μm, which exceeds the base value amplitude level by 14 μm. The rule R1 in Appendix B corresponds to a rotor unbalance malfunction. In practice, rotors can never be perfectly balanced because of manufacturing errors, such as inhomogeneties in material, manufacturing tolerances and gain or loss of material during operation, which increases the vibrational response of the rotor-bearing system. A state of imbalance occurs when the center of mass of the rotating system does not coincide with the center of rotation. If the frequency of excitation coincides with one of the natural frequencies of the system and the unbalance response envelop is narrow (In rules of Appendix B: =1 corresponds to true) and predominantly in the radial plane, and amplitude of vibration of journal at bearing point 1 exceeds the base value, then rule R1 is triggered. Since all the antecedents and consequents of the rule R1 satisfies the above input data, rule R1 gets triggered and deduces that shaft unbalance is true. This new inference causes the rule R4 gets triggered, and

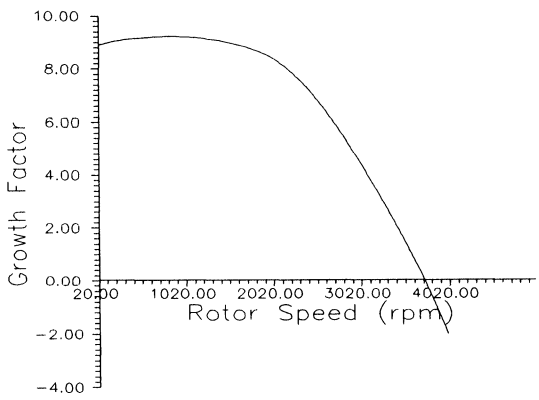

FIGURE 8 Rotational speed vs growth factor.

deduces "shaft unbalance detected". Check for eccentricity and mass unbalance; if necessary, balancing is to be done, is the recommendation and suggestion given by the system from rule R4.

Fig. 8 shows growth factor versus speed of the rotor, from which one can find that the threshold speed of the rotor is 3740.5 rpm. If the rotor reaches this speed and the amplitude of vibration exceeds the base value at bearing point 1, rule R3 gets triggered and deduces that the rotor has reached threshold speed. If 1X, 2X order frequencies are existing with 1X amplitude of vibration greater than 2X amplitude, the phase angle is equal to 180°, the envelop is narrow, the dominant plane is radial, and the actual vibration amplitude levels at bearing point 1 at frequency 1*RPM exceeds their respective allowable percentage change of base value amplitude level, then rule R5 gets triggered, and deduces that shaft misalignment has occurred. This new inference causes the rule R6 gets triggered, and deduces "misalignment of the shaft is detected". Check alignment is the recommendation given by the system. If frequencies 0.25*RPM, 0.33*RPM, 0.66*RPM, 1*RPM exist in the radial dominant plane with bearing lubricating oil temperature exceeding base value temperature 50°C by 60.0% (i.e. $50 \times 1.6 = 80$°C, $\Delta t = 30$°C) and bearing coefficient of friction exceeds 0.1, then R15 gets triggered and deduces that the malfunction is "light rotor rub in bearings". Similarly, the system uses data to match the compiled rules and gives the type of malfunction, such as a rotor crack, looseness, gear box problem, etc. and necessary action should be taken. The expert system should be run each time to take into account any effect due to changes in the parameters.

5. CONCLUSIONS

In the present work the static and dynamic characteristics of journal bearing and the vibration characteristics of a rotor-bearing system were obtained using the finite element method in an OOP framework. Using detailed deep knowledge of vibration analysis and bearing analysis one can predict the condition of the machine accurately. An off-line expert system for condition monitoring of a rotating machinery has been developed in an object oriented programming frame work. An object orientation is a promising paradigm for handling complexities in the large scale software applications. The development of a fault diagnostic system for rotating machinery requires a resolution of a number of issues involving expert systems, analytical methods and user interaction.

