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Dynamic and fatigue analysis of steel space supporing frame structure for turbogenerator

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Анотація: У статті розглядається динамічний вплив на сталеву просторову рамну конструкцію. Результати даного дослідження показують, що дефект будь-якої несучої конструкції впливає на спектр амплітуди вібрації залежно від його розташування, збільшуючи або зменшуючи значення амплітуди. Виявлено безліч видів дефектів турбогенератора, небезпечних для роботи визначеної системи F-V-M, які повинні враховуватися для аналізу явища втоми.

Аннотация: В статье рассматривается динамическое влияние на стальную пространственную рамную конструкцию. Результаты данного исследования показывают, что дефект любой несущей конструкции влияет на спектр амплитуды вибрации в зависимости от его расположения, увеличивая или уменьшая значения амплитуды. Выявлено множество видов дефектов турбогенератора, опасных для работы определенной системы F-V-M, которые должны учитываться для анализа явления усталости.

Abstract. In this paper the steel spatial supporting frame structure is investigated for taking into consideration the dynamic influence. The results of this analysis show that defect of any supporting structure exerts some action on vibration amplitude spectrum, depending on its location, increasing or decreasing the amplitude values. It is revealed that there are many types of defects in turbo-generator, regarded as dangerous for proper F-V-M system determination, which should be taken into account during fatigue analysis.

Key words: analysis, frame structure, vibroisolation, turbogenerator.

Introduction. Until late years of the last century a reinforced concrete had been most commonly applied as a constructional material for turbogenerators supporting structures. For the sake of reinforced concrete features such as fragility and a local loss of durability caused by oiling up of sparkling plugs, after several decades of utilization, concrete foundations have to be carry out a thorough and expensive repairs. In many cases it leads to the inevitability of knocking out the entire supporting structure (fig. 1a, fig. 1b).

Concrete supported structures also have some significant advantages such as:

- vibration dumping because of its own large mass;
- possibility of using simplified analytical model in design process;
- possibility of easy forming of structural shape.

Steel supported structures require much more advanced dynamic and fatigue calculations and particularly careful design vibration dumping system as well as proper forming a shape of structure regard fatigue influence.

a b

Fig. 1. An example of knocking out of concrete supported structure at Patnow power plant

Tools of dynamic analysis of supporting structures. For concrete supported structures dynamic and static analysis is commonly bounded to plane frames analysis. Such simplification is a result of assumptions:

- cooperation of concrete plate joining spandrel beams of frames is omitted;
- mass of structure is dumps upper forms of structure vibrations;
- large stiffness of structure lets for ignoring of fatigue influence.

At present, construction of steel space supporting frames for turbine sets becomes a common trend. Such ones can be repaired in relatively simple way or even be taken apart without trouble in case of serious damages. Modern power industry is based on biomass burning. Because of biomass store troubles 200 MW generators are in common use in a new and modernized power plants.

Steel supported structures (turbogenerator foundations) are *stricte* space bar frames. In static-dynamic calculations, foundations' work has been defined as an foundation-machine (F-M) integrated feedback system's work.

Differently to reinforced concrete supporting structures low mass steel structure elements are not able to dump vibrations in desirable extent. For that reason, all steel supporting foundations must be definitely designed with the integral vibroisolation system. In static-dynamic calculations, steel supporting structures have to be treated as a foundation-vibroisolation-machine (F-V-M) feedback system. So defined F-V-M system's analysis is necessary in taking into consideration various defects of a turbine set. For a high-speed turbogenerators used in power industry turbine bearing chamfering is the most frequent defect typical one generate in turbogenerators during their long-term utilization.

