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Computational Analysis of the Vibration of a Large - Scale Piping System with Emphasis to Fatigue Defect Growth Evaluation

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1. Introduction

One of the most important components of many industrial enterprises is a piping system carrying technological gasses and fluids. The piping is particularly characteristic to the thermal energy, chemistry and oil processing plants where the breakdown of a piping system can cause serious consequences - from the troubles in technological processes to the significant environmental damage and casualties among the staff. The goal of this work was to develop a computational approach supported by measurement results for the evaluation of the vibrations of large scale piping excited basically by the hydrodynamic forces and its stress - strain state with the emphasis on the growth rate of hypothetic cracks that inevitably exist in any engineering structure. As it is very difficult to evaluate the vibratory forces caused by hydrodynamic origin explicitly, an assumption was made that elastic vibrations of the pipeline take place as a consequence of internal pressure pulsation and the level of excitation is being identified by means of the analysis of a finite element computational model and by employing the given measured amplitudes of vibration at certain points of the pipeline.

2. Valid Approaches for Piping Vibration Reduction and Evaluation

The mechanism of vibration excitation in real engineering systems usually is multi - source and very complex. The currently valid normative documents [1-3] formulate the requirements for the evaluation of steady state vibrations and vibration strength of piping and other equipment. The main condition of vibration strength of piping is achieved by detuning the natural frequencies of the piping from the determined frequencies of steady state excitation. The hydrodynamic vibration excitation forces take place due to the operation of pumps, regulating valves and other equipment installed in piping. The excitation frequencies of interest are as follows:

- the main frequency of rotation of the pump shaft $\omega = 2\pi n/60$, where n – speed of rotation of the shaft, rpm;
- the frequency of alternation of electromagnetic forces in the electric motor of the pump $\omega = 2\pi nZ/60$, where Z - the number of grooves in the rotor of electric motor of the pump;
- the frequency of alternation of hydrodynamic forces, depending on the number of vanes of pump blade wheel Z - $\omega = 2\pi nZ/60$;
- the frequency of hydrodynamic forces arising due to disruption of whirls in the flow over local obstacles in the piping $\omega = 2\pi St v/d$, where St - Struhal's number, v - velocity of the flow, d - characteristic dimension of obstacle.

In reality, despite of detuning the main harmonic components of the excitation from natural frequencies of the structure, the vibration can still reach considerable levels and the more detailed analysis of its effect upon the structure is necessary [4, 5]. The vibration strength can be evaluated by calculating the lifetime of the structure on the base of the values and distribution of stress amplitudes over the structure and by applying the approved approaches based on fracture mechanics [6]. The sum of defect caused by wide - spectra response is obtained by combining the harmonic component responses contributing more than 10% of the maximum component amplitude. The asymmetry of the loading cycle is to be determined by evaluating the average stress value which is considered to be equal to the constant local stress value taking place due to the effect of mechanical and thermal loading and residual tensile stresses.

3. Computational Procedures for Vibration Analysis

At the design stage the procedure of detuning of the natural frequencies from the frequencies of the main harmonic components of excitation is performed on the base of natural frequency analysis of the structure. The procedure is carried out by solving the structural eigenproblem as:

$$det([K] - \omega^2[M]) = 0 \qquad (1)$$

where $[K]$ is stiffness matrix of the structure; $[M]$ is mass matrix of the structure including the mass of the fluid in pipes. After solving problem (1) the natural frequencies ω_1, ω_2, ... , ω_n and corresponding mode shapes $\{y^{(1)}\}$, $\{y^{(2)}\}$, ... , $\{y^{(n)}\}$ are obtained.

The excitation of piping by the forces arising due to pulsations of pressure and velocity of the flow can be qualified as a random loading of a wide frequency range. The frequency spectrum of the variable component of pressure is obtained experimentally by measurement. As the internal pressure in the pipe is one of the possible loads applied to pipe elements in ANSYS, the vibration amplitudes of the structure are calculated. In order to obtain the values of excitation forces over all frequency range two different approaches can be employed.

