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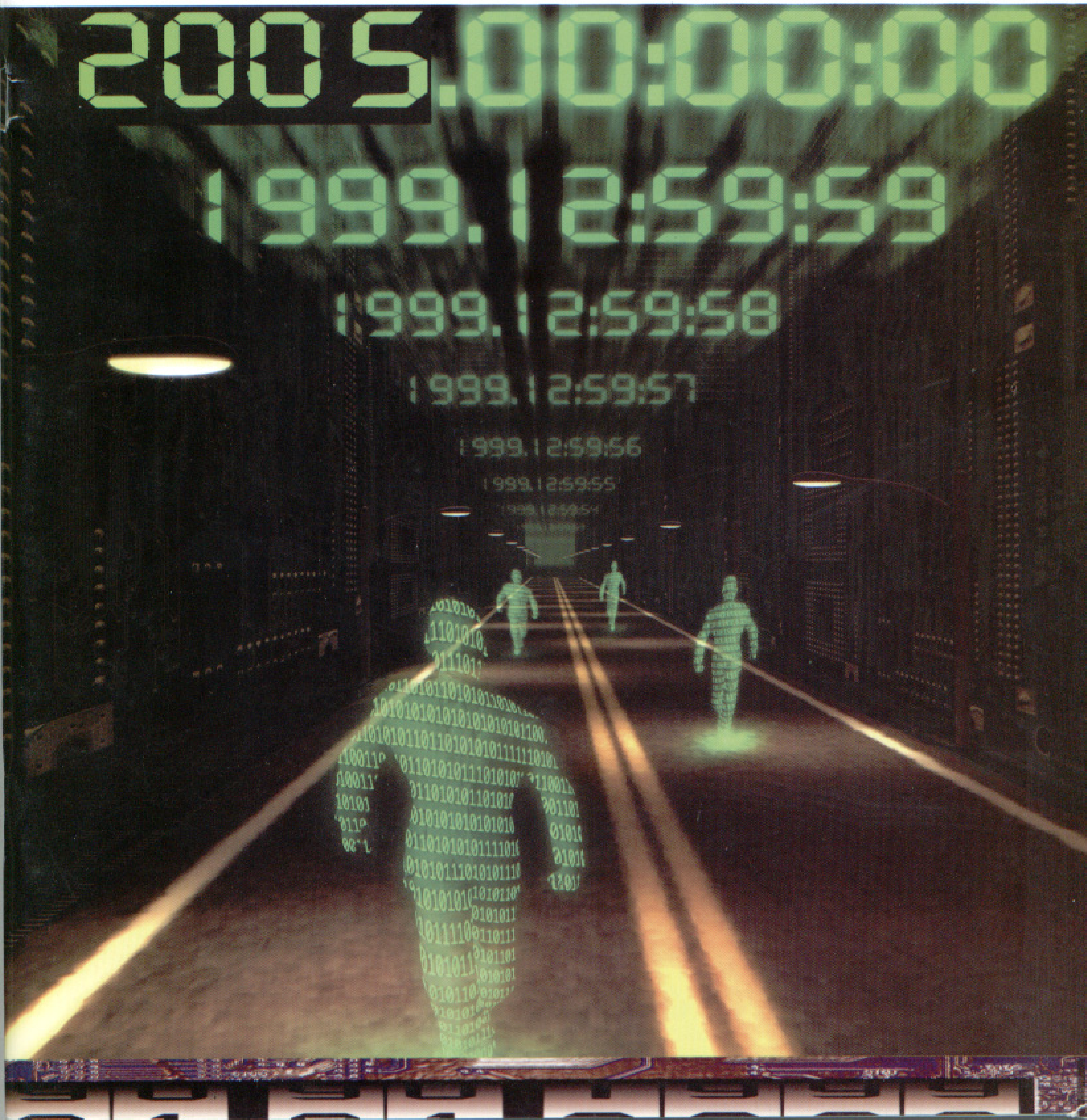

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Hybrid Fault Patterns for the Diagnosis of Gas Turbine Component Degradation

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1. Abstract

A degree of wearing out or scaling of the internal components (blading, seals, etc.) can be estimated analyzing the efficiency of the turbo compressor [1]. The wearing out or deposits on the blade or seal deterioration affects the efficiency of the machine which in turn compared to a reference data permits to identify faulty components or a group of components and a degree of their deterioration [2]. However a precision of this identification can be improved involving vibration analysis. It is important to make this identification on line thus facilitating constructing an appropriate plan for a major overhaul.

