Development and Testing of the Code and Procedure for Balancing the Flexible Vertical Electromagnetically Suspended Rotor

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Annotation

It is known that the basic component of forces, which affects electromagnetic bearings (EMB), is centrifugal forces produced by the unbalanced rotor. The unbalanced forces must be reduced to the allowed level for the entire frequency band, including critical frequencies (GOST ISO 11342). For effective balancing, balancing weights shall be installed in accordance with actual rotor bending modes. Therefore, under research and development work carried out in JSC "Afrikantov OKBM" (Russia) activities are performed aimed at studying possibilities for balancing the flexible vertical rotor using EMBs in the machine.

During this work, the special BALANS code and procedure for balancing the flexible rotor using EMBs were developed. At a small test facility having the vertical rotor [1, 2], experimental investigations were carried out to balance the flexible rotor using EMBs. Based on preliminary results, it is possible to balance the flexible vertical rotor using EMB without balancing the rotor in factory conditions. In 2011, JSC "Afrikantov OKBM" carried out work to balance the vertical rotor using electromagnetic suspension in the large test facility, i.e. in the Turbomachine (TM) Rotor Scale Model (RSM).

1 Introduction

The key component of the power unit having the High-Temperature Gas Cooled Reactor and direct gas turbine cycle is the TM with the EMBs (Fig. 1) [1]. The power unit is the combination of the helium modular reactor, which is capable to generate high-temperature heat, and the power conversion system.

The vertical rotors of the generator and turbocompressor pass two bending critical frequencies at startup and shutdown. Significant bending during passing of critical speeds can be eliminated and, consequently, the flexible rotor can be qualitatively balanced only by the system of weights installed in such a way that the law of distribution of discrete correction counterweights along the rotor is maximally similar to the law of distribution of initial unbalances and compensates for them at satisfactory accuracy.

Moreover, qualitative balancing will make it possible to reduce loads on electromagnetic and catcher bearings (CB) and decrease the vibration level in the TM, especially at passing of critical

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frequencies. Recommendations of the standrad GOST ISO 11342 "Mechanical vibration methods and criteria for the mechanical balancing of flexible rotors" can hardly be fulfilled for real asymmetrical rotors having variable rigidity. In addition, this standard does not consider issues on quality estimation of rotor balancing using EMBs. Thus, JSC "Afrikantov OKBM" together with the Research Institute of Mechanics, Nizhny Novgorod State University developed the special BALANS code and procedure for flexible rotor balancing using EMBs. The code is intended to define the initial rotor unbalance and balance the flexible vertical rotor in the TM using information received from the EMB control system.

2 Task Statement. Quality Criteria

In order to validate the RP TM design having the high-temperature gas cooled reactor, JSC "Afrikantov OKBM" tests the flexible jointed rotor using full electromagnetic suspension in the TM RSM test facility. One of the tasks resolved in the course of testing is investigation of possibilities for balancing the rotor in the TM, i.e. on its own EMBs. Although the rotor can be satisfactorily balanced in the low-speed or high-speed balancing machine, the current experience shows that after the rotor has been installed in the machine additional balancing can be required. This is caused by the following factors:

- different dynamic characteristics of supports of the TM rotor and balancing machine, as EMB characteristics are assuredly not the same as those of conventional sliding supports of balancing machines;
- imperfect assembling during installation of the jointed rotor in the machine;
- modified rotor unbalance in real machine operation conditions induced by, for example, displacement of coupling elements, turbine and compressor discs, generator-rotor winding elements.