<u>*1 approach (simplified)*</u>. The spectral density of the excitation is calculated by applying the Fourier transform to the time law of pressure pulsation obtained by averaging the time laws recorded during several independent measurements. Further, the obtained spectral density of

harmonic excitation is regarded to be deterministic and is approximately replaced by a discrete spectrum over the frequency range of interest, Fig. 1. The response of the structure to every harmonic component of the discrete excitation spectrum is computed by solving the matrix equation in terms of complex amplitudes of vibration as

$$\left([K]-\omega^2[M]+j\omega[C]\right)\left(\{U_R\}+j\{U_I\}\right)=\{F_R\}+j\{F_I\} \quad (2)$$

where $[C]=\alpha_1[M]+\alpha_2[K]$ is the proportional damping matrix of the structure; α_1, α_2 are coefficients, the values of which are selected in order to ensure the given or assumed damping ratio of structural vibrations; ω is frequency of excitation; $j=\sqrt{-1}$ is imaginary number. Vectors of nodal forces $\{F_R\}$, $\{F_I\}$ are real and imaginary parts of the excitation vector. As by obtaining the excitation forces on the base of amplitude - frequency spectrum the phase relations between harmonic components are not available, it is reasonable to assume $\{F_I\}=0$, and excitation vector $\{F_R\}$ at every frequency of excitation ω is calculated in proportion to the amplitude of alternation of the pressure with a given frequency. By using the HARMIC analysis procedure implemented in ANSYS [7] équation (2) is being solved at every frequency of the discrete spectrum of excitation. As a result, the discrete spectrum of amplitudes of nodal displacements and stresses in the piping are obtained and used as initial information for determining the fatigue defect growth rate.

The equivalent level of displacements and stresses taking into account the influence of all harmonic components is calculated as a combination of component amplitudes of the response by using the square root of sum of squares (SRSS) technique giving rather conservative results.

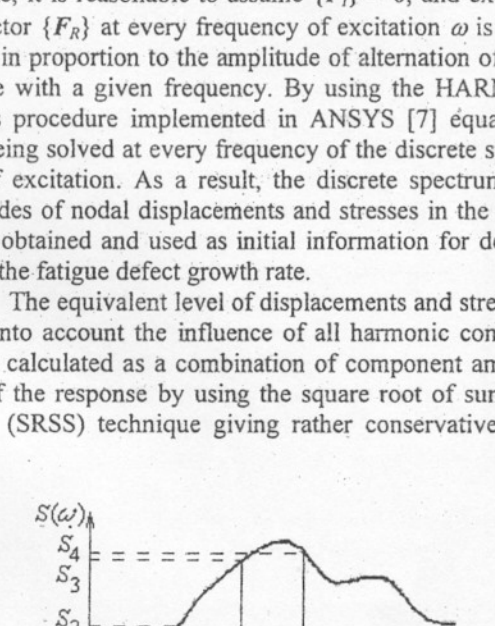

Fig. 1 The approximate transformation of the spectral density (a) into discrete spectrum (b)

2 approach. The random excitation process is assumed to be steady in time. The power spectral density $S(\omega)$ (PSD spectrum) is obtained on the base of measurement results. According to the definition, the PSD spectrum is equal to the spectral density of the autocorrelation function $R(\tau)=E[x(t)x(t+\tau)]$. Through symbol E here we denote the mathematical expectation, and function $x(t)$ is the time law of the random process under consideration. In this way, $S(\omega)$ is obtained by performing the Fourier transform over $R(\tau)$.

The solution is performed as follows. Eigenvalue problem (1) is solved and natural frequencies ω_1, ω_2, ..., ω_n and corresponding mode shapes $\{y^{(1)}\}$, $\{y^{(2)}\}$, ... , $\{y^{(n)}\}$ of the structure are obtained. The structural dynamic equation is presented in modal coordinates z_i as n independent equations

$$\ddot{z}_i+2\omega_i\vartheta_i\dot{z}_i+\omega_i^2 z_i=\left\{y^{(i)}\right\}^T\{f\}x\{t\}\ ,\ i=1,\ \dots\ ,\ n \quad (3)$$

where ϑ_i is damping ratio of the i th mode; $\{f\}$ is dimensionless vector of nodal forces describing the distribution of force amplitudes over the structure; $x(t)$ is time function of the variable component of pressure.

