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Shaft sensor based on modeling diagnostic signs of power unit defects

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Abstract

A perfection concept is proposed for automated control diagnostics systems used at power plants with application of shaft sensors. It is proposed a systematic approach to solve a number of practical problems related with safety and performance of power generating turbosets. Diagnostics of multi-rotor turboset technical condition are significantly improved due to "complete" set of shaft motion sensors, applied at both sides of each journal bearing. There have been given diagnostics signs of defects measured with shaft sensors and suggested some extra criteria of reliability of the shafting and supports.
To illustrate the method let us compute correcting alignments of fragment of shafting.
Using simplified 2-rotor 4-support turboset, it is shown that: corrective alignments of supports, recommended by manufacturers of turboset and generator, may be not sufficient to compensate misalignments of supports, caused by thermal deformations of turboset support system. Those corrective misalignments should be computed individually for each turboset, using real thermal deformations and real clearances of journal bearings.

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Keywords: automated control diagnostics systems, shaft sensors, journal bearings, turboset, shaft, supports, defects, misalignments, corrective alignments of supports, criteria of reliability.

1. Introduction

Automated control systems for safe money-saving real-time operation and maintenance (COMPACS®) [1,2] are used in many industries where such systems can work based on different sensors including shaft displacement ones. Shaft displacement which causes basic disturbance is sensitive to a number of defects like rotor half-coupling

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assembly defects, defects of rotor centering in the turbine setting, support malfunction, etc. At domestic thermal power-plants, where at least one pair of shaft sensors is installed in bearings on turbine units [3], the vibration experts cannot see the turbine unite adjustment without important initial information about rotor movement. However, these sensors information is processed by the existing systems within minimum scope, only for the purpose of vibration problem solution. These tasks are covered by the new GOST [4] instead of the old GOST 27165-1998. But the key information contained in the static characteristics read with shaft sensors (levitation curve, minimum clearance, etc.) and in other proper characteristics (power friction loss, misalignment, rigidity and oil film damping, etc.) is not used and not analyzed. Also, the new GOST [4] is supported with no guidance papers for defect diagnostics and for use of characteristics dependable on the bearing design and operation conditions including very important parameters of bearing capacity and friction loss.

Some foreign companies use levitation curves for diagnostics of different plants and units, but they cannot go beyond the problems covered by ISO 7917-2:2009. According to this ISO, shaft displacement sensors are installed on one side only or on the bearing transverse axis. However, it is impossible to evaluate correct technical condition of the support because of rotor journal misalignment caused by a number of factors.

It is very difficult and laboriously to discover defects using the current method support and software provided by tens of companies, so far as the main problem, defect search automaton and development of criteria for evaluation of residual defects risk or acceptability, has not been solved. As opposed to other ones the COMPACS® system can make complete diagnostics in automatic mode [1].

In such a way, the following new problems now arise:

- adaptation of the COMPACS® systems for power units with journal bearings;
- development of methodological support and diagnostic power unit failure indicators based on shaft sensors, as well as auxiliary diagnostic problems solutions;
- check and development of diagnostic signs by means of physical and mathematical modeling;
- perfection of vibration monitoring specifications and diagnostic systems for different turbine units.

This work solves in part the first two problems resulting in exemplary modeling of support misalignment and determination of correcting alignment. In [7,8] for the COMPACS® systems there are prospects of shaft sensors in full set considered for the purpose of unit rotor balancing and centering perfection. The full set of shaft sensors means installation of two sensor pairs on both sides of the rotor journal in the bearing according to [4].

2. Diagnostic signs of defects based on the shaft sensors complete system

Measurement of shaft displacement based on the complete set of sensors allows specifying the following defect diagnostic signs (see Table 1). The table also includes extra criteria of reliability. The criteria and defect diagnostic signs given in Table 1 may be grounded by the current document in part only. The rest should be analyzed and perfected, if necessary, in respect to an individual unit during the system adaptation. *Alongside with the suggested criteria, the turbine unit reliability criteria must be performed according to GOST R 55263-2012*[4].

Realization of the Table 1 criteria involves the following problems solutions in the course of monitoring ONLINE:

- calculation of minimal journal misalignment and gaps, i.e. the modes with inadmissible gaps and loss in bearing capacity;
- connection of minimal gaps and misalignment with babbit temperature for all loading conditions and rotary speeds;
- calculation of gap correlation and journal misalignment based on the data of housing absolute extension;
- calculation of static and dynamic loads in supports;

Table 1. List of defects measured or diagnosed with shaft sensors.