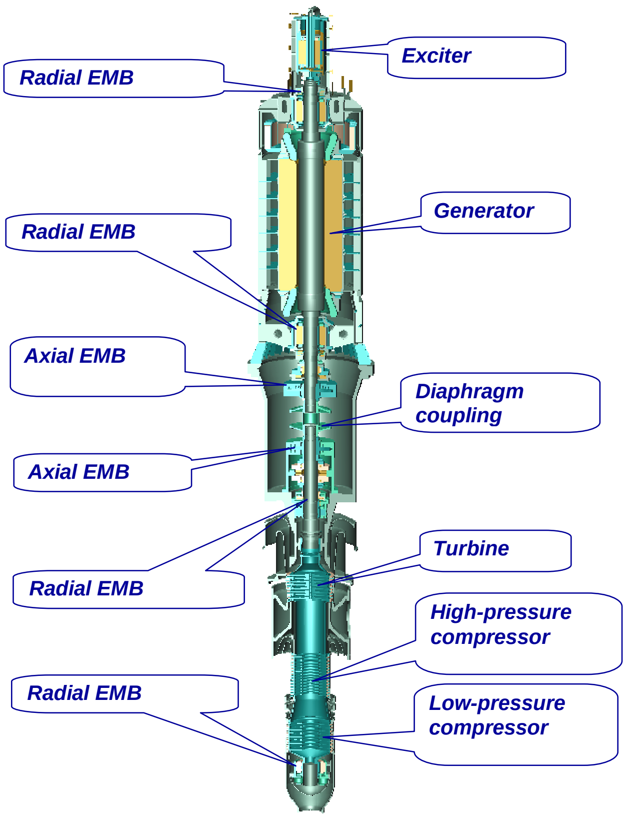

Figure 1: RP TM

In addition, balancing in the TM can be made due to absence of the balancing machine for the jointed rotor or economic inexpediency of balancing associated with machine reassembling.

3 Development of the Balancing Code and Procedure

In order to balance the rotor based on the data from the EMB control system using balancing weights and with account of recommendations [4, 5], the "Procedure for flexible rotor balancing using EMBs" and BALANS computer code were developed. The first verification step for the BALANS code was carried out in the course of flexible vertical rotor balancing using EMBs in the Minimockup test facility in 2008.

Development of methods for identification of the real unbalance and methods for balancing flexible vertical rotors using EMBs implies fine-tuning of the balancing procedure and stepwise verification of the code using the EMB-supported rotor mockup and model. This work will result in balancing the rotor of the full-size TM as per the worked-out procedure based on verified computer codes.

The flexible vertical rotor is balanced using EMBs to provide for the allowed unbalance, at which the rotor oscillation amplitude does not exceed assigned values. Balancing includes identification and correction of rotor unbalance. As the rotor model/mockup should have frequency properties of the TM full-size rotor, the operation band of rotational frequencies for rotor models/mockups shall have two rigid and two bending rotational frequencies. Respectively, in the course of balancing such rotors shall be balanced at four natural oscillation modes, which are given in Fig. 2 where the rotor rotation axis is shown using the dashed line.

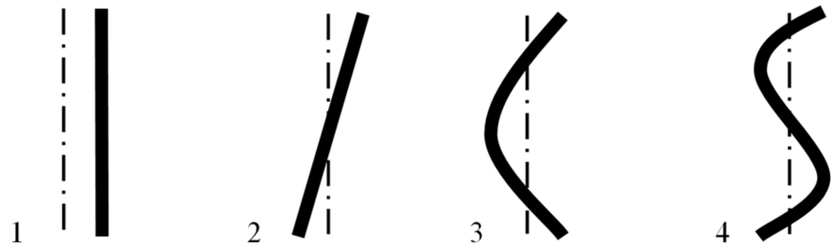

Figure 2: Natural rotor oscillation modes

The rotor is balanced using EMBs by executing balancing cycles as per [4]. Each cycle is a combination of actions to reduce the residual unbalance at frequencies of zero to the maximum achieved value. During the cycle, the real unbalance compensated for by the system of balancing weights differs from the calculation one due to limitations for mass and location of weights. In addition, during program refining in the part that describes weight mass it was necessary to specify several variants of mass weights; the angular position was retained.

The testing procedure includes:

- actuation of equipment, launching of software, check of test facility component operability, warming-up of equipment, two speed-ups and outages and recording in different formats;
- data processing, graph plotting and experimental data analysis (rotor oscillation amplitude for all cross-sections, deviation of the rotor rotational axis from zero marks as per sensors, structure of generated frequencies for all cross-sections, EMB currents);
- comparison with previous testing;
- discussion of results, data output for calculation using the BALANS code.

Speed-up is carried out to the value that exceeds (by 2 Hz) next resonance frequency or to the lower value, if the rotor oscillation amplitude at speed-up exceeded the maximum allowed level.

The accuracy of the BALANS code is improved by its stepwise verification based on obtained experimental data. This shall be done to improve accuracy of unbalance identification, i.e. definition of the value for its angular direction for all cross-sections and calculation of balancing weights. During verification, description that is more exact is entered into the code:

- design features of the test facility (angular positions of sensor installation axes);
- deviations in the rotor-suspension axis position with respect to axes of the test facility specified using sensors during rotation.

In addition, in the course of deviation detection and balancing procedure perfection, calculation algorithms are corrected.