During stabile (normal) turbine work mechanical vibrations of F-M feedback system are characterized by so called “small” displacements and deformations. By “small” stresses σ and deformations δ in the literature e.g. [1], [2] are defined as elastic. Turbines work in such way, in small vibration range may be treated as linear. In this case the stiffness matrix of supporting structures can be described by using of simple motion equations for common system. Calculations get complicated when even one of F-M subsystems has to be described by a nonlinear equations. For example as a nonlinear subsystem in turbine mechanics is an axis of rotor with geometrical imperfections (shaft disalignment) and structural imperfections (bearing chamfering, shaft cracking). Supporting structures of turbogenerators are usually treated as linear subsystems [1].

Mathematical conjugation of a system where subsystems are described both by liner and nonlinear equations is very sophisticated and difficult for numerical dissolving. This problem is one of most difficult in dynamic of rotor machines. Non proper solution of that problem causes significant errors of results. In Poland a team lead by Kicinski and Rzadkowski worked out a group of computer programs called MESWIR [1, 3-5].

MESWIR lets for description of dynamic work in whole range of rotor machine rotary speed both within linear and nonlinear scope also after crossing the limit of the system stability. MESWIR software enables to conduct a machine work analysis with material and geometrical imperfections giving a picture of joints vibration spectrum and a determination of their absolute dislocation trajectories. Simulation of shaft cracking, turbine bearing chamfering or rotor axis deflection is possible. Program enables also for regarding of kinetic-static forces including ones come from residual unbalance, aerodynamics and magnetic tensions. The scheme of MESWIR software analysis range is presented on fig. 2.

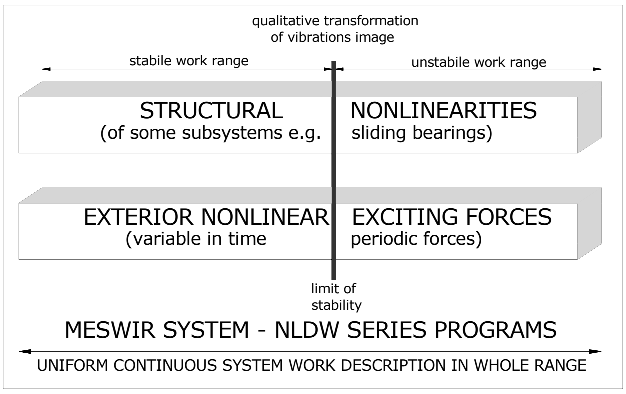

Fig. 2. Scheme MESWIR software rotor machine analysis range

Dynamic and static analysis of supporting structure. Dynamic and fatigue analysis has been conducted for a most popular in polish power industry 200 MW turbogenerator 13K215 (fig. 3) supported on the steel space frame given on fig. 4.

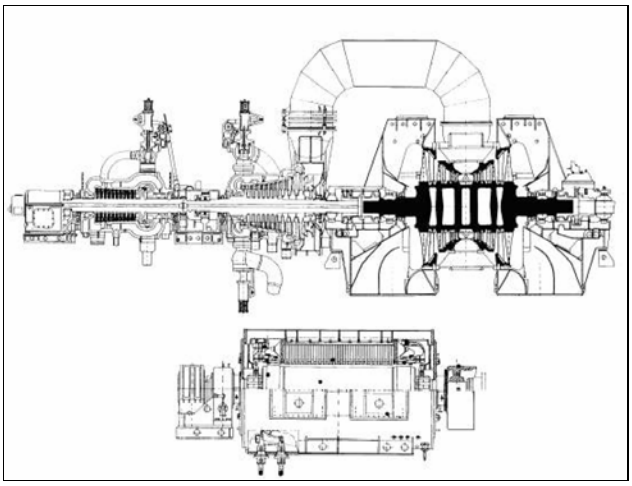

Fig. 3. Scheme of power transmission system of turbogenerator 13K215 [4]

