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OBJECT ORIENTED PROGRAMMING AND EXPERT SYSTEMS IN ROTATING MACHINERY

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ABSTRACT

In any rotating machinery, unbalance and misalignment of the rotor are the two major malfunctions. Perfect alignment of shafts never exists. Improper aligning of shafts through couplings often leads to severe vibration problems in many rotating machines. In the present work effect of flexible coupling misalignment on lateral vibrational characteristics of a rotor-bearing system has been studied in an Object Oriented Programming framework using finite element method. An off-line rule based expert system has been developed for condition monitoring of a gas turbine, which predicts the machine condition severity and the cause of the problem. The system is implemented in C++.

NOMENCLATURE

English symbols

A = cross section area of rotor, m^2
c = distance between articulated points, m
C = circulation or pseudo-gyroscopic matrix
E = young's modulus, N/m^2
G = gyroscopic matrix
G = shear modulus, N/m^2
I_a = cross sectional moment of Inertia, m^4
I_p = polar moment of Inertia, m^4
K = stiffness matrix
K_b = bending stiffness of diaphragm, N/m
k_f = profile correction factor
M = mass matrix
n = number of gear teeth in meshing
n_d = number of diaphragms
N = rotor speed, rev/min
N^e = number of nodes
q = generalized displacements, m
r = nodal unbalance force vector, N
R_a, R_b = diaphragm root and outer radii, m
r^c = coupling nodal force vector, N
${}^i r^c$ = local unbalance force vector, N

Greek symbols

η = viscous loss factor, s
η_1, η_2 = auxiliary damping factors
ρ = mass density of the shaft, kg/m^3
ω = angular velocity, rad/s

1. INTRODUCTION

Computers remain an incomplete tool for engineers and scientists. The capabilities of Object oriented programming in recent years have now made the programming paradigm a very promising tool for the development and implementation of large-scale scientific codes. The technique or methodology to organize a program is called Object Oriented Programming (OOP). In an Object-oriented programming, physical or logical entities are represented by objects. Each object is an instance of class. A class describes a set of physical or conceptual objects that have similar properties, constraints and operations. The definition of operations between objects are called methods and the invocation of the methods is referred to as 'passing a message'. The main features of the object oriented programming are Data encapsulation, Operator overloading and Inheritance. Each object encapsulates (hides) its data attributes from other objects and only shows its behaviour, which enhances portability of application codes. Operator overloading allows the programmer to think in the same terms for application of a function to, two different objects. A class may have

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several derived classes that may inherit some of its attributes and behaviour, which enables efficient and reusability of codes. In OOP both data and subroutines are linked intrinsically. Fig. 1 shows how data and subroutines are linked in OOP and non-OOP programs.

Makie (1992) has described the implementation of OOP for structural dynamics problems using finite element method and illustrated the advantages of the approach. He showed OOP approach reduces the scope for bugs by encouraging clear thinking about program design and allowing programs to be substantially altered without the need to change existing code. Zeglinski et al. (1994) have discussed the concepts of OOP for finite element method using C++ language in generalized matrix library and described that the efficiency, flexibility and maintainability of the C++ program is shown to be superior to a comparable version written in a non-OOP language, such as FORTRAN . Ross et al. (1992a, 1992b) have described OOP techniques and design for scientific code development using C++. Bettig et al. (1995) describes the concepts of OOP for predictive maintenance of a rotating machinery, in which Trend variables (e.g. Bearing rotordynamic coefficients) are trended to predict the machine life/maintenance period using finite element model in graphical framework.