During calculation of balancing weights, the higher oscillation mode takes into account all lower oscillation modes. If as a result of balancing of the higher mode balancing of the lower mode failed, balancing of the lower mode takes place again.

During investigation, loads of different mass (which produce unbalance) are applied at different sections of the rotor. The following unbalance parameters can be changed at that:

- location of the load along the vertical axis of the test facility;
- location of the load based on the angular position;

- weight mass.

Testing is carried out in the operating region of rotational frequencies, at frequencies maximally close to resonance ones. During testing, it is necessary to register the rotor radial position, rotor angular position, rotational frequency. It is necessary to define dependence of rotor oscillation amplitudes on the above parameters for the entire allowed (or maximally possible) variation range of these parameters. When registering the dependence, it is necessary to record amplitude values for at least three points of the parameter variation range, i.e. lower range limit, upper range limit and middle range limit. This should be done to determine the impact of the parameter on the oscillation amplitude.

To carry out testing as per this procedure, it is necessary to provide that the rotor will pass four critical speeds at the experimental test facility in the operating rotational frequency range under conditions of full electromagnetic suspension.

At the initial working-out step, natural frequencies are defined, and rotor natural oscillation modes are identified at different fastening conditions, i.e. at free fastening without radial supports and with foil bearings. Natural frequencies are defined in two ways, i.e. by hitting and with the attached vibrator as per [6, 7]. The rotor is actuated by hitting it with the rubber hammer or by the special vibrator. When the rotor is actuated by hitting, acceleration of different rotor points changes. Using peaks of the acceleration spectrums obtained for different parts, values of natural frequencies are determined.

At actuation using the white noise signal, the force in the actuation point and vibration accelerations of different rotor points were simultaneously measured. When bending natural frequencies were determined, the actuating force was directed to the rotor rotational axis, and vibration accelerations were measured at shaft points in two directions perpendicular to the rotational axis; the actuating force was measured at the vibrator pull rod fastening point. Based on experimental data, acceleration speed is defined using relation of acceleration and the actuating force. Using peaks of the acceleration speed spectrum, values of natural frequencies are determined.

The algorithm for identification of unbalance for the flexible rotor supported by EMBs is based on solution of the inverse problem of rotor dynamics [8]. One of important applications of this algorithm is to use it for rotor balancing using EMBs in their operating conditions. The balancing problem is solved in two steps: a) identification of residual unbalance distribution; b) selection of corrective weights. The vertical rotor is considered, of which the axis coincides with the *0X* axis in the Cartesian coordinate system *0XYZ*. Unbalance distribution for the rotor is described by its value and direction in all cross-sections of the rotor, i.e. by two functions of the rotor axis coordinate or by the vector function:

$$e(x) = (e_1(x), e_2(x)). \tag{1}$$

Unbalance identification is finding approximation for the above two functions. For the approximation, the system of base functions is used, which includes rotor static deformation modes $U^0_k(x)$ and natural oscillation modes $U_n(x)$. The problem is formally reduced to defining expansion coefficients of unbalance functions using the above system of functions. It should be noted that the system of functions used is not orthogonal. The developed rotor-unbalance identification algorithm and balancing-weight selection procedure use the mathematical model of dynamics of the rotor balanced using EMBs. For initial rotor dynamics equations with distributed parameters, the solution is tried using the following expansion

$$U_y(x,t) = \sum_{k=1}^{K} c_k(t) \cdot U_k^0(x) + \sum_{n=1}^{N} d_n(t) \cdot U_n(x). \tag{2}$$

K is the number of radial EMBs; *N* is the number of natural oscillation modes considered in the solution expansion. The function (2) describes rotor deviation in the horizontal direction *0Y*. The similar expression is correct for rotor deviation in the direction of the 0Z axis. When the rotational frequency is constant ω = *const*, rotor stationary oscillations along the *0Y* are described by the following system of equations (3)

$$M^{00}\frac{d^2c}{dt^2}+M^{01}\frac{d^2d}{dt^2}+Cc=D(t)+\omega^2[f_1^c\cos\omega t+f_2^c\sin\omega t],$$
$$(M^{01})^T\frac{d^2c}{dt^2}+m\frac{d^2d}{dt^2}+m\Omega d=\omega^2[f_1^d\cos\omega t+f_2^d\sin\omega t], \tag{3}$$

where $c=(c_1,\ldots,c_K)^T$, $d=(d_1,\ldots,d_N)^T$ are vectors of generalized coordinates, of which components are coefficients of the solution expansion (2). *m* is the rotor mass. M^{00}, M^{01} are mass matrices of dimension of *KxK* and *KxN*, respectively. *C* is the rigidity matrix of the *K* degree for the vector of generalized coordinates *c*. Ω is the diagonal matrix of the *N* degree, of which the elements are squares of ω_n frequencies for respective natural oscillation modes (the upper index *T* corresponds to the matrix transposition procedure). *D(t)* is the vector of EMB active forces. $f_i^c,\ f_i^d\ (i=1.2)$ are vectors of dimension of *K* and *N*, respectively, which describe rotor unbalance and which are defined by the following expressions.