No	Defects	Diagnostics indicators	Remarks
	a) static defects		

1.	minimum static clearance less than allowable	$\delta \leq [\delta]$;	Los of carrying capacity, babbit contact threat. $[\delta]$ is calculated for every individual bearing.
2.	Journal skewing	To be developed	Support uneven loading. It may be caused by bearing incorrect alignment or by support bite at restricted spread.
3.	Static load deviation	$\Delta Q \geq 0.4*Q$	Static load deviation over 40 %
4.	Babbit contact	Minimum negative clearance. $T_{bab} \geq [T_{bab}]$;	$[T_{bab}]$ – specified by the manufacturer. Signal spectrum scattering.
5.	Support misalignment	End or radial misalignment overriding	Allowed deviation of alignment is specified for every individual half-coupling.
6.	Jump of a journal relative position in the bearing boring	Irreversible change of a relative position of the journal and boring	Specified for every individual support
7.	Packing rubbing	Minimum clearance in packing в is assessed less than allowable one	
8.	Exceeded fluctuating stress in rotor journals, weld joints and half-coupling bolts caused by misalignment	$\sigma \geq [\sigma]$	$[\sigma]$ = 20 MPa
	b) dynamic defects		
9	Exceeded dynamic load	$Qd \geq [Qd]$	Babbit destruction probability.
10	Residualunbalanceexceedsallowed	$\varepsilon \geq [\varepsilon]$	Specified for every individual for each support depending on it flexibility [7].
11	Minimum dynamic clearance less than allowable	$\delta_d \leq [\delta_d]$;	Babbit contact threat. $[\delta_Д]$ is specified for every individual bearing.
12	Insufficient resonance tune-out at running speed	Critical frequency tune-off less than 10 %	Danger of high vibration at rotational frequency
13	Insufficient resonance tune-out at doubled speed	Critical frequency tune-off less than 10 %	Danger of vibration at doubled rotational frequency
14	Residual deflection	Rotor beat inside the cylinder at slow speed exceeds allowed by 0.05mm	Heightened vibration and packing damage at critical frequency.
15	Crack	Complex criterion of spectral characteristic changing	
16	“Elbow”	Rotor beat from the side of half-coupling at law speed exceeds allowed and has anti-phase nature for adjacent supports	
17	“Pendulum”	Rotor beat from the side of half-coupling at low speed exceeds allowed and has in-phase nature for adjacent supports.	
18	Heightened dynamic flexibility of support	High vibration of support at small perturbation	Dynamic flexibility of supports are to be studied on prototype models and to be controlled.
19	Support resonance	Independent position of support resonance at absence of shafting resonance	Support resonance must be removed.
20	Rotor (rotor system) resonance in shafting	Rotor resonance in shafting exists	Rotor resonance is a design defect which can be removed in part with incorrect rotor half-coupling alignment

			and bearing clearance changing.
21	Exceeding fluctuating stress in journals, weld joints and half-coupling bolts	$\sigma \geq [\sigma]$	$[\sigma]$ = 20 MPa
	s) spectral characteristic		
22	Subharmonic resonance	Frequency is exactly ½ ; 1/3; ¼, etc., of operating frequency $V_{SH} \geq$ 0,1 mm/s	
23	Low-frequency vibration at XX	$V_{LFV} \geq$ 0,1 mm/s	
24	Low-frequency vibration under load	$V_{LFV} \geq$ 0,1 mm/s	

- determination of rotor journal relative position jumps in bearing boring;
- identification and prevention of babbit/unit gland rubbing;
- correlation of journals dynamic misalignment and rotor vibration determination;
- evaluation of the shafting operation condition according to the results of shaft oscillation under the requirements [4];
- analyzing the shaft and support amplitude-phase-frequency characteristics for different load systems of startup or rundown, and information accumulation for more effective automated calculation of disbalance and correcting load systems;
- evaluation of rotors residual disbalance;
- identification of support resonance (stator-foundation) and rotor resonance in shafting;
- calculation of trajectories, spectral and other characteristics of shaft movement;
- evaluation of rotor beating is connected to its residual bowing or to imperfection of the rotor half-couplings assembly;
- calculation of stress in rotor journals, weld joints or in half-coupling bolts caused by misalignment of supports relative position;
- calculation of stress in the specified cross-sections and their time-frequency dependence caused by elevated vibration of supports cased by defects [7,8];
- evaluation of end-of-life of rotor half-coupling bolts according to the current standards;

In the long view, problems of torsion oscillation and thermomechanical fatigue of high-temperature rotors will be solved.