Rotor unbalance and shaft misalignment are the two main sources of vibration in rotating machinery. 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. The flexible couplings are used to accommodate misalignment between the machines connected. Woodcock (1977) has studied the effect of coupling geometry, mass, its location and the coupling mass unbalance level on the lateral vibration of the rotating machines. He found out that the misalignment is not the cause of vibration but the reaction forces and moments developed on the machines due to misalignment are the causes of vibration. Gibbons (1976) has shown the forces and moments acting at the coupling centers of articulation in the presence of parallel misalignment and has also given the expressions for calculating the forces and moments for gear and flexible element couplings. Dewell et al. (1984) have developed expressions for the expected frequencies of vibration for a misaligned metallic-disc flexible coupling and shown experimentally only 2ω and 4ω harmonic components have significant changes as misalignment increases. Simon (1992) has added a pair of moments and forces at the coupling locations to simulate the parallel and angular misalignment conditions. Xu et al. (1994a, 1994b) have carried out theoretical vibration analysis of a rotor coupling system with a universal joint. And also experiments have been carriedout with a helical coupling. Sekhar and Prabhu (1995) has introduced the reaction forces and moments due to misalignments at the coupling node and studied the vibration response at 2ω harmonic using a higher order finite element with eight degrees of freedom per node . Arumugam et al. (1995) have discussed the effect of flexible actual data in the data base and detects the discrepancies by using coupling element reaction forces on the vibration characteristics of a two-stage turbine rotor-bearing system using FEM. In the present work the effect of coupling misalignment on lateral vibrational characteristics of a rotor bearing system at 2ω and 4ω harmonics are analyzed. Gmür et al. (1991) have proposed 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.

Condition monitoring aims at monitoring parameters of the machines so that the required preventive maintenance could be carried out as and when problems begin to develop in machines. The key parameters to be monitored for sophisticated rotating machinery are many. It is very difficult to interpret all the data received from the sensors and to conclude necessary inferences without missing a single parameter. Recently, the literature has given emphasis on expert systems for condition based maintenance (Zimmer and Bently, 1986; Rao, 1993). Expert system is an Artificial Intelligence system created to solve problems in a particular domain. An expert system for Condition Monitoring purposes consists of three key components as follows (Hill and Baines, 1988):

- Knowledge base
- Inference engine
- Data base

These components are linked together to produce a system that advises, informs and solves problems in a manner similar to that of human experts. Knowledge base consists of machines knowledge in the form of "experiential" (rules, decision trees etc.) (Vesonder et al., 1983) or "Deep models" (analytical models, qualitative models, etc.) (De Kleer and Williams, 1987; Mahabala et al., 1994) depending upon the precision and the complexity of the machine to be monitored. In the present paper a rule-based expert system for condition monitoring of a Gas turbine is discussed. Knowledge of the machine is stored in the form of IF-THEN rules. A rule has the following form

IF <antecedent1>
AND <antecedent2>
AND <antecedent3>
....
AND <antecedent n>
THEN <consequent1>
AND <consequent2>
AND <consequent3>
....
AND <consequent m>

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 rules. Forward chaining and Backward chaining are the two most common strategies employed to perform the inference process. The discrepancies are reported to the operator. 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 architecture of the Expert System is shown in appendix-

1. The rule compiler converts the rules about a machine to a form expected by the inference engine. Experimentally measured values through sensors can fed to the data base directly (e.g. FFT-Fast Fourier Transform , which converts the data received from the sensors in the form of time domain to frequency domain). If the sensor data need not to be converted, that data could be directly fed to 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.

2. ANALYSIS

The rotor analysis using finite element method consists of different classes of objects. They are :Class shaft, Class material, Class matrices and Class eigenvalue. Class shaft consists, details of number of elements, number of nodes, type of element, type of shape function . Class material consists of physical and geometrical properties (ρ, E, G, A,I_p, I_a, etc.) of shaft and disks . Class matrices consists of element stiffness matrices and global stiffness matrices (stiffness matrix, mass matrix, gyroscopic matrix, damping matrix and unbalance force matrix, etc.). Class eigenvalue is for getting the critical speeds of the rotor. Class coupling is derived from the class shaft that consists of details of coupling misalignment. The information from one class to another class was brought through messages.

2.1 Rotor-bearing system modelling

The equations of motion for a linearly damped rotor system (Gmür and Rodrigues, 1991) are expressed in the matrix form as follows:

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

where r represents the $4N^c$ dimensional nodal unbalance force vector. The matrices M and K are symmetric ,where as 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 directly incorporated into the formulation by adding adequate nodal contributions.