$$f_i^c=(f_{i1}^c,\ldots,f_{iK}^c),\quad f_i^d=(f_{i1}^d,\ldots,f_{iN}^d),\quad f_i=((f_i^c)^T,(f_i^d)^T)^T,$$
$$f_{ik}^c=\int_0^l\mu(x)\cdot e_i(x)\cdot U_k^0(x)dx,\quad f_{in}^d=\int_0^l\mu(x)\cdot e_i(x)\cdot U_n(x)dx,\quad i=1.2. \tag{4}$$

μ is the length mass of the rotor; *l* is its length.

The rotor motion process is monitored using *P* sensors $(P\ge K)$ located in cross-sections having coordinates $x_i\ (i=\overline{1,P})$. Based on (3), rotor displacement at location of sensors $H=(H_1, H_2,\ldots,H_P)$ relates to introduced vectors of generalized coordinates *c*, *d* as follows.

$$H=V\cdot X,\quad V=(V^c,V^d),\quad V^c=\{V_{ij}^c\},\quad V^d=\{V_{ij}^d\},$$
$$V_{ij}^c=U_j^0(x_i),\quad V_{ij}^d=U_j(x_i),\quad X=(c^T,d^T)^T. \tag{5}$$

The procedure for definition and solving of the system of equations to define vector components $f_i\ (i=1.2)$ is similar to that stated in [8]; the only difference is that each measurement of the rotor oscillation process at one rotational frequency makes it possible to set up not *K* but *P* equations.

Values of EMB forces that control rotor motion (*D*-vector components) depend on EMB magnet currents (*I(t)* vector of dimension of *2K*) and rotor displacements at EMB locations (*c* vector) [9]. The functional dependence *D(c,I)* is defined by the EMB design, magnet currents are directly measured in the course of experiments, and rotor displacements at EMB locations are measured using connection (5) with account of measured rotor displacements at sensor locations (*U* vector).

Rotor balancing presupposes installation of additional masses $m_j\ (j=\overline{1,S})$ in assigned *S* cross-sections x_j^0 of the rotor (in correction planes). Mass position defines the r_j distance from the rotational axis and the α_j angle between the line, which connects the rotational center with the mass installation point, and the selected radial direction (angle null).

Corrective balancing weights are intrinsically the same as artificially introduced concentrated unbalance forces and their consideration in dynamics equations (3) results in additional summands in right parts.

$$\omega^2(g_1^c\cos\omega t+g_2^c\sin\omega t)\quad u\quad \omega^2(g_1^d\cos\omega t+g_2^d\sin\omega t). \tag{6}$$

Herein, $g_i^c, g_i^d \ (i = 1.2)$ are vectors of dimension of *K* and *N*, respectively. Components of these vectors relate to characteristics of corrective weights as the following expressions.

$$g_i = ((g_i^c)^T, (g_i^d)^T)^T, \quad i = 1{,}2. \quad g_1 = W \cdot Y, \quad g_2 = W \cdot Z,$$
$$Y = \{Y_j\}, \quad Z = \{Z_j\}, \quad Y_j = m_j r_j \cos\varphi_j, \quad Z_j = m_j r_j \sin\varphi_j, \quad j = \overline{1,S}.$$
$$W = (W^c, W^d), \quad W^c = \{W_{ij}^c\}, \quad W^d = \{W_{ij}^d\},$$
$$W_{ij}^c = U_i^0(x_j^0), \quad W_{ij}^d = U_i(x_j^0). \tag{7}$$

Thus, resultant action of periodic forces, which are induced by residual rotor unbalance and corrective weights, on rotor dynamics in equations of the mathematical model (3) is defined by sums of vectors $f_i + g_i \ (i = 1.2)$. Apparently, the condition of $f_i + g_i = 0 \ (i = 1.2)$ is the condition for ideal rotor balancing. With account of (8), this condition is as follows.

$$W \cdot Y = -f_1, \quad W \cdot Z = -f_2. \tag{8}$$

The developed algorithm implements rotor balancing for natural oscillation modes of the rotor supported by resilient supports, of which rigidity equals to rigidity of radial EMBs. In this connection, (8) transits to normal coordinates (similar to [4]).

$$W_H Y = -F_1, \quad W_H Z = -F_2, \quad W_H = X_*^T W, \quad F_i = X_*^T f_i, \quad i = 1.2. \tag{9}$$

In matrix equations (9), the sequence of equations corresponds to sequence of critical rotor rotational frequencies (including rotor as the solid body).