On the other hand, wearing out of the blades changes their natural frequencies and vibration spectrum; seals deterioration are caused by excessive rotor vibration or casing distortion mostly during start-ups. Analyzing simultaneously the efficiency and vibrations the faults and their locations can be identified more precisely.

An algorithm for identification of some faults of the gas turbine using hybrid patterns (efficiency & vibration) is developed.

2. Resumen (Modelos híbridos de fallas para el diagnóstico de la degradación de los componentes de una turbina de gas)

El grado de desgaste o incrustación en los componentes internos de un turbocompresor (álabes, sellos, etc.) puede ser estimado mediante el análisis de su eficiencia [1]. El desgaste, depósitos en los álabes o el deterioro de los sellos afectan la eficiencia de la máquina la cual comparando con las especificaciones permite identificar componentes defectuosos o a un grupo de ellos y un índice de su degradación [2]. Sin embargo puede lograrse una identificación más precisa cuando se involucra el análisis de su respuesta vibratoria. Es importante hacer dicha identificación en-línea ya que facilita la elaboración de un plan apropiado para el mantenimiento general.

Por otro lado, el desgaste de los álabes cambia su frecuencia natural y su espectro de vibración; el deterioro de los sellos es causado por la excesiva vibración del rotor o la distorsión de la cubierta, sobre todo durante los arranques. Analizando simultáneamente la eficiencia y la respuesta vibratoria se pueden identificar de una manera más precisa las averías y sus localizaciones.

Se desarrolla un algoritmo para la identificación de algunas averías en una turbina de gas usando patrones híbridos (eficiencia - respuesta vibratoria).

Nomenclature

- C Damping matrix.
- E Modal eccentricity.
- F Force vector.
- f Frequency.
- K Stiffness matrix.
- k Coordinate general.
- M Mass matrix.
- M_i Modal mass for the rotor i-th mode.
- n Rotor rotational speed.
- $U(x)$ Distribution of unbalance or residual unbalance of the rotor.
- X Displacement vector.
- $\overline{U}(x)$ Eccentricity of the bent shaft.
- x Displacement coordinate.
- Y Amplitude of vibrations.

z Number of blade in the row.

Greek Symbols

$\{\phi_j\}$ Displacements of the normalized modes.
$\phi_i(x)$ Modal shape coordinates for rotor i–th mode.
ω_i Critical speed for rotor i–th mode.
Ω Angular velocity of the rotor.
ξ_i Modal damping for rotor i–th mode.

Subscripts

i Matrix row denotation.
j Matrix column denotation.
k Matrix denotation.

3. Introduction

The most common faults, which affect the gas turbine seal and blading and rotor bearing system are:

- Increase in roughness of the blading surface, figure 1.
- Wearing out of the blades by solid particle erosion, corrosion and loss of material.
- Deposits on the blade surface.
- Wearing out of the labyrinth seals and tips of the moving blades.
- Severe vibrations caused by surge, rotating stall, oil whirl or oil whip or blade loss. This severe vibration can lead to a heavy rubbing in the seals or tip of the blades with casing, which in turn increases the clearance and in consequence diminishes the efficiency.
- Unbalance (balancing of the rotor not done very well), which during transitions, can cause rubbing in the blading system.

The loss of the blades or blade material affects the efficiency and increases vibrations as well. Of course, the rubbing of the blade tips decreases the blade length thus affects the efficiency of the compressor and/or gas turbine.

It is possible to locate the rubbing point along the turbo gas train placing an accelerometer in the various points of the casing to measure a spectrum of the vibration of the frequency, which indicates the rubbing. With a reasonable degree of precision a zone and therefore the stage(s) number can be indicated. Analyzing efficiency, its decrease will normally indicate a group of stages where the temperature and pressure are measured (before and after the stage group).