2.2 Coupling misalignment modelling

The misalignment moments and the reaction forces are acting as periodic loads on the rotating shafts. The periodic function is half sinusoidal function with period of π/ω, where ω is the angular velocity of the shaft. Knowing the magnitude of the reaction forces one can find the amplitude at the different harmonics by Fourier method. In the present analysis 2ω and 4ω components of the reaction forces are considered and they are incorporated into the Eq. (1).

$$M\ddot{q}+(G+\eta K)\dot{q}+(\eta_1 K+\eta_2 C)q=r+r^c \qquad (2)$$

$$r^c=\sum_{i=1}^{2}\left({}^iQ^c\right)^T\,{}^ir^c \qquad (3)$$

where ${}^iQ^c$ represents 4 x $4N^c$ transformation matrices that transfer the local contributions, due to the coupling misalignment, to the appropriate locations in the global force vector. Flexible diaphragm coupling is taken for analysis.

3. SOLUTION PROCEDURE

The problem has been formulated by using 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.

4. RESULTS AND DISCUSSIONS

The schematic diagram of a two stage turbine rotor-bearing-coupling system Trivisonna (1973) is shown in Fig. 2, for which vibration characteristics are obtained in parallel and angular misalignment condition. The bearings are assumed to be isotropic. The values of the stiffness and damping coefficients of the bearings 1 and 2 are 8.75 x 10^5 N/m and 100 Ns/m and for the bearings 3 and 4 are 8.75 x 10^7 N/m and 100 Ns/m. The viscous loss factor is taken as 5.0 10^{-4}s . The flexible diaphragm coupling details considered for analysis are, diaphragm thickness ratio t_a is 0.008; radius R_a is 55 mm; radii ratio (R_b/R_a) is 0.52; the numerical coefficient k_3 is 0.99 and profile correction factor k_8 is 1.0 and the number of diaphragms n at the either end of the coupling spacer is three. The bending spring rate of a single diaphragm is calculated using the formulae (Bloch and Geitner, 1990)

$$K_b=n_d k_3 k_8 \pi E R_a^3 (t_a/R_a)^3 \qquad (4).$$

Figure 3 shows the unbalance response of the rotor bearing system and one can observe that the first critical speed is occurring at 4070 rpm. The vibration response of a turbine rotor at second and fourth harmonic of ω due to parallel misalignment is shown in Figs. 4 and 5. The dynamic response is non dimensionalised against the magnitude of the parallel misalignment. The magnitude of 2ω component is higher than 4ω component. The resonance peak of 2ω is occuring at half of the first critical speed and 4ω at one-fourth of the first critical speed. From the Fig. 6 it is clear that vibration amplitudes are varying linearly with the magnitude of parallel misalignment, this is due to the assumption of linear variation of torque with speed. The rated torque is equal to 100 N-m at the speed of 6000 rpm. Figure 7 shows the vibration response of rotor due to angular misalignment in which subcritical resonance of 4ω is weak when compared to 2ω.

4.1 Development of rule based expert system for condition monitoring of a Gas Turbine:

The schematic diagram of a Gas Turbine is shown in Fig. 8.The experimentally measured values regarding Gas Turbine has taken from the reference (Kurup et al., 1995). Pick ups are taken at the points 1, 2, and 3 which corresponds to Low pressure compressor forward position, Low pressure aft bearing, High pressure compressor forward bearing. Point 4 corresponds to High pressure compressor aft bearing. Frequency spectrum values for points 1 to 3 at various frequencies are given in appendix-2. The percentage change of values allowed at the points 1 to 3 are 10%, 15% and 20% respectively. Tables 1 and 2 summarizes the magnitudes of vibration levels of the gas turbine at the bearing points 1, 2 and 3, at 60 Hz and 120 Hz.