Assume $S(\omega)$ to be the PSD spectrum of random process $x(t)$. Then the PSD spectrum corresponding to the i th modal coordinate is obtained as $N_iH_i(\omega)S(\omega)$, where $N_i=\{y^{(i)}\}^T\{f\}$ and $H_i(\omega)=1/\left[\omega_i^2-\omega_i+j(2\vartheta_i\omega_i)\omega\right]$ is the frequency transfer function of a single - mass vibrating system. The PSD response spectrum of the amplitude of j th degree of freedom (d.o.f) u_j of the structure is determined by summation over all modal coordinates as

$$S_{u_j}(\omega)=\sum_{i=1}^{n}\sum_{r=1}^{n}\left\{y_j^{(i)}\right\}\left\{y_r^{(i)}\right\}N_iN_rH_i^*(\omega)H_r(\omega)S(\omega) \quad (4)$$

where asterisk (*) means the complex conjugate quantity.

From the PSD spectrum for displacements the PSD spectrum for stresses is obtained. The integral of the PSD spectrum over all frequency range gives the mean root square value of the quantity the random process of which has been considered. E.g., the mean root square value of the displacement amplitude of j th d.o.f. is obtained as

$$\overline{u}_j^2=\int_0^{\infty}S_{u_j}(\omega)d\omega \quad (5)$$

The approach is implemented in ANSYS as analysis procedure Random Vibration Method.

Validation of the model. The presented description of excitation forces is very simplified therefore the validation of finite element models of the structures to be analyzed is necessary. The computed amplitudes of the structural response have to be compared with measured ones in measurement points. It is possible to obtain maximum adequacy of the computational model to the real structure by selecting appropriate values of certain parameters of the model, e.g. the damping ratios of modal components or the amplitudes of pressure pulsations in different points of the piping. As the first approximation, the excitation vectors (i.e. vector $\{F_R\}$ in the above mentioned 1st approach or vector $\{f\}$ in the 2nd approach) are being scaled in order to obtain the best coincidence of calculated and measured amplitudes at the points of the structure where the measurement results are available.

4. Results of Vibration Analysis of Ignalina NPP Main Feed - Water Piping

For the evaluation of the vibrations of Ignalina NPP main feed - water piping finite element model of the piping was created in ANSYS [7], Fig. 2, by using finite elements PIPE16 charged with water, as well as, rigid, sliding and damping supports under filters, valves and flaps and elastic suspensions of piping.

Modal analysis of the model revealed 164 modes of the feed - water piping within the frequency range 0-200 Hz. The natural frequencies were distributed rather uniformly over the frequency range. The lowest natural frequency equals 3.44 Hz.

The computed natural frequencies were compared to the main frequencies of excitation that have to be considered according to normative documents:

- the main frequency of rotation of the pump shaft: $f = (2985\ \text{rpm}) / 60\ s = 49.75$ Hz;
- the frequency of alternation of hydrodynamic forces, depending on the number of vanes of pump blade wheel: $Z = 7$, $f = 49.75 \times 7 = 348.25$ Hz .

The spectrum of measured pressure pulsations in the piping was essentially different, Fig. 3. The pulsations with the frequencies characteristic to the hydrodynamic processes taking place in regulatory valves - 11, 21, 37, 67 and 100.5 Hz dominate. Neither of them is a multiple number of the main frequency of rotation of the pump shaft nor of the frequency corresponding to the product of angular frequency of rotation of the pump and the number of vanes of the pump blade wheel. The conclusion follows that the real excitation mechanism is very complex and the prevailing harmonic components could be hardly explained basing only upon angular frequencies of the rotating machines. From the site of obeying the recommendations of the normative documents, no obvious mistakes have been made during the design stage in detuning the natural frequencies from excitation ones.