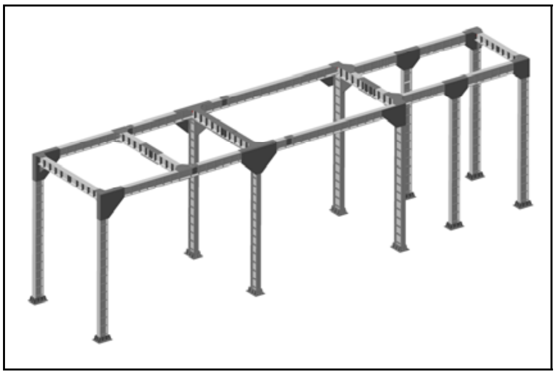

Fig. 4. The space supporting steel frame model rendering
of considered structure

Turbine bearing chamfering was considered as an imperfection giving nonlinear effects in dynamical analysis. The scheme of turbine bearing chamfering is presented on fig. 5.

A location and a size of bearing chamfering are described by two angle α i β that determine the shape of grease crack what enables modeling of bearing chamfering in two planes perpendicular to shaft axle. Further results concerns the chamfering of bearing no. 6 that was modeled as maximum and minimum chamfering value (angle α) in horizontal and vertical plane for which no physical contact of bearing pivot material appears. It means the determination of 4 defect values for considered bearing: α *max*, $\beta = 0°$; α *min*, $\beta = 0°$; α *max*, $\beta = 90°$; α *min*, $\beta = 90°$; [1].

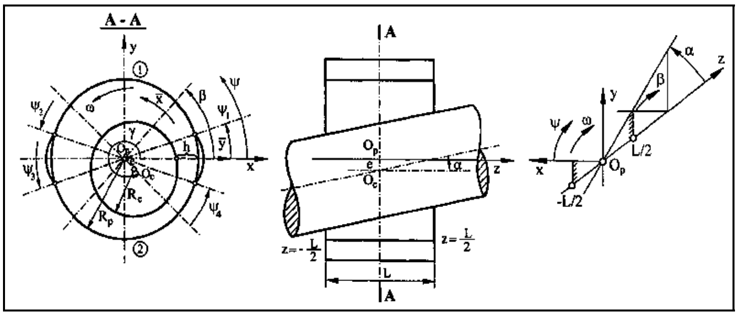

Fig. 5. The scheme of turbine bearing chamfering

Static scheme of analyzed supporting structure of turbogenerator 13K215 is shown on Fig. 6 where preliminary proposed steel profiles are also given. The values of acting forces are given in table 1.

Table 1

The list of turbogenerator static loads and exciting forces

Symbol	Name of element	Static loads G_w from rotating elements of machine [kN]		Exciting forces 0,20 G_w [kN]	
		vertical	horizontal	vertical	horizontal
P_1	Frame and turbine rotor	73,60	23,55	14,72	4,71
P_2	Frame and turbine rotor	88,30	28,50	17,66	5,70
P_3	Frame and turbine rotor	171,70	54,00	34,34	10,80
P_4	Frame and turbine rotor	$4 \times 52,00$	$2 \times 12,26$ $2 \times 16,70$	$4 \times 10,40$	$2 \times 2,45$ $2 \times 3,34$
P_5	Frame, bearing, rotor	147,15	46,10	29,43	9,22
P_6	Generator stator, hydrogen radiator covers, proofings	$4 \times 42,50$	-	-	-
P_7	Generator rotor, bearings	30,40	9,81	6,08	1,96
P_9	Condenser	$4 \times 58,90$	-	-	-
$P_{11\text{-}37}$	Other loads: dampers, pipeline suspensions etc.	76,50	-	-	-
Total of loads		719,15	219,88	143,83	43,98

Above listed values of forces were taken into static calculations with load factors according to [6]:

for static loads:

$$P_s = \gamma_{f1} \cdot G_w \tag{1}$$

and for exciting forces:

$$P_d = 0,20 \cdot \gamma_{f2} \cdot G_w \tag{2}$$

where: G_w – weight of rotating parts; $\gamma_{f1} = 1,2$ $\gamma_{f2} = 5,0$ – load factors.