TABLE 1 Vibration magnitude values at 60 Hz

point	BV (dB)	AV (dB)	Allowed %change	Remarks
1	80	110	10	Abnormal
2	86	101	15	Abnormal
3	90	95	20	Normal

TABLE 2 Vibration magnitude values at 120 Hz

point	BV (dB)	AV (dB)	Allowed % change	Remarks
1	65	76	10	Abnormal
2	69	82	15	Abnormal
3	75	94	20	Abnormal

Some of the rules developed for the Gas turbine are indicated in appendix-3. The system uses data to match with the antecedents of every rule. If any rule matches, that rule will be fired and the corresponding inference is added into the database. Then suggestions and recommendations will be displayed by the system. For example the actual vibration amplitude level at bearing point 1 at 60 Hz (1*RPM) is 110 dB, which exceeds allowable percentage change of base value amplitude level (i. e. 80 x 1.10 = 88 dB) by 22 dB. So the rule R1 gets fired, and deduces that shaft unbalance is true. This new inference causes the rule R13 gets fired, and 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 R13. From the table 2, the actual vibration amplitude levels at bearing points 1, 2 and 3 at 120 Hz (2*RPM) exceeds their respective allowable percentage change of base value amplitude level. So the rules R4, R5 and R6 gets fired, and deduces that shaft misalignment is true. This new inference causes the rule R14 gets fired, and deduces " misalignment of the shaft is detected". Check alignment is the recommendation given by the system.

5. CONCLUSIONS

In the present work the Vibration response of a two stage turbine rotor-coupling-bearing system at 2ω and 4ω harmonics have been obtained considering parallel and angular misalignment in the coupling, in an Object oriented programming paradigm. The magnitude of 2ω harmonic is higher than 4ω harmonic component. The results obtained are compared with the existing literature and found to be in agreement. An Expert system for condition monitoring of a Gas Turbine has been developed, which is expected to be a cost effective tool for predictive maintenance of machines by minimizing the machines downtime, increasing its life and reducing operating costs.