The ideal rotor balancing condition falls into two unrelated systems (*K*+*N*) of linear algebraic equations with *S* unknowns.

Three situations are possible.

1)*S*>*K*+*N*. The number of correction planes is more than the number of dynamic degrees of freedom for the rotor that influence forced rotor oscillations in the frequency-operating band. The system of equations (9) is underdefinite; it has infinitely many solutions. In order to close the system of equations, the condition for minimization of all weights is additionally applied.

$$\min\left(\sum_{j=1}^{S} (Y_j^2 + Z_j^2)\right).$$

2)*S*=*K*+*N*. In this case, the system (9) has one solution.

$$Y = -(W_H)^{-1} F_1, \quad Z = -(W_H)^{-1} F_2. \tag{10}$$

3) *S*<*K*+*N*. The number of correction planes is less than the number of dynamic degrees of freedom for the rotor. The system of equations (9) is superdefinite; some equations are contradictory. In practice, this means that when there are so many correction planes it is impossible to ideally balance the rotor. In this case, it is necessary to select passage of which frequencies proper shall be provided during balancing. In this connection, *S*-number of equations shall be left out from the group of equations (9), which are responsible for balancing at selected *S* oscillation modes. As a rule, these are *S* equations from each the system. However, if in the rotor operating rotational band there are more critical frequencies than correction planes selected modes shall include the modes, of which natural oscillation frequencies are close to rotor operating rotational frequencies. Thus, (9) generates the following system of equations,

$$W_H^S Y = -F_1, \quad W_H^S Z = -F_2, \tag{11}$$

of which the solution looks like

$$Y = -(W_H^S)^{-1} F_1, \quad Z = -(W_H^S)^{-1} F_2 \quad , \tag{12}$$

and completes calculation of the value and position of corrective weights.

Accuracy of identification of unbalance vectors F_1, F_2, which are used to calculate corrective weights, largely depends on the available set of and accuracy of experimental data. It should be taken into account that the accuracy of identification of components of vectors F_1, F_2, which correspond to natural frequencies that significantly exceed frequencies for experimental investigations, can be poor. In this connection, it is expedient to carry out balancing only for natural oscillation modes, of which natural frequencies are close to the band of experimental frequencies. Therefore, even if the number of correction planes is larger than dimensions of vectors F_1, F_2 it is expedient to generate the system of equations (9) only for lower natural frequencies close to the band of experimental frequencies.

Using the calculated components of vectors Y, Z, values of correction masses m_j, their distance from the r_j rotational axis and angle of installation of mass in the φ_j correction plane are defined in accordance with (7).

The procedure for identification of the rotor residual unbalance in operation conditions and the procedure for calculation of positions of corrective weights are implemented in the BALANS code.

Using the BALANS code, rotors of the small-size [2] and large-size [1] test facilities were balanced.

The procedure for rotor balancing using EMBs in operation conditions is based on solution of the inverse problem of rotor dynamics using the mathematical model (3). In this connection, the balancing process was preceded by the step of design parameter identification, after which rotor rigidity characteristics that influence its own dynamic properties (frequencies and modes of natural frequencies, static deformation modes), as well as dependences of radial EMB forces on winding currents and rotor displacements were specified.

In order to exclude the influence of forces not taken into account in the model (3) on rotor behavior, all experimental records for the dynamics process (rotor rotation angles and rotational speed, rotor displacements in cross-sections of location of horizontal displacement sensors, currents of radial EMB magnets) were made at rotor coastdown.

The developed procedure has the advantage as compared to the current ones [10, 11] due to the following. Firstly, the rotor in operation conditions is balanced in conditions when it is supported the same as in real life and, consequently, with account of dynamic characteristics of the specific design. Secondly, the number of rotor start-ups (speed-ups) is decreased many times that appreciably reduces the labor input of the balancing process; and thirdly, balancing can be made using the conventional measurement system without application of additional equipment.