On the other hand, measuring the amplitude of vibration on the bearing or in the close vicinity, and having the rotor model for the modal unbalance response program it is possible to calculate amplitude of vibration at any point of the gas turbine rotor. In the points where the amplitude of vibration is greater than the seal clearance then the rubbing occurred. It should be confirmed by vibration analysis with a real time analyzer (spectrum) and the accelerometer either placed on the bearing or on the casing. The rubbing is characterized either by the low frequency components $2n$, $3n$, $4n$ or the high frequency components:

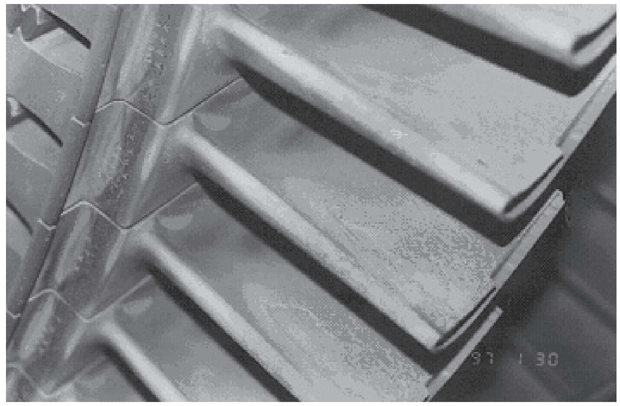

Fig. 1. Increase in roughness of the blading surface.

$$f = n \cdot z \tag{1}$$

If vibration is severe and rubbing occurs then the efficiency change which took place is identified by aero thermodynamic analysis of the gas turbine.

Resuming, it is seen that the efficiency change, either of compressor or turbine, or both, increases amplitude of vibration; this can be simulated by a modal unbalance response program [3] plus the spectrum of vibration obtained from a real time analyzer might constitute a hybrid fault pattern.

The following faults can be included in this category:

1. For compressor.

- Blade deposit.
- Blade material deterioration by solid particle erosion.
- Internal seal rubbing.
- Blade tip rubbing.
- Blade loss.

2. For gas turbine.

- Blade deposit
- Blade corrosion or damage by the solid particle erosion.

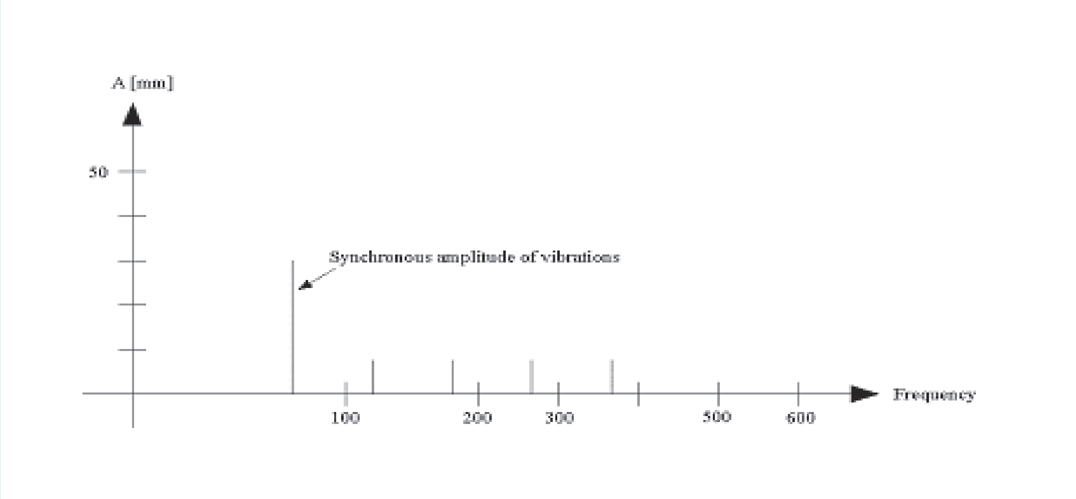

Fig. 2. The spectrum of the vibrations with pronounced harmonics.

- Blade loss.
- Internal seal rubbing.
- Blade tip rubbing.

Also, the identification of any fault is possible when they occur simultaneously in the compressor and the turbine as well.