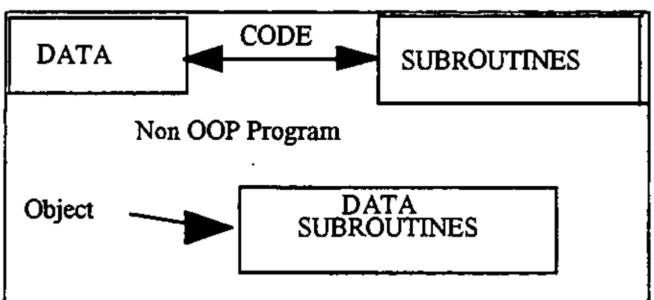

Fig . 1 How data and subroutines are linked in OOP and Non-OOP programs

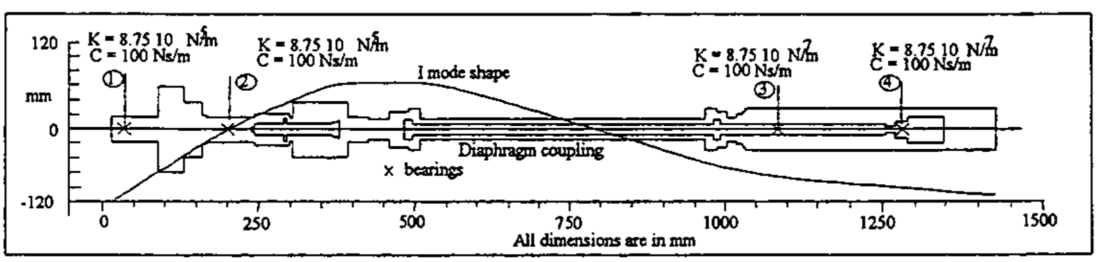

Fig. 2 A two-stage rotor-bearing-coupling system

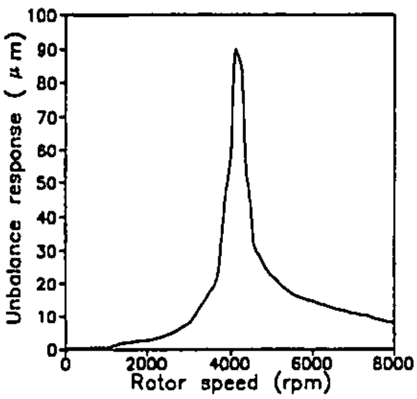

Fig. 3 Unbalance of the rotor at the second bearing

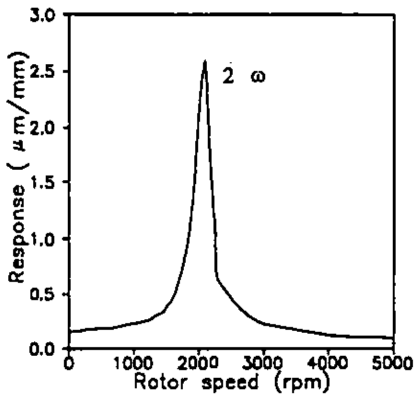

Fig. 4 Parallel misalignment response at the second bearing non dimensionalised against magnitude of misalignment.

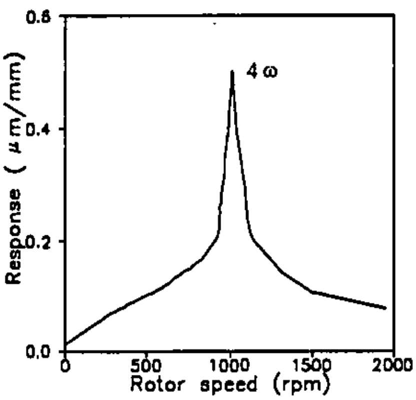

Fig. 5 Parallel misalignment response at the second bearing non dimensionalised against magnitude of misalignment.

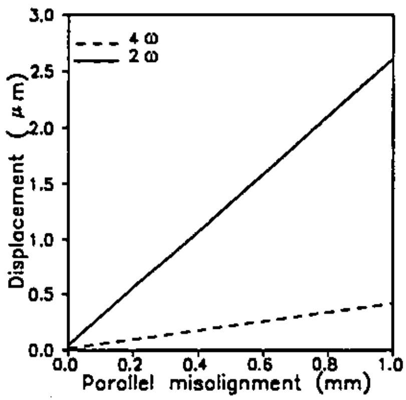

Fig. 6 Misalignment responses of 2ω and 4ω components

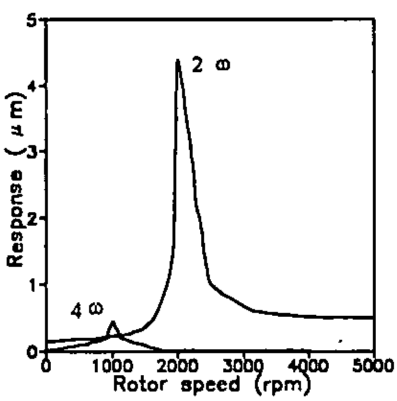

Fig. 7 Angular misalignment response at the second bearing

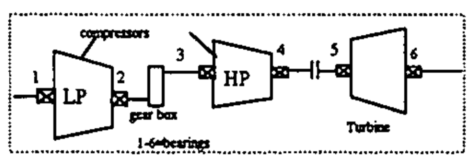

Fig. 8 Schematic diagram of Gas Turbine

6. REFERENCES

Arumugam, P., Swarnamani, S., and Prabhu, B. S., 1995b, "Effects of coupling misalignment on the vibration characteristics of a two stage turbine rotor," ASME, Proceedings, Design Engineering Technical Conference, Boston, USA, Vol. 3, 1049-1054.

Bettig, B. P., and Han, R. P. S., 1995, "An Object-Oriented Program framework for Predictive Maintenance in Rotordynamics," Proceedings, International Conference on Advances in Mechanical Engineering, IISc, Bangalore, India, Vol. 2, pp. 1079-1096.

Bloch , H. P. and Geitner, F. K., 1990, An introduction to Machinery reliability assessment, Van Nowrand Reinhold.

De Kleer, J., and Williams, B. C., 1987, "Diagnosing multiple faults, Artificial Intelligence," Vol. 32, pp. 97-130.

Dewell, D. L., and Mitchell, L. D., 1984 "Detection of misaligned disk coupling using spectrum analysis," ASME Journal of Vibration, Acoustics, Stress and Reliability in Design, Vol. 106, pp. 9-16.