All the above problems are impossible to be solved manually, we can solve them in automatic mode only. Imported diagnostics systems, due to limited number of shaft sensors, allow solving only few of the mentioned problems.

Fig. 1 shows a simplified block diagram of the operation condition monitoring system which covers the biggest problems to be solved. A part of the methodological support applied in this work is presented in works [7-11] and based on experience in methodological support and software for design calculation of static and dynamic characteristics of turbine unit shafting. As against problems assigned before [11], when developing methodological support and software we offered new improved approaches to calculation of static and dynamic characteristics of oil film [10,12], which allow considering the latest achievements in this field. Let us emphasize that it is for the first time when some defects can be measured directly based on the complete system of sensors.

3. Samples of alignments defects modeling for 4-support system of rotors

To illustrate the method let us compute correcting alignments of fragment of shafting.

The task is the following. All constructional properties of shafting are known, including geometrical and inertial parameters as well as stiffness for rotors, journal bearings and coupling halves [9,10].

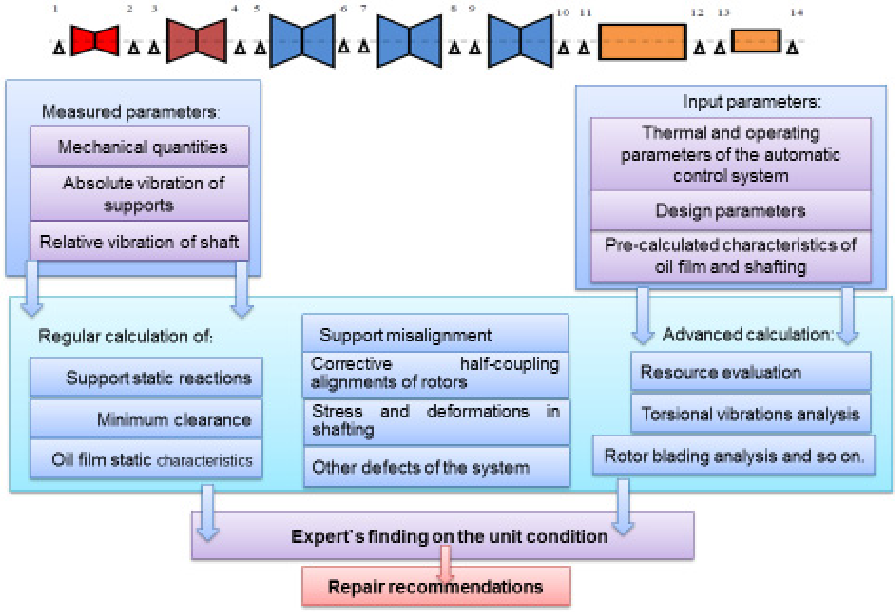

Fig. 1. Simplified block diagram of the COMPACS-T® control systemfor operating conditions.

For each support of unit # 1 and #2 it is known average thermal displacement of support unit due to heating while operation (see item 5 of Table 2, 3). Those displacements are measured by Ltd.«SibAtomGeodezia». It is known that manufacturer of turboset recommends to raise support units of exciter by 0.3 mm with respect to support units of the generator. The task is to find reaction forces at support units during operation mode, as well as to compute corrective alignments of supports, aimed at correction of reaction forces towards nominal values.

To demonstrate the power of technique proposed, we'll compute corrective alignments for 2-rotor fragment of turbosets shaft. Let's subdivide "electric generator rotor – exciter rotor" subsystem. There are overall input and output data for unit # 1 and unit # 2 listed in Table 2 and 3.In both tablenormalized displacements of supports are used, i.e. supports elevations are measured relative to line, connecting first and last supports (# 11 and # 14) of fragment.

It can be seen, that corrective displacements recommended by manufacturer are not sufficient to compensate displacements of supports, caused by heating as well as skew ascent of shaft pin in bearings. In hot statethere are significant deviations of forces in supports. Particularly, support #13 is overloaded by 55.4% for turboset #1and underloaded by 77.4 % for turboset #2.