The paper presents a part of the fault identification algorithm where the developed hybrid fault patterns are included.

4. Development

4.1 Development of the hybrid fault patterns

To establish the hybrid fault patterns of the gas turbine the following parameters should be measured:

- The flow mass of air and (burned) gas.
- The pressure and temperature of air before and after the compressor and, if possible, at other points.
- The pressure and temperature of the combustion gases before and after the turbine.
- Efficiencies.

The hybrid fault pattern is composed of the three following parts:

1. Decrease (change) in the thermodynamic parameters of the compressor and/or turbine like efficiency, compression ratio and the enthalpy drop.

2. Appearance of the vibration spectrum measured on the bearing and/or compressor-turbine casing of the vibration components corresponding to the frequencies $2n$, $3n$, $4n$…, with significant amplitude.

This indicates rubbing in the labyrinth seal system between the rotor and diaphragm. An example of the spectrum is given in figure 2. Also, when the amplitude of vibration at the frequency equal to the number of moving blades multiplied by a rotational speed has a measurable value; it indicates rubbing of the blade tips with a casing, according to the formula:

$$f = k \cdot n \cdot z \qquad (2)$$

Identifying the frequency of rubbing by analyzing the spectrum allows the stage number, either of the compressor or turbine, to be identified. If the moving blades have the cover bands, then the frequency of rubbing might be a strange number. This is also valid for the steam turbines. An example of a spectrum is given in figure 3.

3. Increase in amplitude of vibrations beyond the value of the clearances in the labyrinth seal systems.

Normally the amplitude of the rotor vibration is measured either at the bearings or in the closest vicinity of the bearing. By knowing a modal shape of the rotor vibration at the critical speeds or at any other speed, the amplitude of vibration can be calculated at any point; as it is presented in figure 4.

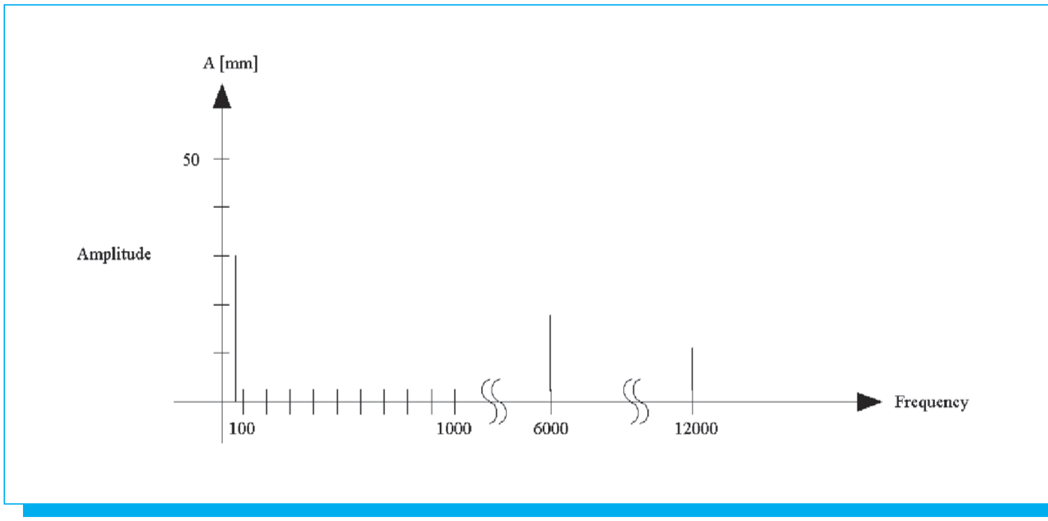

Fig. 3. The spectrum of the vibrations with pronounced harmonics.

As far as other speeds are concerned which are different than the critical speed an unbalance response program may be used. Then, having measured the vibration amplitude at one point of the rotor, for other point the amplitude of vibration can be calculated, as it is illustrated in figure 5.

For critical speed calculations a Velcri program is used [3], and a modal unbalance response program is given as in [4]

4.2 Modal method of vibration amplitude calculation at any speed

There are various methods and computer programs to calculate rotor unbalance response. In this work the modal method has been chosen. The method is based and solving the forced equation of motion in matrix form.