Gibbons, C. D., 1976, "Coupling misalignment Forces," Proceedings, 5th Turbomachinery Symposium, Gas Turbine Laboratories, Texas A&M University, College Station Texas, pp. 125-131.

Gmür, T. C., and Rodrigues, J. D., 1991, "Shaft Finite Elements for Rotor Dynamics Analysis," ASME Journal of Vibration and Acoustics, Vol. 113, pp. 482-493.

Hill, J. W., and Baines, N. C., 1988, "Application of an expert system to rotating machinery, Vibrations of Rotating Machinery," Proceedings, I Mech.E, C307/88, pp. 449-454.

Mahabala, H. N., Arunkumar, A. T., Kurup, R. R., and Ravi Prakash, G., 1994, "Diagnosis of Machine Faults," SADHANA, Proceedings, Engineering Sciences, Indian Academy of Sciences, Bangalore, India, Vol. 19,pp. 23-50.

Makie, I. R., 1992, "Object Oriented Programming of the Finite Element Method ," Int. Journal for Numerical Methods in Engineering, Vol. 35,pp. 425-436.

Kurup, R. R., Mahabala, H. N., Ravi Prakash, G., and Prabhu, B. S., 1995, "An Intelligent Predictive Maintenance System," Proceedings,. National Conference on Improved Maintenance Strategy in Power Plants, May 19-20, Hyderabad, India, pp. 209-215.

Rao, J. S., 1993, "Future perspective condition monitoring with special emphasis on expert systems, Frontiers of Tribology and Condition Monitoring," Tata Mc Graw Hill, New Delhi, pp.87-101.

Ross, T. J, Wagner, L. R. and Lugner, G. F., 1992a, "Object-Oriented Programming for scientific codes. I: Thoughts and Concepts-I," ASCE Journal of computing in Civil Engineering, Vol. 6, pp. 480-496.

Ross, T. J, Wagner, L. R. and Lugner, G. F., 1992b, "Object-Oriented Programming for scientific codes. II: Thoughts and Concepts, ASCE Journal of computing in Civil Engineering, Vol. 6, pp. 497-514..

Sekhar, A. S., and Prabhu, B. S., 1995, "Effects of coupling misalignment on vibrations of rotating machinery," Journal of Sound and Vibration, Vol. 185, pp. 655-671.

Simon, G., 1992, "Prediction of vibration behaviour of large turbomachinery on elastic foundations due to unbalance and coupling misalignment," Proceedings, Institution of Mechanical Engineeers, Journal of Mechanical Engineering Science, Vol. 206C, pp. 29-39.

Trivisonno, R. J., 1973, "FORRAN IV Computer program for calculation critical speeds of rotating shafts," NASA Technical Note, NASA TN D-7385.

Vesonder, G. T., Stolfo, S. J., Zieliski, J. E., Miller, F. D., and Copp, D. H., 1983, "An Expert system for telephone cable maintenance". Proceedings, 8th International Joint Conference on Artificial Intelligence, Karlsruhe, Germany, Vol.2, pp. 116-121.

Woodcock, J. S., 1977, "The effects of couplings upon the vibrations of rotating machinery," Proceedings, International Conference on Flexible Couplings, University of Sussex Bright U. K., Farnham, UK, Michael Neale and Associates Ltd, pp. E1. 1-20.

Xu, M., and Marangoni, R. D:, 1994a, "Vibration analysis of a motor-flexible coupling-rotor system subject to misalignment and unbalance, Part I: Theoretical model and analysis," Journal of Sound and Vibration, Vol. 176, pp. 663-679.

Xu, M., and Marangoni, R. D., 1994b, "Vibration analysis of a motor-flexible coupling-rotor system subject to misalignment and unbalance, Part II : Experimental validation," Journal of Sound and Vibration, Vol. 176, pp. 681-691.

Zeglinski, G. W., Han, R. P. S., and Aitchison, P., 1994, "Object Oriented Matrix Classes for use in a Finite Element Code using C++," Int. Journal for Numerical Methods in Engg., Vol. 37, pp. 3921-3937.