This example shows capabilities of method to compute optimal corrective displacements. Besides, the example shows that in many cases manufacturers don't take into account heat deformations of foundations and supports of turboset, as well as skew ascent of shaft pins.

Table 2 Computation of shafting static characteristics for generator rotor and electric exciter rotor for turboset #1

Step #	Characteristics	units	supports #11	#12	#13	#14
1	Nominal forces at supports (at nominal locations of supports)	kN	813.1	751.9	118.6	111.1
2	Nominal elevation of geometric centers of journal bearings	mm	0	-1.603	-1.812	0
3	Corrective displacements recommended by manufacturer, before normalization	mm	0	0	0.3	0.3
4	Corrective displacements recommended by manufacturer, after normalization	mm	0.000	-0.189	0.084	0.000
5	cold-to-hot vertical displacements of supports, before normalization	mm	2.196	1.625	1.311	0
6	cold-to-hot vertical displacements of supports, after normalization	mm	0	0.812	0.697	0
7	Overall vertical displacement of supports (item 4 + item 6)	mm	0	0.623	0.781	0
8	Vertical reaction forces at final location of supports	kN	815.5	695.7	184.2	99.1
9	Difference between final vertical reaction forces and nominal values	kN	2.42	-56.16	65.67	-11.93
10	Relative difference between final vertical reaction forces and nominal values	%	0.3%	-7.5%	55.4%	-10.7%
11	Corrective displacements of supports, computed using linear model	mm	0	-0.622	-0.792	0
12	Relative error of corrective displacements (item 11 + item 7)/(item 7)	%	0%	0.1%	-1.4%	0%
13	Vertical reaction forces at final location of supports after corrective displacement from item 11	kN	812.5	757.5	113.1	111.5
14	Relative difference between vertical reaction forces and nominal values (item 13 – item 1)/(item 1)	%	-0.07%	0.74%	-4.59%	0.37%

Table 3. Computation of static characteristic of shafting for generator rotor and electric exciter rotor for turboset #2.

Step #	Characteristics	units	supports # 11	# 12	# 13	# 14
1	Nominal forces at supports (at nominal locations of supports)	kN	813.1	751.9	118.6	111.1
2	Nominal elevation of geometric centers of journal bearings	mm	0	-1.603	-1.812	0
3	Corrective displacements recommended by manufacturer, before normalization	mm	0	0	0.3	0.3
4	Corrective displacements recommended by manufacturer, after normalization	mm	0.000	-0.189	0.084	0.000
5	cold-to-hot vertical displacements of supports, before normalization	mm	1.587	1.206	0.529	0.000
6	cold-to-hot vertical displacements of supports, after normalization	mm	0.000	0.619	0.086	0.000
7	Overall vertical displacement of supports (item 4 + item 6)	mm	0.000	0.430	0.169	0.000
8	Vertical reaction forces at final location of supports	kN	801.3	853.0	26.8	113.5
9	Difference between final vertical reaction forces and nominal values	kN	-11.75	101.10	-91.79	2.43

10	Relative difference between final vertical reaction forces and nominal values	%	-1.4%	13.4%	- 77.4%	2.2%
11	Corrective displacements of supports, computed using linear model, iteration 1	mm	0.000	-0.428	-0.215	0.000
12	Relative error of corrective displacements (item 11 + item 7)/(item 7) , iteration 1	%	0.0%	0.6%	- 27.1%	0.0%
13	Vertical reaction forces at final location of supports after corrective displacement from item 11, iteration 1	kN	810.9	774.3	96.6	112.7
14	Relative difference between vertical reaction forces and nominal values (item 13 – item 1)/(item 1) , iteration 1	%	-0.3%	3.0%	- 18.5%	1.5%
15	Corrective displacements of supports, computed using linear model, iteration 1	мм	0.000	-0.430	-0.165	0.000
16	Relative error of corrective displacements (item 15 + item 7)/(item 7) , iteration 1	%	0.0%	0.0%	2.4%	0.0%
17	Vertical reaction forces at final location of supports after corrective displacement from item 11, iteration 1	kN	813.3	749.9	120.5	110.9
18	Relative difference between vertical reaction forces and nominal values (item 17 – item 1)/(item 1) , iteration 1	%	0.0%	-0.3%	1.6%	-0.1%