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=F$$

$$[M]\ddot{x}+[C]\dot{x}+[K]x=F \tag{3}$$

Assuming a synchronous motion the forcing function vector is expressed as:

$$\{F\}=\{F_0\}e^{i\omega t}$$

Or as an unbalance and operating speed

$$\{F\}=\{U\}w^2e^{i\omega t}$$

For the undamped homogenous system

$$[M]\ddot{\bar{x}}+[K]\bar{x}=0$$

$$[K]-\omega_n^2[M]\bar{x}=0$$

For each value of ω_i (natural frequency of the system for i mode), there is a corresponding mode shape $\bar{x}_i$ which when normalized, will be referred to as the normal mode ϕ_i and they are orthogonal to each other.

The amplitude vector x (Eq. 1) can be expanded in terms of the normalized undamped modes of the system.

$$\{x\}=\sum_{i=1}^{\infty}A_i\phi_i$$

For the first four modes of the system, Eq. 3 become

$$\{x\}=a_1\phi_1+a_2\phi_2+a_3\phi_3+a_4\phi_4 \tag{4}$$

Therefore the Eq. 1 can be transferred into

$$[M]\ddot{A}_k\phi_k+[C]\dot{A}_k\phi_k+[K]A_k\phi_k=\{U\}w^2e^{i\omega t} \tag{5}$$

where $$A_k\phi_k=\sum_{i=1}^{k}a_i\phi_i$$

Pre-multiplying the Eq. 5 by the transpose of the first mode $\{\phi_i\}$ we obtain:

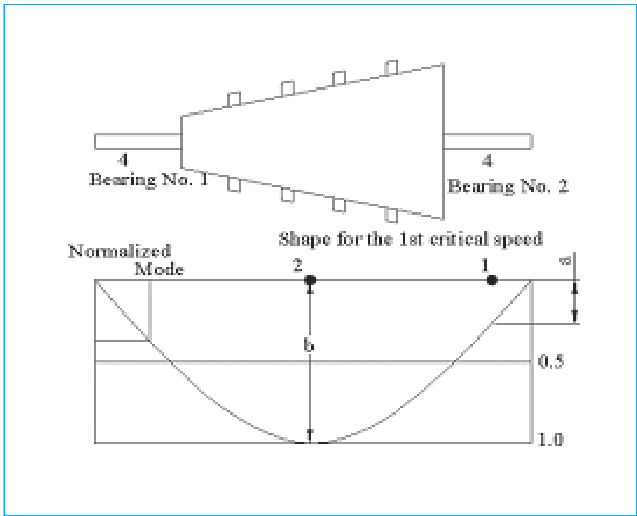

Fig. 4. Using critical speed program for determination of amplitude of vibrations at any point of the rotor. Measurement point 1", calculated point 2". Example: Vibrations measured at point *A* = 3 mils, Then the vibration at point 2" pointing the critical speed on *A2* = *An* = *b*/*a* = 3(10/0.1) = 30 mils.

$$\{\phi_1^T\}[M]\ddot{A}_k\{\phi_k\} + \{\phi_1^T\}[C]\dot{A}_k\{\phi_k\} + \{\phi_1^T\}[K]A_k\{\phi_k\} = \{\phi_1^T\}\{U\}\omega^2 e^{i\omega t} \tag{6}$$

Using orthogonal property of the normalized mode shapes, we have:

$$m_1\ddot{A}_k + \{\phi_1^T\}[C]\{\phi_k\}\dot{A}_k + \{\phi_1^T\}[K]A_1\{\phi_1\} = \{\bar{U}_1\}\omega^2 e^{i\omega t} \tag{7}$$