Zimmer, S., and Bently, D. E., 1986, "Predictive maintenance programs for rotating machinery using computerized vibration monitoring systems," Proceedings, 1st European Sponsored Turbomachinery Symposium, Brunel University, PP. 91-107.

Appendix-1

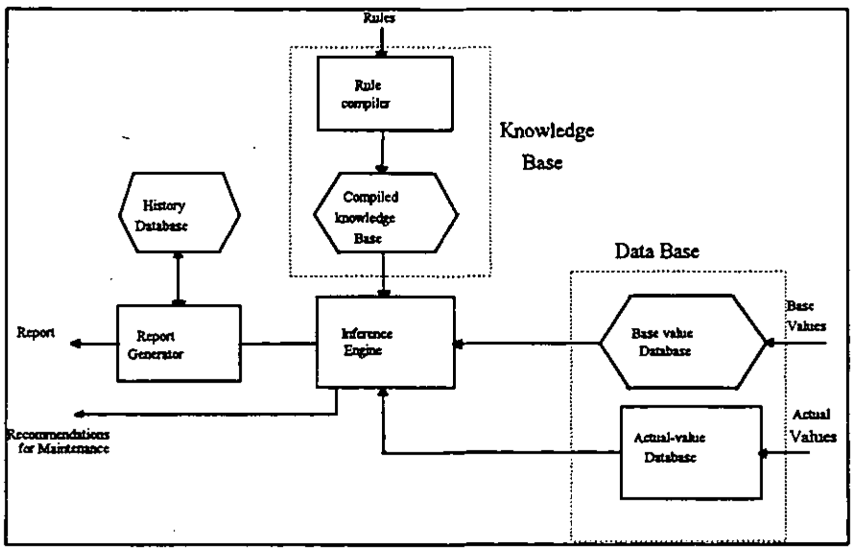

Fig. 9 The architecture of the Expert system

Appendix - 2

Frequency (Hz)	BV-p1 (dB)	AV-p1 (dB)	BV-p2 (dB)	AV-p2 (dB)	BV-p3 (dB)	AV-p3 (dB)
26	20	24	25	27	31	33
40	24	26	30	35	31	32
60	80	110	86	101	90	95
70	28	32	30	32	35	33
100	28	32	35	36	38	39
120	65	76	69	82	75	94
160	28	31	32	34	36	38
224	30	34	37	42	44	46

BV-p1 Base Value at point 1, AV-p1-Actual Value at point 1,BV-p2, p3 and AV-p2, p3 corresponds to second and third points.

Fig. 10 Frequency spectrum values of gas turbine at bearing points 1, 2 and 3, for different frequencies