4 Testing of the Balancing Code and Procedure

By the BALANS code, the rotor was balanced using EMBs in one of the experimental test facilities of JSC "Afrikantov OKBM", i.e. in Minimockup test facility. The Minimockup test facility (Fig. 3) was developed and made in JSC "Afrikantov OKBM" in order to work out the control algorithms for the EMB control system and in order to verify the DIROM code for calculation of the rotor supported by EMBs.

Figure 3: Minimockup test facility

The mechanical part of the test facility consists of the casing and rotor that rotates in two radial and one axial EMBs. In ther lower part of the rotor, the drive achyncronous motor locates that is fed by the frequency transducer. The test facility is fitted with cabinets of the control system and information measurement system. The rotor has four critical frequencies in the operating rotational frequency band (0 to 6000 rpm).

The measurement system being the part of the EMB control system makes it possible to continuously obtain information on rotor rotational speed and displacements, EMB magnet currents. The vertical flexible non-uniform rotor 1 m long with the weight of 16 kg is supported by two radial EMBs, which are located at a distance of 230 mm and 750 mm from the upper butt end of the rotor, and one axial EMB. Three lower natural frequencies of rotor bending oscillations are 30 Hz, 60 Hz and 115 Hz.

The balancing process was preceded by the design parameter identification step, at which rotor rigidity characteristics, dependences of radial EMB forces on winding currents and rotor displacements were specified.

In order to exclude the influence of forces not taken into account in the model (4) on rotor behavior, all experimental records for the dynamics process (rotor rotation angles and rotational speed, rotor displacements in cross-sections of location of horizontal displacement sensors, currents of radial EMB magnets) were made at rotor coastdown. In the course of balancing, three experiments were made at rotor spinup and coastdown. After the first and second experimend using the BALANS code, two systems of corrective weights were calculated and installed on the rotor. To calculate weights, fragments of coastdown oscillograms were used that correspond to 30 rotor revolutions in time. As the rotor was running down for long enough, in 30 revolutions the rotor rotational frequency practically did not change. The difference between rotor rotational frequencies for different oscillogram fragments was 3 Hz. Fig. 4 gives typical experimental dependence of rotor displacement along one of horizontal axis on time, and Fig. 5 shows EMB winding current change.

Corrective weights were calculated for the variant when the rotor is balanced using five natural oscillation modes (two solid body oscillation modes and three bending oscillation modes). The sum of all corrective weights for the first system is 700 g mm and for the second system is 180 g mm.

Dependences of the rotor deviation level at coastdown on its rotational frequency at location of displacement sensors before balancing (black curve), after installation of the first portion of corrective weights (green curve) and after installation of the second portion of corrective weights (red curve) are given in Fig. 6–8. Fig. 6 corresponds to the cross-section of the rotor upper part; Fig. 7 corresponds to the cross section of the rotor central part; Fig. 8 corresponds to the cross-section of the rotor lower part.

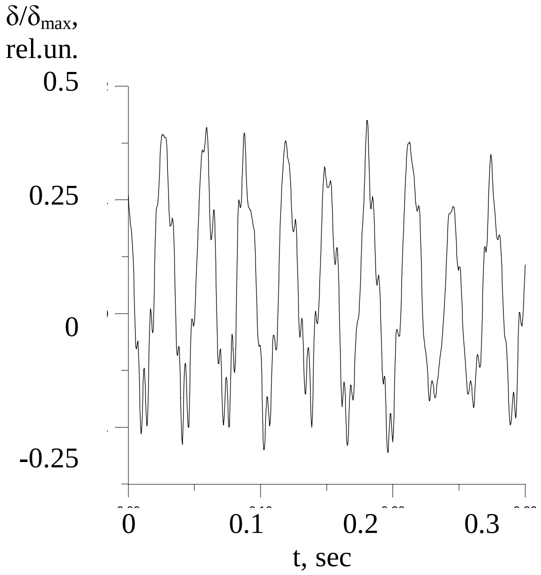

Figure 4: Typical experimental dependence of one horizontal axis displacement on time