Where m_1 is the modal mass of the first mode,

$$m_1 = \{\phi_1^T\}[M]\{\phi_1\}$$

and from Eq. 2

$$m_1\omega^2 = \{\phi_1^T\}[K]\{\phi_1\}$$

Therefore Eq. 7 becomes

$$m_1\ddot{A}_1 + C_{11}\dot{A}_1 + C_{12}\dot{A}_2 + C_{13}\dot{A}_3 + C_{14}\dot{A}_4 + m_1\omega^2 A_1 = \bar{U}_1\omega^2 e^{i\omega t} \tag{8}$$

where

$$a_{ij} = \{\phi_1^T\}[C]\{\phi_j\}$$

$$\bar{U}_i = \{\phi_1^T\}[\bar{U}]$$

or

$$\ddot{A}_1 + C_{11}\dot{A}_1 + C_{12}\dot{A}_2 + C_{13}\dot{A}_3 + C_{14}\dot{A}_4\ \omega_1^2 A_1 = E_1\omega^2 e^{i\omega t} \tag{9}$$

where

$$\bar{C}_{ij} = \frac{C_{ij}}{m_i}$$

$$E_{ij} = \frac{\bar{U}_{ij}}{m_i} \quad \text{Modal unbalance eccentrinty}$$

Assuming $A = \bar{A}e^{i\omega t}$ the equation 9 is changed into:

$$\omega^2\bar{A}_2 + i\omega(\bar{C}_{21}\bar{A}_1 + \bar{C}_{22}\bar{A}_2 + \bar{C}_{23}\bar{A}_3 + \bar{C}_{24}\bar{A}_4) + \omega_2^2\bar{A}_2 = E_2\omega^2 \tag{10}$$

In the same way the equation (5) is multiplied by (ϕ'_2), (ϕ'_3), and (ϕ'_4) becoming:

$$\omega^2\bar{A}_2 + i\omega(\bar{C}_{21}\bar{A}_1 + \bar{C}_{22}\bar{A}_2 + \bar{C}_{23}\bar{A}_3 + \bar{C}_{24}\bar{A}_4) + \omega_2^2\bar{A}_2 = E_2\omega^2 \tag{11}$$

$$\omega^2\bar{A}_3 + i\omega(\bar{C}_{31}\bar{A}_1 + \bar{C}_{32}\bar{A}_2 + \bar{C}_{33}\bar{A}_3 + \bar{C}_{34}\bar{A}_4) + \omega_3^2\bar{A}_3 = E_3\omega^2 \tag{12}$$

$$\omega^2\bar{A}_4 + i\omega(\bar{C}_{41}\bar{A}_1 + \bar{C}_{42}\bar{A}_2 + \bar{C}_{43}\bar{A}_3 + \bar{C}_{44}\bar{A}_4) + \omega_4^2\bar{A}_4 = E_4\omega^2 \tag{13}$$

The set of 4 equations can be put into matrix form and can be solved for the complex coefficients a_i. Then, the response amplitude can be computed using equation 3 the amplitude of vibration and phase angle can be computed for each solution. Based on this algorithm a computer program has been coded which is used for a calculation of the vibration amplitude for the selected rotor station at various speed.

For all four modes the matrix of four simultaneous equations can be solved and the vibration amplitude at any joint of the rotor (rotor stators) can thus be calculated.

Using this approach the modal unbalance response program has been developed [4] which is used for this type of diagnostics.

The input data to the program consist of:

- Rotor section diameters.
- Rotor section length.
- Modal damping coefficient.
- Eccentricity (unbalance).
- Young modulus.

The outputs are given as imaginary and real parts of:

- Speed of the rotor.
- Amplitude of vibration at any pre-selected point.
- Phase angle.

Also, a simplified method can be utilized [5] for the calculation of the vibration amplitude at any point and any speed of the gas turbine.

The amplitude of vibration is given by the following formula:

$$Y(X,\Omega)=\sum_{i=1}^{\infty}\frac{\int_0^l\left(u(x)\phi_i(x)\Omega^2+\bar{U}(x)\phi_i(x)\omega_i^2\right)+dx}{\left(\omega_i^2\cdot\Omega^2+2i\xi_i\omega_i\Omega\right)M_i} \quad (14)$$

Modal shape coordinates and the critical speed for each considered mode can be calculated using a critical speed program [3].

The modal damping ξ_i for each mode can be assumed from experience, and the modal mass can be calculated by formula (10).

Before applying the formula for a diagnostic case the unbalance $U(x)$ or/and $\bar{U}(x)$ should be estimated.