For dynamic and fatigue calculations load factors are neglected.

Apart from loads listed in table 1 supporting structure is loaded by other variable short-time and long-time forces [6] listed in table 2.

Table 2

The list of additional short and long-time loads from turbogenerator

Force type	Value, kNm	Force type	Value
Short-time loads		Long-time loads	
Rotary moment $\overline{M}$	$108,12*$	Loads of machine thermal deformation (uniform warming of the structure)	35°C for whole structure
Short circuit moment M_z	$9626,40**$	Loads of machine thermal deformation (uneven warming of the structure)	65°C for turbine part 35°C for generator

For the forces listed in table 2 factor $\gamma_f = 1,2$ is taken into static calculations.

The value of rotary moment $\overline{M}$ and short circuit moment M_z are given by formulas:

$$\overline{M} = \gamma_f \cdot \overline{F} \cdot r \tag{3}$$

$$M_z = \gamma_f \cdot 9,55 \cdot \frac{W}{n_m} \cdot k \tag{4}$$

where: $\overline{F}$ – force of turbogenerator own weight; r – force action arm; W – machine rating power [kW]; n_m – rotary speed [rot/min]; $k = 12$ factor for turbogenerators,

* – moment attached to a single joint; ** – force from short circuit moment acting on single bolt are calculated according to formula:

$$P = \frac{M_z}{4a_s} \tag{5}$$

and resulted $P = 567{,}60$ kN for a_s – distance between anchorages of generator part of considered 200 MW turbogenerator.

For static and dynamic calculations of supporting structure were considered load schemes shown on fig. 6.