NOMENCLATURE

English Symbols

A	cross section area of rotor, m^2
c	radial clearance of bearing, m
C	circulation or pseudo-gyroscopic matrix
C_{ij}	damping coefficients of bearing, N s/m ($i,j = 1$ to 2)
C_p	specific heat of oil, kCal/kg K
D	diameter of the journal, m
e	distance between bearing center and journal center, m
E	Young's modulus, N/m^2
G	gyroscopic matrix
$\mathbf{G}$	shear modulus, N/m^2
h	film thickness $[c\,(1+\varepsilon\cos\theta)]$, m
I_a	cross sectional moment of inertia, kg m^2
I_p	polar moment of inertia, kg m^2
K	stiffness matrix
K_{ij}	stiffness coefficients of bearing N/m ($i,j = 1$ to 2)
L	length of the bearing, m
m	mass flow rate of the oil, kg/s
M	mass matrix
n	number of gear teeth in meshing
N	rotor speed, rev/min
N^e	number of nodes
p	hydrodynamic pressure, Pa
PT1	location of bearing 1
q	generalized displacements, m
Q	total leakage of oil through bearing, m^3/s
r	nodal unbalance force vector, N
R	radius of the journal, m
Δt	temperature rise °C
U	velocity component in x direction, m/s
$\mathbf{W}$	weight of the rotor, N
W	load carrying capacity of bearing, N
z	axial coordinate of bearing, m
1X	1*rpm

Greek Symbols

α	film extent, rad
ε	eccentricity ratio (e/c)
$\eta 1$	viscous loss factor, s
η	viscosity of oil, Pa s
η_1,η_2	auxiliary damping factors
μ	coefficient of friction
θ	circumferential coordinate, rad
$\boldsymbol{\rho}$	mass density of the shaft, kg/m^3
ρ	density of the oil (900), kg/m^3
τ	shear stress, Pa
ω	angular velocity, rad/s

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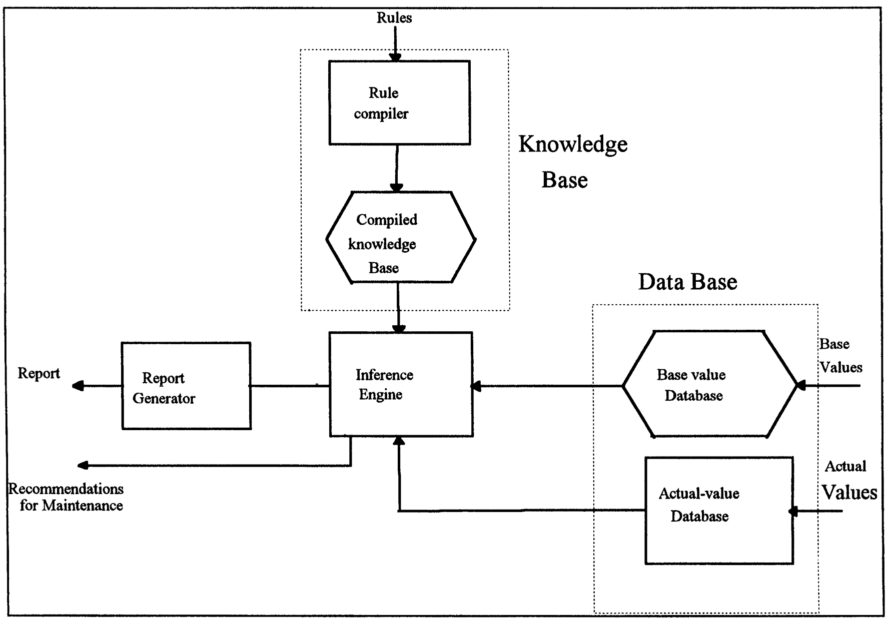
APPENDIX A: THE ARCHITECTURE OF EXPERT SYSTEM