Knowing the amplitude of vibration at the closest vicinity of the bearing a measure with any suitable vibration analysis instrumentation with displacement sensors, an unbalance can be calculated, estimated or assumed; the vibration amplitude of the measured bearing will be equal to the calculated one. The eccentricity of the rotor $U(x)$ can be measured during an overhaul or when the turbine is operated in turning gear. Then the amplitude of the vibration can be calculated at any point of the rotor and any speed, while the point with maximum amplitude of vibration is selected for a diagnostic purpose.

The value of the amplitude of vibration is compared to the seal clearance taken down during an overhaul. If the value of the vibration amplitude is greater than the seal clearance, then rubbing occurred and the most probable localization of rubbing is indicated. It also can be confirmed by analysis of the bearing spectrum vibration.

In the case where rubbing appears the vibration components of either $2n$, $3n$, $4n$, … or higher frequencies will appear and are given by equation (1).

Using this approach, the amplitude of vibration can be estimated and, jointly with the measured ones at the bearing, the following faults (for example) can be identified:

- Deposits.
- Heavy unbalance material.
- Blade or blade covers internal loss.
- Blade loss.
- Temporary thermally bent shaft.

4.3 Application of hybrid faults patterns.

The hybrid fault patterns can represent the wearing out of the internal seals (between the shaft and diaphragm) either of compressor or the gas turbine or both. The wearing out of the tip of the moving blades either of the compressor or turbine can be estimated in the case when the rubbing is caused by the shaft vibration or bent shaft; also when the rubbing is caused by the casing distortion.

In the last case only two fault components are important for the diagnostic: efficiency decrease, and appearance of considerable high frequency vibration components (amplitude). Some of the hybrid patterns are presented below

4.3.1 Fault patterns of the wearing out of the compressor and turbine moving blades tips

Normally this kind of fault occurs in a short period of time and is characterized (during start up or load changes) by an increase in vibration (temporal, permanent) or the appearance of the new amplitude components (peaks) in the vibration spectrum measured at the bearing such as:

$$f=n\cdot z\cdot k \quad (15)$$

If the amplitude of the vibration at this frequency f is considerable, then the row with z blade numbers is indicated as been potentially affected by rubbing.

- Simultaneously, the pressure ratio p_2/p_1, the compressor efficiency and gas turbine output are decreased.
- In the case when only the gas turbine is affected, the efficiency of the compressor is constant and the gas turbine efficiency and its output are decreased.

In both cases the amplitude of the vibration of the shaft is calculated using formula (14).