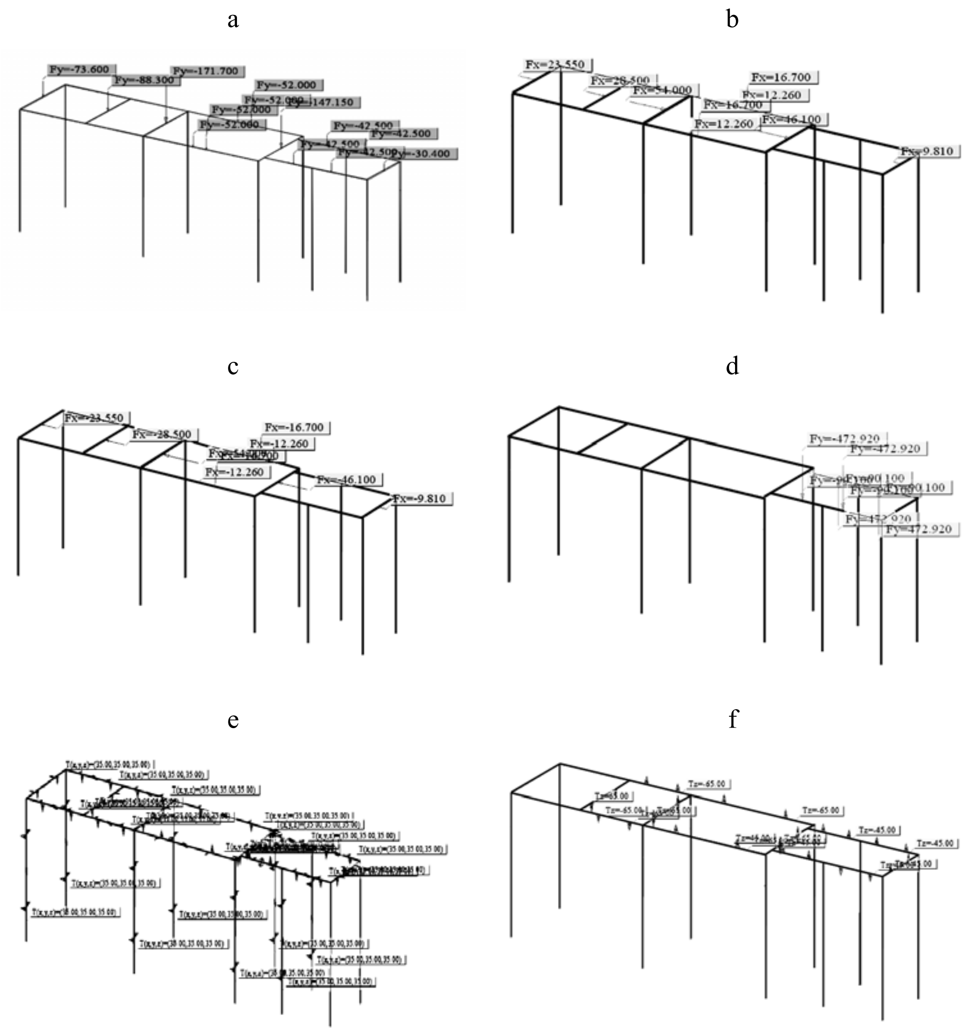

Fig. 6. The schemes of loads for static and dynamic calculations:
a – vertical permanent loads; b – horizontal permanent loads – fore; c – horizontal permanent loads – back; d – vertical forces of short circuit moment and rotary moment; e – uniform warming loads; f – non-uniform warming loads.
Additional exciting forces given in table 1 are located similarly to scheme a, b and c.

Chosen results of dynamic analysis. The main problem in dynamic analysis was determination of a proper vibration spectrum. The analysis of vibrations is essential for fatigue calculations of steel supporting structure. Particularly in steel structures vibration sourced in defects may have a great importance in fatigue analysis.

Vibration spectrums of selected space frame node (bearing no. 6) gained by MESWIR are shown on fig. 7 and 8.

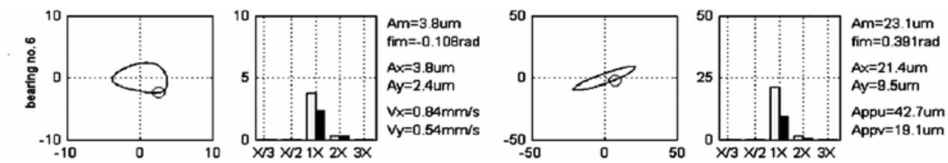

Fig. 7. Forced vibrations of bearing node no.6 in F-M integrated feedback system without defects given in MESWIR software:
left side - relative vibrations, right side - absolute vibrations

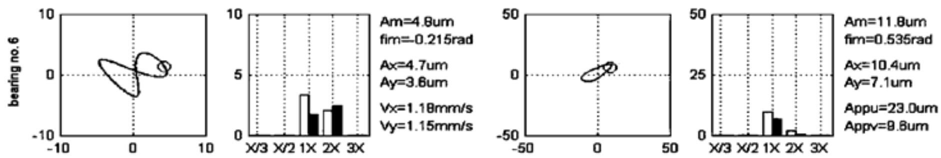

Fig. 8. Forced vibrations of bearing no. 6 in F-V-M integrated feetback system with chamfering of turbine bearing:
left side - relative vibrations, right side - absolute vibrations

The influence of chamfering of bearing no. 6 on vibrations level in the other bearings is presented on diagrams on fig. 9