APPENDIX B: RULES FOR ROTATING MACHINERY

R1:
IF FREQUENCY = (1*RPM)
AND NARROW_BAND_ENVELOP = 1
AND DOMINANT_PLANE_RADIAL = 1
AND AMPLITUDE_AT_PT1 >
(BASE_VALUE_AT_P1*1.20)
THEN SHAFT_UNBALANCE
R2:
IF FREQUENCY = (1*RPM)
AND AMPLITUDE_AT_PT1 >
(BASE_VALUE_AT_P1*1.20)
AND PHASE_ANGLE == 90
THEN CRITICAL_SPEED
R3:
IF RPM = 2.1*CRITICAL_SPEED
AND AMPLITUDE_AT_PT1 >
(BASE_VALUE_AT_P1*1.20)
THEN THRESHOLD_SPEED
R4:
IF SHAFT_UNBALANCE
THEN "SHAFT UNBALANCE DETECTED"
AND "CHECK FOR ECCENTRICITY AND MASS UNBALANCE"
AND "IF NECESSARY BALANCING IS TO BE DONE"
R5:
IF FREQUENCY = (1*RPM)
IF FREQUENCY = (2*RPM)
AND 1X_AMPLITUDE > 2X_AMPLITUDE
AND PHASE_ANGLE = 180
AND NARROW_BAND_ENVELOP = 1
AND DOMINANT_PLANE_RADIAL = 1
AND AMPLITUDE_AT_PT1 >
(BASE_VALUE_AT_P1*1.20)
THEN SHAFT_MISALIGNMENT
R6:
IF SHAFT_MISALIGNMENT
THEN "MISALIGNMENT OF THE SHAFT IS DETECTED"
AND "CHECK ALIGNMENT"
R7:
IF FREQUENCY = (0.48*RPM)
IF FREQUENCY = (1*RPM)
AND 0.48X_AMPLITUDE > 1X_AMPLITUDE
AND DOMINANT_PLANE_RADIAL = 1
AND AMPLITUDE_AT_PT1 >
(BASE_VALUE_AT_P1*1.20)
THEN ROTOR_INSTABILITY
R8:
IF ROTOR_INSTABILITY
THEN "ROTOR INSTABILITY DETECTED"
AND "LOWER OR RAISE OIL VISCOSITY"
R9:
IF FREQUENCY = (*n**RPM)
AND DISCRETE_ENVELOP = 1
AND AMPLITUDE_AT_PT1 >
(BASE_VALUE_AT_P1*1.20)
THEN GEAR_BOX_PROBLEM
R10:
IF FREQUENCY = (1*RPM)
AND FREQUENCY = (2*RPM)
AND FREQUENCY = (3*RPM)
AND AMPLITUDE_AT_PT1 >
(BASE_VALUE_AT_P1*1.20)
AND DECREASE_IN_CRITICAL_SPEED
THEN CRACK_IN_THE_ROTOR
R11:
IF FREQUENCY = (1*RPM)
AND AMPLITUDE_AT_PT1 >
(BASE_VALUE_AT_P1*1.20)
AND SPLIT_IN_CRITICAL_SPEED
AND
ONE_CRITICAL_SPEED_INCREASES_AND_ANOTHER_DECREASES
THEN GYROSCOPIC_EFFECT
R12:
IF GEAR_BOX_PROBLEM
THEN "PROBLEM WITH GEAR BOX DETECTED"
AND "CHECK FOR RUN OUT ECCENTRICITY AND CHECK ALIGNMENT"
R13:
IF FREQUENCY = (1*RPM)
AND FREQUENCY = (2*RPM)
AND FREQUENCY = (3*RPM)
AND FREQUENCY = (4*RPM)
AND 2X_AMPLITUDE > 1X_AMPLITUDE
AND AMPLITUDE_AT_PT1 >
(BASE_VALUE_AT_P1*1.20)
THEN LOOSENESS
R14:
IF LOOSENESS
THEN "PROBLEM WITH LOOSENESS OF PARTS DETECTED"
AND "CHECK FOR LOOSENESS OF BEARINGS, PEDESTAL, FOUNDATION ETC."
AND "CHECK ALIGNMENT"
R15:
IF FREQUENCY = (0.25*RPM)
AND FREQUENCY = (0.33*RPM)
AND FREQUENCY = (0.66*RPM)
AND FREQUENCY = (1*RPM)
AND DOMINANT_PLANE_RADIAL = 1
AND AMPLITUDE_AT_PT1 >
(BASE_VALUE_AT_P1*1.20)
AND ACTUAL_VALUE_OIL_TEMP_AT_PT1 >
BASE_VALUE_OIL_TEMP_AT_P1*1.60)
AND COEF_FRICTION > 0.1
THEN ROTOR_RUB.