Appendix -3

Rules for Gas Turbine Machine

R1 :
IF FREQUENCY = (1*RPM)
AND AMPLITUDE_AT_PT1 > (BASE_VALUE_AT_P1*1.10)
THEN SHAFT_UNBALANCE
R2 :
IF FREQUENCY = (1*RPM)
AND AMPLITUDE_AT_PT2 > (BASE_VALUE_AT_P2*1.15)
THEN SHAFT_UNBALANCE
R3 :
IF FREQUENCY = (1*RPM)
AND AMPLITUDE_AT_PT3 > (BASE_VALUE_AT_P3*1.20)
THEN SHAFT_UNBALANCE
R4 :
IF FREQUENCY = (2*RPM)
AND AMPLITUDE_AT_PT1 > (BASE_VALUE_AT_P1*1.10)
THEN SHAFT_MISALIGNMENT
R5 :
IF FREQUENCY = (2*RPM)
AND AMPLITUDE_AT_PT2 > (BASE_VALUE_AT_P2*1.15)
THEN SHAFT_MISALIGNMENT
R6 :
IF FREQUENCY = (2*RPM)
AND AMPLITUDE_AT_PT3 > (BASE_VALUE_AT_P3*1.20)
THEN SHAFT_MISALIGNMENT
R7 :
IF FREQUENCY = (n*RPM)
AND AMPLITUDE_AT_PT1 > (BASE_VALUE_AT_P1*1.10)
THEN GEAR_BOX_PROBLEM
R8 :
IF FREQUENCY = (n*RPM)
AND AMPLITUDE_AT_PT2 > (BASE_VALUE_AT_P2*1.15)
THEN GEAR_BOX_PROBLEM
R9 :
IF FREQUENCY = (n*RPM)
AND AMPLITUDE_AT_PT3 > (BASE_VALUE_AT_P3*1.20)
THEN GEAR_BOX_PROBLEM
R10 :
IF FREQUENCY = (0.48*RPM)
AND AMPLITUDE_AT_PT1 > (BASE_VALUE_AT_P1*1.10)
THEN ROTOR_INSTABILITY
R11 :
IF FREQUENCY = (0.48*RPM)
AND AMPLITUDE_AT_PT2 > (BASE_VALUE_AT_P2*1.15)
THEN ROTOR_INSTABILITY
R12 :
IF FREQUENCY = (0.48*RPM)
AND AMPLITUDE_AT_PT3 > (BASE_VALUE_AT_P3*1.20)
THEN ROTOR_INSTABILITY
R13 :
IF SHAFT_UNBALANCE
THEN "SHAFT UNBALANCE DETECTED"
AND "CHECK FOR ECCENTRICITY AND MASS UNBALANCE"
AND "IF NECESSARY BALANCING IS TO BE DONE"
R14 :
IF SHAFT_MISALIGNMENT
THEN "MISALIGNMENT OF THE SHAFT IS DETECTED"
AND "CHECK ALIGNMENT"
R15 :
IF ROTOR_INSTABILITY
THEN "ROTOR INSTABILITY DETECTED"
AND "LOWER OR RAISE OIL VISCOSITY"
R16 :
IF FREQUENCY = (1*RPM)
AND AMPLITUDE_AT_PT1 > (BASE_VALUE_AT_P1*1.10)
AND AMPLITUDE_AT_PT2 > (BASE_VALUE_AT_P2*1.15)
AND AMPLITUDE_AT_PT3 > (BASE_VALUE_AT_P3*1.20)
THEN CRITICAL_SPEED
R17 :
IF FREQUENCY = (1*RPM)
AND FREQUENCY = (2*RPM)
AND FREQUENCY = (3*RPM)
AND AMPLITUDE_AT_PT1 > (BASE_VALUE_AT_P1*1.10)
AND DECREASE_IN_CRITICAL_SPEED
THEN CRACK_IN_THE_ROTOR
R18 :
IF FREQUENCY = (1*RPM)
AND AMPLITUDE_AT_PT1 > (BASE_VALUE_AT_P1*1.10)
AND SPLIT_IN_CRITICAL_SPEED

AND
ONE_CRITICAL_SPEED_INCREASES_AND_ANOTHER_DECREASES
THEN GYROSCOPIC_EFFECT
R19 :
IF GEAR_BOX_PROBLEM
THEN "PROBLEM WITH GEAR BOX DETECTED"
AND "CHECK FOR RUN OUT ECCENTRICITY AND CHECK ALIGNMENT"
R20 :
IF FREQUENCY = (1*RPM)
AND FREQUENCY = (2*RPM)
AND FREQUENCY = (3*RPM)
AND FREQUENCY = (4*RPM)
AND AMPLITUDE_AT_PT1 > (BASE_VALUE_AT_P1*1.10)
THEN LOOSENESS
R21 :
IF LOOSENESS
THEN "PROBLEM WITH LOOSENESS OF PARTS DETECTED"
AND "CHECK FOR LOOSENESS OF BEARINGS, PEDESTAL, FOUNDATION ETC." AND "CHECK ALIGNMENT"
R22 :
IF FREQUENCY = (0.25*RPM)
AND FREQUENCY = (0.33*RPM)
AND FREQUENCY = (0.5*RPM)
AND FREQUENCY = (1*RPM)
AND FREQUENCY = (1.5*RPM)
AND FREQUENCY = (2*RPM)
AND AMPLITUDE_AT_PT1 > (BASE_VALUE_AT_P1*1.10)
THEN ROTOR _RUB
R23 :
IF ROTOR_RUB
THEN " ROTOR RUB DETECTED"
AND "CHECK FOR RUBS "