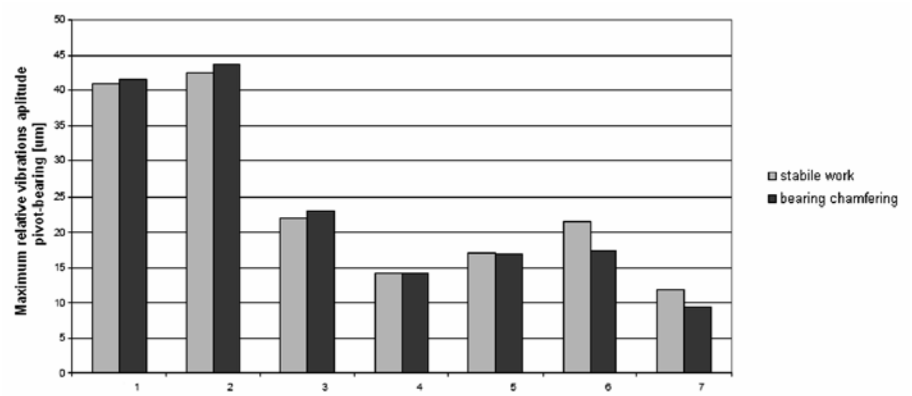

Fig. 9. Diagrams comparing relative vibrations amplitude in all bearings caused by chamfering of bearing no. 6 defect to stabile work (without machine defect)

Supporting structure durability without vibroisolation. An example of fatigue calculation for node no. 6 was conducted. The period of the structure utilization was taken as 30 years. By machine work ($n_m = 50Hz$) without vibroisolation (uniform vibrations spectrum), the utilization time was converted to periods quantity: $n = 4,73 \cdot 10^{10}$. By machine work ($n_m = 50Hz$) with designned vibroisolation (natural frequency of vibroisolation $n = 1,2Hz$) the utilization time was converted to periods quantity: $n_V = 9,46 \cdot 10^8$. The range of variable tensions was determined as: I range: $\sigma_{p1} = \pm 88,75$ MPa for $n_1 = 9,46 \cdot 10^8$, II range: $\sigma_{p2} = \pm 88,75$MPa for $n_2 = n_1$. For fatigue analysis was also taken [7]:

maximum normal tensions inconstancy range for whole spectrum and for S235JR steel:

$$\Delta\sigma = \sigma_{max} - \sigma_{min} = 177,5 \text{ MPa} \tag{6}$$

heterogeneity spectrum factor $\alpha_K = 1$,

equivalent normal tensions inconstancy range:

$$\Delta\sigma_e = \alpha_K \, max \, \Delta\sigma = 1,0 \cdot 177,5 = 177,5 \text{ MPa} \tag{7}$$

- fatigue category: $\Delta\sigma_C = 160\text{MPa}$;
- equivalent fatigue permanent durability: $\Delta\sigma_L = 65\text{MPa}$.

Fatigue durability $\Delta\sigma_R$ was calculated as [7]:

$$\Delta\sigma_R = 0,735\Delta\sigma_C\left(\frac{5 \cdot 10^6}{N}\right)^{\frac{1}{m}} = 0,735 \cdot 160 \cdot \left(\frac{5 \cdot 10^6}{2 \cdot 9,46 \cdot 10^8}\right)^{\frac{1}{5}} = 35,88 \text{ MPa} \tag{8}$$

$$\Delta\sigma_{R,red} = \Delta\sigma_R\left(\frac{25}{28}\right)^{\frac{1}{4}} = 35,88 \cdot 0,97 = 34,80 \text{ MPa} \tag{9}$$

Because $\Delta\sigma_R = 34,80 \text{ MPa} < \Delta\sigma_L = 65 \text{ MPa}$, was taken according to [7] $\Delta\sigma_R = \Delta\sigma_L 65 \text{ MPa}$.

Because structure utilization conditions were far from average so destruction consequence factor was taken as $\gamma_{fat} = 1,10$ [8].

Beam fatigue load capacity was determined as:

$$\Delta\sigma_e = 88,75 \text{ MPa} > \frac{\Delta\sigma_L}{\gamma_{fat}} = \frac{65,0}{1,1} = 59,09 \text{ MPa} \tag{10}$$

Because condition (10) was not fulfilled (fatigue load capacity was not enough) double sheet overlays were designed to reinforce flanges.