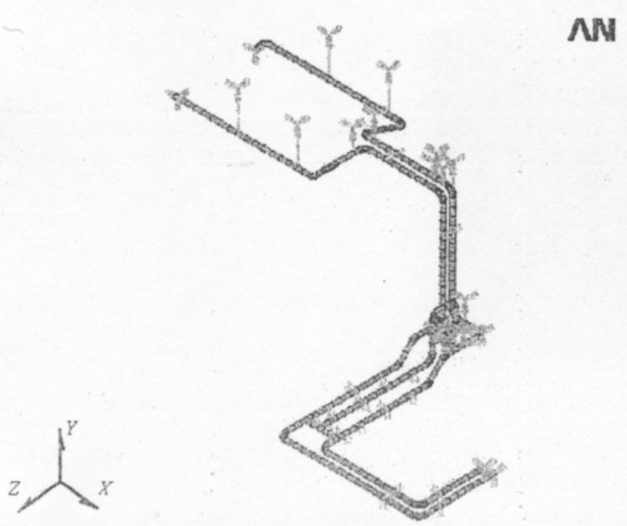

Fig. 2 The finite element model of Ignalina NPP main feed - water piping

As the spectrum of natural frequencies of the piping is dense and close to uniform, the pulsations of pressure inevitably take place at the frequencies close to some natural frequencies of the piping. Therefore a significant level of vibrations is to be expected. The further analysis has been performed using computational methods supported by measurement results as described in section 3. The measurements of the vibrations in 27 points of the piping were performed. The vibration speed and acceleration in the direction of the three coordinate axes were recorded when the reactor has been operated at half and at full power. The amplitude spectrum and level of vibrations were recorded, Fig. 4.

The vibration displacements of the piping under the loading of temperature of 190°C, constant internal pressure of 8.7 MPa and pressure pulsations according to data of measurements were computed using ANSYS. The satisfactory coincidence of experimental and computational results was obtained despite the fact that pressure pulsations were measured not in the feed - water piping directly, but at the end of 20 m length control branch pipe. It can be seen from Fig. 5 that the spectra of displacements in measurement points are similar with respect to frequency components, as well as vibration amplitude level. The comparison of computational and measurement results allows to consider the pressure pulsation as the main source of the excitation and its values can be assumed uniformly distributed over the length of the piping.

In order to obtain more comprehensive information, a special case of loading was simulated. The piping was loaded by the pulsations of pressure of the same constant level over all frequency range 2-150 Hz. The obtained spectra of computed vibration displacements in measurement points are also very close to those obtained by measurement. Thus the assumption was confirmed that the excitation of the main feed - water piping of Ignalina NPP during operation is of random wide range spectrum nature. The frequency relationship of the reaction of the piping to such kind of loading practically does not differ from it's amplitude - frequency characteristic at the same level of excitation. Therefore the combination of responses at different frequencies in order to get the estimation of the maximum stress amplitude level can be performed by combining peak responses taken from the amplitude - frequency characteristic higher than 10% of the maximum one. The obtained results are equivalent stress amplitudes in pipes under the action of separate components of the loading spectrum and under the loading by all 6 components simultaneously.

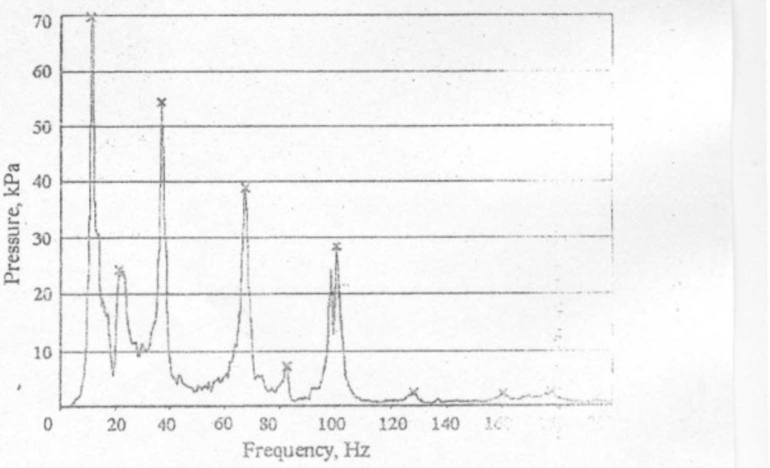

Fig. 3 The pressure pulsations in main feed - water piping of Ignalina NPP operating at full power (1252 MW)

5. Fatigue Crack Growth Analysis for the Lifetime Estimation

As no real defects have been registered during the inspection of the pipeline, normative documents [1-3] require to estimate the growth rate of a hypothetic crack at the location of maximum stress concentration. Therefore