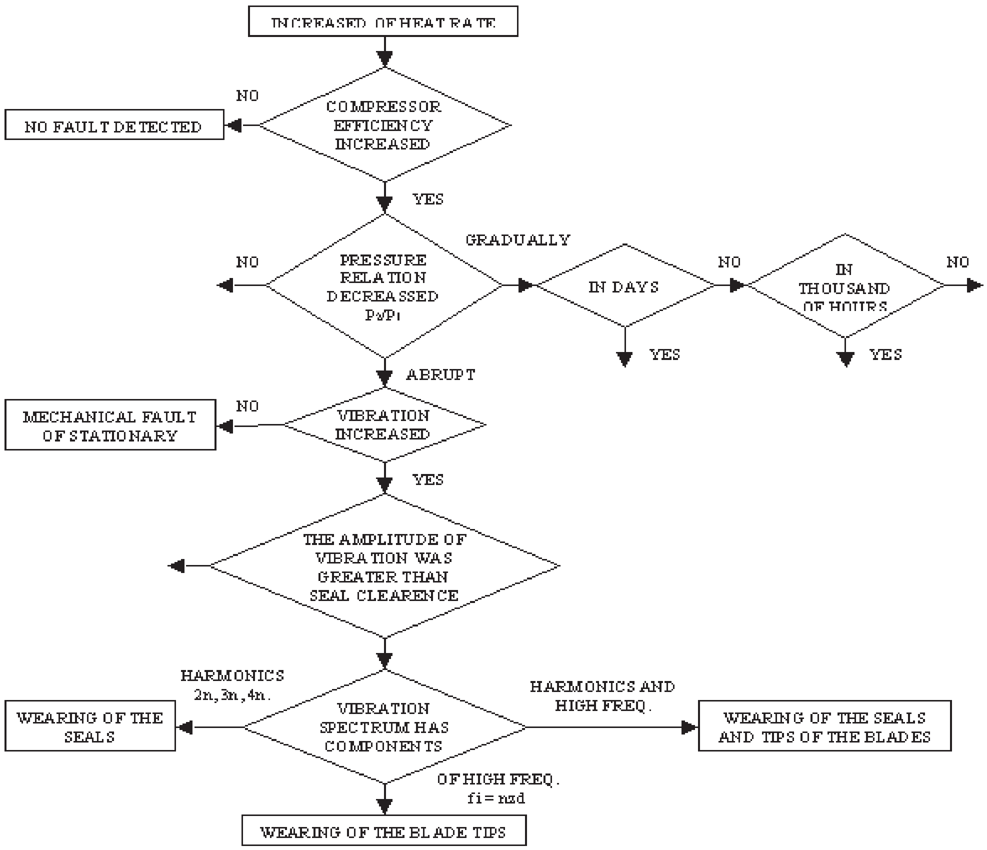

Fig. 5. Diagram of the turbocompresor faults identifications algorithm.

4.3.2 Hybrid faults patterns of seals wear out (caused by rotor vibration)

During start up or load changes the rotor can temporally be bent, increasing its vibration. Using the formula [4]:

$$Y[X,\Omega]=\sum_{i=1}^{\infty}\frac{\int_0^l\left(U(x)\phi_i\Omega^2+\bar{U}(\alpha)\phi_i(x)\cdot\omega^2\right)dx}{\left(\omega_i^2\Omega^2+2\xi_i\omega_i\Omega\right)M_i} \qquad (16)$$

The maximum amplitude of the vibration of the rotor (shaft) and its localization along the rotor can be determined.

If $Y(X,\Omega)>C_{seals}$ then, rubbing occurred and the significant vibration components at $2n$, $3n$, $4n$ frequencies appear.

Simultaneously, the efficiency either at the compressor (p_2/p_1 and M_{aire} decreases) or the gas turbine decreases.

If the increase in the vibration is caused by rotor faults, firstly the increase in synchronous amplitude of vibration is observed, then other the contact with stationary parts or any of the harmonic $2n$, $3n$, $4n$ components or higher appear. If casing distortion occurs, firstly the harmonics and high frequency components increase followed by the synchronous amplitude of vibration.

4.4 Implementation of the hybrid fault pattern into the gas turbine fault identification algorithm

A gas turbine fault identification algorithm has been developed [6] applying the hybrid fault pattern, but due to a space limitation, it can not be presented in this paper. Briefly speaking, the algorithm identifies the faults, which affect the compressor, gas turbines or both. Many faults can be identified. A small part of the algorithm is present in figure 5. The figure is self-explanatory. Simple logical rules, «YES» and «NO» are used. In the fault identification part the wearing of the seals, blade tips and mechanical faults, which affect the compressor, are illustrated.

Based on the algorithm, an Expert System TURDIAG has been developed. Thirty-nine quantifiers, forty-nine choices and forty-two rules were used to construct the diagnostic expert system.

To detect some of the faults a probability factor ≤ 1 has been applied.

As an example, the history of the rule number 6 is illustrated.

Rule 6: [Increase in heat rate]

IF:
Gas flow is normal and combustion chamber works smoothly and (pressure) compression ratio p_2/p_1 and efficiency of the compressor is decreased and compression rate decreased abruptly and event occurred after the vibration increased and the amplitude of the vibration was greater than the seal clearance and the vibration spectrum has significant components (amplitude) at $2n$, $3n$, $4n$ frequencies

THEN:
Wearing out of the seals of the compressor occurs with probability = 10/10.

Example: (Hybrid fault):

5. Conclusions

The hybrid faults patterns developed here can improve the precision of the diagnosis of gas turbine engines as well as steam turbines.

In the case rubbing its location can be indicated.

Further Work

Further on some numerical examples and specific diagnostic cases will be analyzed to support the method developed here.

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