Further calculations concern the case when the vibroisolation was assumed to be broken and the repair of it would be carried out within one year period:

Damping efficacy on vertical line decreases to zero:

I range: $\sigma_{p1} = \pm 11{,}50$ MPa $\quad n_1 = 1{,}58 \cdot 10^9$ (horizontal element);

II range: $\sigma_{p2} = \pm 17{,}11$ MPa $\quad n_2 = 3{,}15 \cdot 10^7$ (vertical element).

$max\,\Delta\sigma = 34{,}22$ MPa

$$p_1(\sigma) = \frac{\Delta\sigma_1}{max\,\Delta\sigma} = \frac{18{,}62}{27{,}54} = 0{,}67;\ m = 5 \tag{11}$$

$$p_2(\sigma) = \frac{\Delta\sigma_2}{max\,\Delta\sigma} = \frac{27{,}54}{27{,}54} = 1{,}00;\ m = 5 \tag{12}$$

$$N = n_1 + n_2 = 4{,}47 \cdot 10^{10} + 9{,}46 \cdot 10^8 = 4{,}56 \cdot 10^{10}$$

$$\alpha_k = \left[\sum_i (p_i)^{m_i} \frac{n_i}{N}\right]^{\frac{1}{m}} = \left[0{,}67^5 \frac{4{,}47 \cdot 10^{10}}{4{,}56 \cdot 10^{10}} + 1{,}0^5 \frac{9{,}46 \cdot 10^8}{4{,}56 \cdot 10^{10}}\right]^{0,2} = 0{,}69 \tag{13}$$

$$\Delta\sigma_e = \alpha_K\, max\,\Delta\sigma = 0{,}69 \cdot 34{,}22 = 23{,}61 \text{ MPa}$$

$$\alpha_k = \left[\sum_i (p_i)^{m_i} \frac{n_i}{N}\right]^{\frac{1}{m}} = \left[0{,}67^5 \frac{9{,}46 \cdot 10^8}{4{,}56 \cdot 10^{10}} + 1{,}0^5 \frac{4{,}47 \cdot 10^{10}}{4{,}56 \cdot 10^{10}}\right]^{0,2} = 1{,}00$$

$$\Delta\sigma_e = \alpha_K\, max\,\Delta\sigma = 1{,}00 \cdot 34{,}22 = 34{,}22 \text{ MPa} > \Delta\sigma_L = 29{,}09 \text{ MPa}$$

Fatigue durability level was crossed. Apparently failure of any of absorber could cause additional concentration of tensions and a damage of F-M system.

Additionally the load capacity of overlay – overlay junction had to be verified.

Type of notch : 2 wholes $\phi 25$ mm in row:

a) Tension

$$\sigma_{max} = 0{,}5 \cdot 58{,}16 \frac{34{,}0 \cdot 5{,}0}{(238{,}6 + 34{,}0 \cdot 2 \cdot 5{,}0)} = 8{,}55 \text{ MPa}$$

$$\sigma_{min} \approx -8{,}55 \text{ MPa}$$

Average tensions: $\sigma_m = 0{,}00$ MPa

Period amplitude: $\sigma_a = 17{,}1$ MPa

Fatigue safety factor:

$$x_z = \frac{Z_o}{\beta \cdot \gamma \cdot \sigma_a + \sigma_m \left(\frac{2Z_o}{Z_j} - 1 \right)} \geq x_{zw} = x_1 x_2 x_3 x_4 \qquad [9]$$

where: $x_1 = 1{,}3$ – reliability factor for calculation methods of ordinary precision with known steel kind; $x_2 = 1{,}2$ – damage consequence factor; $x_3 = 1{,}1$ – material homogeneity factor for rolled sections; $x_4 = 1{,}0$ – dimension strict factor for geometrical control.