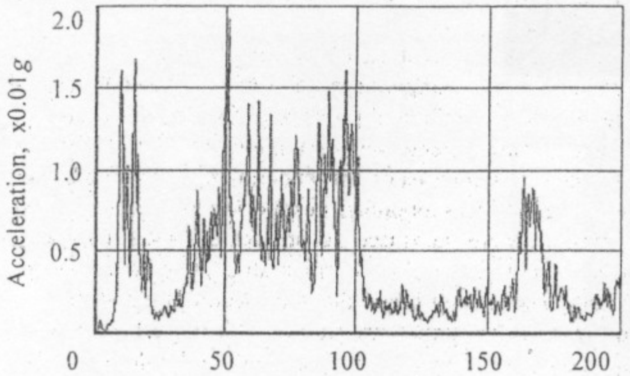

Fig. 4 The example of records of vibration acceleration in the main feed - water piping of Ignalina NPP operating at full power (1252 MW)

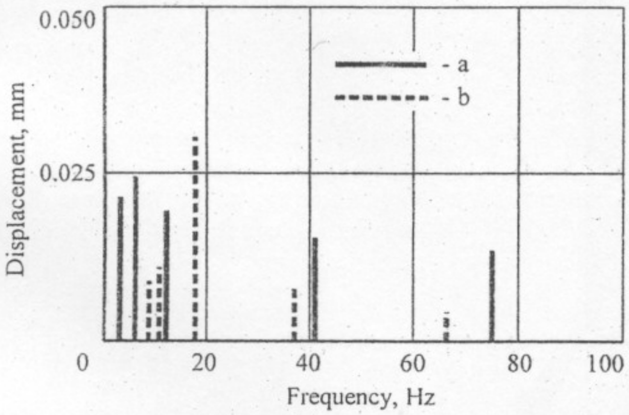

Fig. 5 Measured (a) and computed (b) vibration displacement at the same measurement point of main feed - water piping of Ignalina NPP operating at full power (1252 MW)

the presence of the semi - elliptic surface crack of depth a = 7 mm (1/4 of the pipe wall thickness that equals 28 mm) and length c = 21 mm was assumed.

Norms ASME, Sec. XI, App.A-1000 [3] schedule the following steps for analytical estimation of the fatigue crack growth:

- as initial data, the distribution of crack opening stresses across the thickness of the cracked wall have to be known;
- stress intensity factors are calculated by using analytical relations, as they are available only for typical fragments of structures, the formula derived for the fragment most similar to the structure under investigation is being used;
- the size of the crack at the end of the estimated lifetime and the largest allowable initial crack size are calculated;
- the conclusion about the acceptability of the presence of the hypothetic crack for the further exploitation of the structure is made.

The investigation has been performed using the SACC software [8] incorporating all necessary calculation and estimation steps required by norms [3]. With known values of crack opening stresses across the thickness of the wall (we obtain them by means of the ANSYS program, as described in previous sections) and selected size of the crack the SACC program calculates the stress intensity factors K_{Imin}, K_{Imax} during each loading cycle.

The formula for estimation of the crack growth rate reads as $\frac{da}{dN} = C(\Delta K_I)^n$, where $\Delta K_I = K_{Imin} - K_{Imax}$; C, n - coefficients the values of which depend upon material properties and upon the ratio $R = K_{Imin} / K_{Imax}$; a – linear size of the crack; N- number of loading cycles.

As coefficients C, n were not available for the pipeline under investigation, the values recommended by SACC program for a similar sort of steel have been used. For ferritic steel in air environment we took the following values (ASME, Section XI, Article A-4000):

$$n = 3.07, C = 9.734Q \times 10^{-8}(2.88 - R)^{-3.07} \text{ mm } \left(\text{MPa}\sqrt{\text{m}}\right)^{n}$$

where at $0.79 < R \leq 1$ the scaling multiplier equals $Q = -43.35 + 57.97R$.

An example of the crack growth estimation results is presented in Table. During the real operation time the NPP equipment is subjected to tremendous number of loading cycles. This implies some caution to be taken while interpreting the results. It seems unlikely that the empirical analytical relations have sufficient experimental evidence at so large numbers of loading cycles. Anyway, we present the results of direct application of an approved investigation method.