To determine corrective displacementsof supports, it is proposed to use the matrix of influence. For the sake of simplicity, in this work wetake into account influence in vertical plane only of generator and electric exciter rotors (See Table 4)

Table 4. Matrix A of influence of support displacements on reaction forces, kN/mm.

	dy1	dy2	dy3	dy4
ry1	6.73	-47.25	40.12	0.64
ry2	-51.74	443.27	-418.68	24.69
ry3	44.39	-417.66	410.58	-34.95
ry4	0.62	21.63	-32.03	9.62

Change of reaction forces due to vertical displacements of supports reads:

$$\Delta R = A \cdot \Delta y$$

In this case, there are 4 vertical displacements and matrix of influence with size 4x4. If all supports of turboset are equallyshifted in vertical direction, the reaction forces are not changed. Also, if the whole shaft of turboset is rotated in vertical plane on small angle, the reaction forces are not changed. That's why matrix A has rank N-2, where N is number of supports. In the current case N=4.Let's fix 1st and last supports at zero, and find displacements y_2, y_3.

Substitute $y_1 = 0,\ y_4 = 0$ in the set of linear equations and get over-determined equation set with respect to $\Delta y_2, \Delta y_3$.

To solve the system, we use ΔR from row 9at tables 2 and 3

$$\Delta R_1 = \begin{bmatrix} 2.42 \\ -56.16 \\ 65.67 \\ -11.93 \end{bmatrix}_u \quad \Delta R = \begin{bmatrix} -11.75 \\ 101.10 \\ -91.79 \\ 2.43 \end{bmatrix}$$

Forsupports of turbosets #1 and #2 respecively.

Since the rank of the matrix A is equal to two, the unknown x, y are uniquely determined.

In line 11 in tables #2 and #3 there are corrective displacements computed using linear model for turbosets #1 and #2 respectively.By using the calculated correction displacement for unit # 1, thermal misalignment are compensated almost completely, and reaction forces at supports gets close to nominal ones(see row 12).

When loads significantly deviate from nominal values, that is the case for support # 13 at turboset #2, it can be necessary to perform second iteration to find corrective alignments with required accuracy. In such cases, linear model is not sufficiently accurate, because response of journal bearing to external load may be highly nonlinear.

To perform second iteration, it is necessary to apply corrective alignment computed at first iteration, then measure the difference between current reaction forces and nominal values, and use those values as new ΔR vector with the same sensitivity matrix.Input and output data for second iteration is listed in rows 17 and 18 of Table 3.

The technique proposed can also be used to optimize corrective alignments of coupling halves of adjacent rotors. Online measurement of shaft displacements should be done using averaging to compensate shaft periodic motion effects. It can be used later in automated control diagnostics systems to calculate corrective alignments of coupling halves on use "complete" set of shaft motion sensors.

To compute forces at bearings, it is necessary to know location of shaft pin in clearance, and it is necessary to numerically solve lubrication problem for the journal bearing. In our previous work, it is proposed to use pre-computed results for lubrication problem and interpolation technique to get forces within journal bearing [12].

4. Conclusion

1. A perfection concept is proposed for automated control diagnostics systems used at power plants with application of shaft sensors. It is proposed a systematic approach to solve a number of practical problems related with safety and performance of power generating turbosets. Diagnostics of multi-rotor turboset technical condition are significantly improved due to "complete" set of shaft motion sensors, applied at both sides of each journal bearing.
2. There have been given diagnostics signs of defects measured with shaft sensors.
3. It is suggested extra criteria of reliability of the turbine unit shafting and supports.
4. It is proposed flowchart of system COMPACS-T designed for technical monitoring of power generating turboset.
5. Using simplified 2-rotor 4-support turboset, it is shown that:

5.1. Corrective alignments of supports, recommended by manufacturers of turboset and generator, may be not sufficient to compensate misalignments of supports, caused by thermal deformations of turboset support system. Those corrective misalignments should be computed individually for each turboset, using real thermal deformations and real clearances of journal bearings.

5.2. It is proposed a technique for assessment of misalignments of supports based on online measurement of shaft displacements. Using mathematic model of journal bearings it is possible to compute reactions forces in each support, in the case when displacements of shaft pins are known.

6. The technique proposed can be utilized in systems of monitoring of technical state of turbosets as well as for optimal fit of corrective alignments of supports.

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