Fatigue durability value for steel S235JR by variable periods: $Z_{go} = 170$ MPa

$$Z_{ro} = 130 \text{ MPa}; \; Z_{rj} = 210 \text{ MPa} \qquad [9]$$

Tension concentration value factor:

$$\beta = \left[1 + \eta(\alpha_k - 1)\right]\beta_p, \qquad [9]$$

where: $\beta_p = 1{,}1$ – surface state factor, for S235JR ($R_r = 400$ MPa); $\eta = 0{,}66$ – material sensibility factor (on notch influence); $\alpha_k = 2{,}8$ – shape factor for tensioned flat sheet with wholes for proportion $d/t = 25/400 = 0{,}062$ [15]; γ – object size factor; foroverlay cross section area: $F = 78{,}75\,\text{cm}^2 : \gamma = 1{,}75$ hence: $\beta = \left[1 + 0{,}66(2{,}8 - 1)\right]1{,}1 = 2{,}41$

$$x_z = \frac{130}{2{,}41 \cdot 1{,}75 \cdot 17{,}10 + 0{,}0\left(\frac{2 \cdot 130}{210} - 1 \right)} = 1{,}80 \geq 1{,}3 \cdot 1{,}2 \cdot 1{,}1 \cdot 1{,}0 = 1{,}72$$

Additionally a condition below must be verified:

$$x_z = \frac{Q_r}{\beta\gamma\sigma_a + \sigma_m} \geq x_{zw}, \qquad [9]$$

where: Q_r – plasticity limit for tension; for S235JR: $Q_r = 240$ MPa [9]

hence: $x_z = \frac{240}{2{,}41 \cdot 1{,}75 \cdot 17{,}10 + 0} = 3{,}32 \geq 1{,}72$

b) Bending

$$x_z = \frac{Z_{go}}{\beta_g \gamma_g \sigma_{ag}}$$

where: $Z_{go} = 170\text{MPa};\ Z_{gj} = 300\text{MPa}$

$$\beta_g = \left[1 + \eta(\alpha_{kg} - 1)\right]\beta_p, \qquad [9]$$

where: $\beta_p = 1{,}1$, $\eta = 0{,}66$, $\alpha_{kg} = 2{,}25$, $\gamma_s = 1{,}68$

hence: $\beta_g = \left[1 + 0{,}66(2{,}25 - 1)\right]1{,}1 = 2{,}0$

$$x_{zg} = \frac{170}{2{,}0 \cdot 1{,}68 \cdot 8{,}55} = 5{,}9 > 1{,}72$$

Another condition must be verified:

$$x_z = \frac{x_{zg} x_{zr}}{\sqrt{x_{zg}^2 + x_z^2}} \geq z_{zw} \qquad (17.19)\ [9]$$

$$x_z = \frac{5{,}90 \cdot 3{,}32}{\sqrt{5{,}90^2 + 3{,}32^2}} = 2{,}89 > 1{,}72$$

Conclusion: A fatigue load capacity of weakened flat section was sufficient.

Conclusions

Conducted fatigue analysis indicated that defect of any bearing chamfering has an influence on vibration amplitude spectrum of others ones, depending on its location, increasing or decreasing the amplitude values. In detailed dynamic calculations of steel supporting structures for high-rotating machines taking this defect (possible in all bearings configuration) into consideration is very important because increasing of amplitudes directly causes decreasing of fatigue durability both of the structure and the machine. Besides, a designer should realize that there many other kinds of defects of turbogenerator that are dangerous for proper F-V-M feedback system work and should be respected in fatigue analysis.

The main element deciding about fatigue durability of all F-V-M system is vibroisolation. It was revealed that for vibroisolation efficiency η = 100 %, the estimated durability of considering node is 30 years and for η = 50 %, it diminishes to only 0,75 year. Additionally wanting to consider the chosen bearing chamfering, that time is two times less.

Excluding the vibroisolation from F-V-M system or random reduction of its efficiency causes most of all a resonance zone increase what directly brings resonance peaks interference to machine work what particularly leads into total breakdown.

Hence, it is essential for normal utilization of machine and its steel supporting structure to be inspected (especially as most trouble spots – vibration absorbers) not rare then once in six months.

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