Table

Crack type	Initial size, mm	Final crack size, mm
Internal longitudinal	Depth - 7 Length - 21	Depth - 22.4, Length - 53.5 0.89×10^9 cycles
External longitudinal	Depth - 7 Length - 21	Depth - 22.4, Length - 54.0 0.77×10^9 cycles
Internal circumferential	Depth - 7 Length - 21	Depth - 22.4, Length 53.7 0.36×10^9 cycles
External circumferential	Depth - 7 Length - 21	Depth - 22.4, Length - 56.5 0.327×10^9 cycles

6. Conclusions

The computational approach supported by measurement results for the evaluation of vibrations and their influence upon lifetime of a large scale piping subjected to wide spectrum loading has been introduced. The analysis has been performed using ANSYS procedures where loading was scaled in order to match the data obtained at measurement points.

Though no obvious mistakes during the design stage have been identified, the requirements of valid normative documents to detune the natural frequencies of the structure from a - priori determined frequencies of excitation appeared to be not sufficient for ensuring long - time safe operation. The presented computations and measurement results demonstrated that the actual vibration level of the pipeline is considerable and confirm the assumption that the pressure pulsation is the main source of the excitation and its values can be assumed uniformly distributed over the length of the piping. The actual excitation during operation of the pipeline is a wide frequency range random process. In order to evaluate the actual lifetime the maximum allowable number of loading cycles has been determined by means of appropriate fatigue crack growth method implemented in SACC software.

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R. Barauskas, V. Eidukynas, V. Grigas, P. Žiliukas

SUDĖTINGO VAMZDYNO VIRPESIŲ ĮTAKOS NUOVARGIO PLYŠIO DIDĖJIMUI SKAIČIUOJAMOJI ANALIZĖ

R e z i u m ė

Šio darbo tikslas - sudaryti metodiką hipotetinių plyšių vamzdžių sienelėse didėjimui dėl hidrodinaminių jėgų sužadintų virpesių poveikio įvertinti. Siūlomos metodikos pagrindas - vamzdyno virpesių skaitinė analizė baigtinių elementų sistema ANSYS. Viena pagrindinių apkrovų - dinaminis slėgis vamzdyne yra registruojama atitinkamomis realaus vamzdyno eksploatacijos sąlygomis ir panaudojama virpesiams apskaičiuoti. Apskaičiuoti virpesių parametrai palyginami su išmatuotaisiais kontroliniuose taškuose ir, nustačius tam tikrus neatitikimo dėsningumus, koreguojami. Konstrukcijos ilgaamžiškumas įvertinamas programa SACC apskaičiuojant maksimalų apkrovimų ciklų skaičių. Skaičiavimų rezultatai panaudoti ruošiant Ignalinos AE II bloko saugumo analizės ataskaitą.

COMPUTATIONAL ANALYSIS OF THE VIBRATION OF A LARGE-SCALE PIPING SYSTEM WITH EMPHASIS TO FATIGUE DEFECT GROWTH EVALUATION

R. Barauskas, V. Eidukynas, V. Grigas, P. Žiliukas

S u m m a r y

The goal of this work was to develop a computational approach supported with measurement results for the evaluation of vibrations of large scale piping excited basically by the hydrodynamic forces with emphasis to the growth rate of hypothetic cracks that inevitably exist in any engineering structure. The vibration analysis has been performed using ANSYS procedures where the loading applied as variable internal pressure was scaled in order to match the data obtained at measurement points. The lifetime evaluation has been performed in terms of the maximum number of loading cycles by using the SACC software. The results of computations have been used practically in preparation the Safety analysis report of the 2nd unit of Ignalina NPP.

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ЧИСЛЕННЫЙ АНАЛИЗ ВЛИЯНИЯ ВИБРАЦИЙ СЛОЖНОЙ СИСТЕМЫ ТРУБОПРОВОДОВ НА ПРИРОСТ УСТАЛОСТНОЙ ТРЕЩИНЫ

Р е з ю м е

Целью настоящей работы является создание методики по оценке влияния вибраций сложной системы трубопроводов, возбуждаемой гидродинамическими силами, на прирост гипотетической усталостной трещины. Основу методики составляет численный анализ вибраций трубопровода с помощью конечно элементной системы ANSYS. Динамическое давление в трубопроводе измеряется на эксплуатационных режимах и используется в расчетах вибраций в качестве возбуждающего воздействия. Полученные при расчетах параметры вибраций сравниваются с соответствующими данными измерений и при необходимости корректируются. Долговечность конструкции оценивается расчетом максимального числа циклов нагружения с помощью программы SACC. Результаты работы использованы в отчете по анализу безопасности II энергоблока Игналинской АЭС.

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