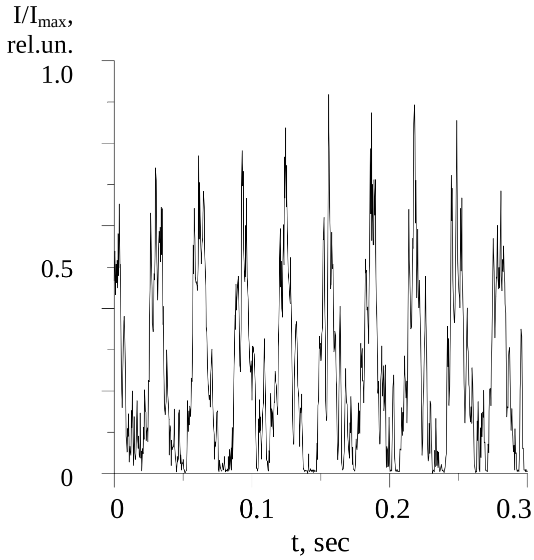

Figure 5: Typical experimental dependence of the EMB winding current change on time

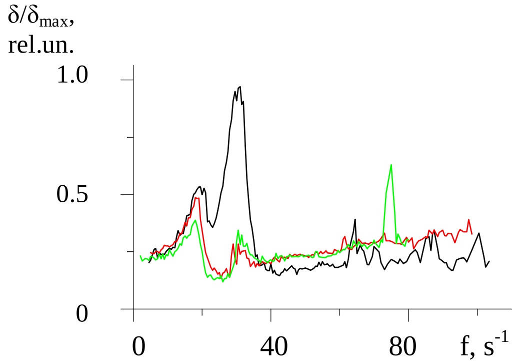

Figure 6: Dependences of rotor upper part deviations at coastdown on its totational frequency

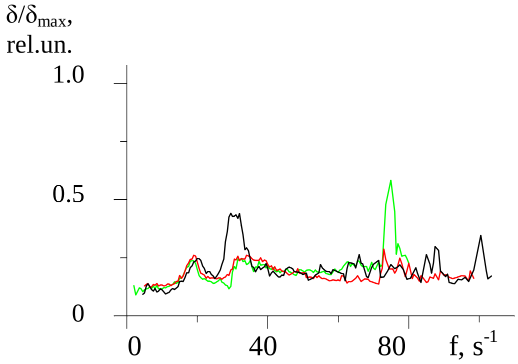

Figure 7: Dependences of the rotor central part deviation level at coastdown on its rotational frequency

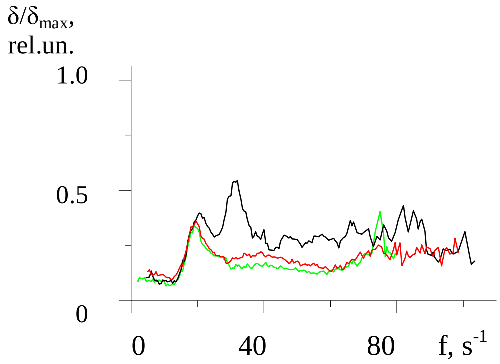

Figure 8: Dependences of the rotor lower part deviation level at coastdown on its rotational frequency

Based on graphs given in Figs. 6–8, after two systems of corrective weights had been installed on the rotor the rotor deviation level in check cross-sections decreased by nearly two times; resonance oscillations were practically eliminated at critical rotational frequency that corresponds to the first rotor bending oscillation mode.

Performed testing of the BALANS code and procedure for balancing the vertical flexible rotor using EMBs in conditions of the Minimockup test facility confirmed a possibility that it can be used to balance rotors of the TM RSM test facility.

In 2011, work was carried out to further test and verify the BALANS code in the TM RSM test facility [1]. Its rotor was designed based on modeling rigidity and mass distribution for the model rotor and full-size TM rotor; equality of number of natural oscillation frequencies and similarity of rotor bending modes were provided. The vertical rotor consists of two parts, i.e. turbocompressor rotor model and generator rotor model connected via the resilient coupling. Each rotor is supported by two radial and one axial EMB. Basic technical specifications of the RSM are as follows:

- rotor full weight……………1171 kg;
- rotor total length…………...10.54 m;
- rotor rotational frequency.....6000 rpm;
- load-bearing capacity of the radial EMB, not less than………………………….250 kg;
- load-bearing capacity of the axial EMB, not less than…….……………………2000 kg.

Investigations were carried out at generator-rotor model rotation, which like the real TM generator rotor had two rigid and two bending critical rotation frequencies. The balancing process was performed consequent to residual balance compensation for each rotor critical rotational frequency. The rotor was speeded up to the rotational frequency maximum possible similar to critical frequency, and then the rotor freely ran out. Data on coastdown recordered by the information measurement system (EMB currents, displacements in all rotor cross-sections) were used as input data for the BALANS code that was applied to process experimental results and define residual unbalance in correction planes. One to three experiments was carried out for each iteration. After installation of the calculated corrective weight set, the rotor was speeded up to the same rotational frequency. With positive results, i.e. when the rotor displacement amplitude did not exceed 30% of the gap in the CB at the nominal rotational frequency and 40% at the critical rotational frequency, a transition was made to balancing at the successive rotor critical rotational frequency. With negative results, the above procedure was repeated. In the same way, the rotor was balanced at successive rotor critical rotational frequencies.

During testing of the BALANS code and procedure for balancing the flexible rotor in the TM RSM test facility, the EMB control system was optimized, the number of correction planes for the generator rotor model was increased, and procedure sensitivity to errors of input data was analyzed.

In the course of balancing, thirty-four balancing cycles were carried out; one hundred and six weights with the total mass of not more than 0.2% of the rotor mass were installed. The rotor-model oscillation amplitude at speed-up and free coastdown during passing of first four natural frequencies does not exceed values specified by the testing program. The maximum value at location of CBs was 34% of the CB gap at the allowed value of 40%). The impact of radial EMBs at rotor speed-up to the nominal rotational frequency on EMB housings is insignificant, i.e. the vibration speed is not more than 0.013 mm/s, vibration displacement is not more than 0.001 mm.

The generator rotor model balanced in the RSM test facility made it possible to carry out speed-up to the nominal rotational frequency of 73.3 s^{-1} (4400 rpm) and pass four critical frequencies; the rotor model oscillation amplitude at CBs did not exceed values specified by the testing program (see Fig. 9).

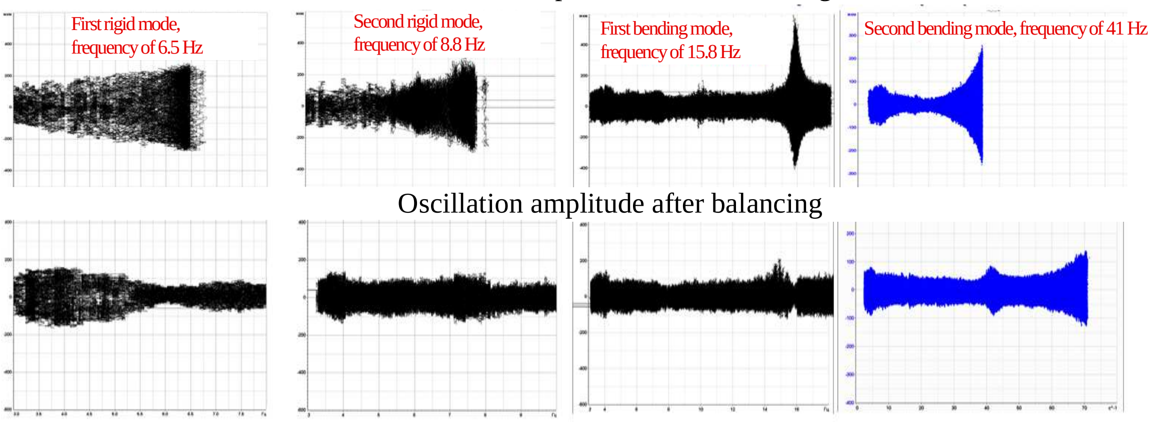

Figure 9: Results of generator rotor model balancing in the RSM test facility (rotor oscillation amplitudes)

5 Conclusions

1 The advantage of the developed procedure as compared with the current ones [4, 9, 210] is due to the following. Firstly, the rotor in operation conditions is balanced in conditions when it is supported by EMBs the same as in real life with account of dynamic characteristics of conventional metal structures. Secondly, this procedure makes it possible to decrease many times the number of rotor start-ups (speed-ups) that appreciably reduces the labor input to the balancing process; and thirdly, balancing can be made using the machine EMB control system without application of additional equipment.

2 Performed rotor balancing made it possible to speed up the generator rotor to the nominal rotational frequency of 73.3 s^{-1} and pass first four critical frequencies. The allowed level of the oscillation amplitude prescribed by the testing program was provided.

3 Solving of the problem to work out the procedure for balancing the flexible jointed rotor using EMBs in the RSM test facility and verification of the BALANS code will allow to obtain a tool for possible rotor balancing using EMBs in units for various applications (gascompressor units, turbocompressors, general devices, etc.).

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