FATIGUE DAMAGE AND FAILURE OF STEAM TURBINE ROTORS BY TORSIONAL VIBRATIONS

A. P. Bovsunovskii,[a] O. Yu. Chernousenko,[b] E. V. Shtefan,[c] and D. A. Bashta[c]

UDC 620.178; 534.1

The fatigue damage and failure of steam turbine rotors by torsional vibrations are investigated. Possible causes of the occurrence of torsional vibrations are discussed. Modeling of torsional vibrations of the shafting of a steam turbine, which occur under its operating conditions, has been performed, and the cyclic strength of the shafting under these vibrations has been estimated.

Keywords: rotor, torsional vibrations, cyclic damage, finite element modeling.

The world experience of the long-term operation of turbine units of thermal and nuclear power stations allows the conclusion to be drawn that one of the causes of accidents and catastrophic failures of turbine rotors is fatigue damage accumulation due to torsional vibrations of shafting. This is indicated, e.g., by accidents at a thermal power station in the USA (Tennessee, 1974) [1], at the state district power station-4 at Kashira in Russia (October 2002) [2] and at one of the overhauled power generating units of the Pridneprovskaya thermal power station in Ukraine (2007). In the first case, an accident resulted in the failure of a medium-low pressure rotor (Fig. 1 shows the lines along which the rotor fractured) and in the second case in the complete failure of the power generating unit No. 3 (K-300-240-1 turbine) and in the partial failure of two neighboring power generating units. Fragments of rotors were found within a radius of several hundreds of meters from the station. In the last case, the turbine had to be urgently stopped because of the occurrence of strong vibration, which prevented it from failure because of a considerable fatigue damage of the rotor, as was found out later. It was concluded from the results of investigations of these accidents that one of the main causes was fatigue of the rotor metal as a result of cyclic torsion.

Taking into account the potential hazard of this phenomenon for all steam turbines being in operation, the task was set to evaluate the degree of the fatigue damage of turbine unit shaftings due to torsional vibrations. To perform this task, it is necessary to ascertain the causes of the occurrence of torsional vibrations of shaftings during operation, to model vibrations under the action of operating load and to evaluate the fatigue properties of their materials in operation.

The ascertainment of the causes of the occurrence of shafting vibrations made it possible to model structure loading conditions which approached the operating conditions. Since no monitoring of torsional vibrations of turbine rotors is performed at present, the causes of their occurrence can only be conjunctured. The results of a number of theoretical studies [2, 3] indicate that the main cause of the occurrence of torsional vibrations of shaftings is the dynamic load acting on the turbine shaft on the turbogenerator side mainly under its abnormal operating conditions, particularly under short-circuit (SC) conditions, at the moments of connection to network with rough synchronization, because of the dynamic instability of the turbogenerator-network system and the nonuniformity of the electric field of the generator, etc.

The only experimental work [4] on this problem, known to the authors, which was carried out on a smaller turbine unit model, confirms that SC on the generator really gives rise to shaft torsion vibrations of considerable

[a]Pisarenko Institute of Problems of Strength, National Academy of Sciences of Ukraine, Kiev, Ukraine. [b]National Technical University of Ukraine "Kiev Polytechnic Institute," Kiev, Ukraine. [c]National University of Food Technologies, Kiev, Ukraine. Translated from Problemy Prochnosti, No. 1, pp. 144 – 151, January – February, 2010. Original article submitted July 29, 2009.

 0039–2316/10/4201–0108 © 2010 Springer Science + Business Media, Inc.

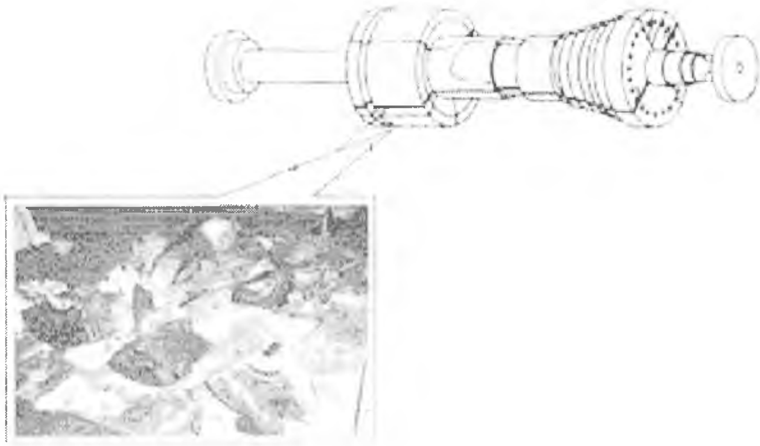

Fig. 1. Failure of the medium-low-pressure rotor of a steam turbine (Gallatin, Tennessee, USA).

amplitude. The SC duration in experiments is 0.03–0.26 s. In spite of the short abnormal action on the turbine unit shafting on the generator side, it gives rise to its torsional vibrations of amplitude which is larger by a factor of 2.8 than the nominal torque. In some works it is assumed that as a result of SC the torque amplitude may exceed the nominal torque by a factor of up to six [6].

The second point is modeling and investigation of turbine shafting vibrations under the action of operating load and determination of the most stressed zones of the shafting. The main difficulty here is that the shafting of steam turbine (e.g., K-200-130 turbine) is a complex mechanical structure, which consists of three rotors joined by couplings. Each of the rotors is a component part the high-pressure, medium-pressure and low-pressure stages, at each of which the conditions of force and thermal action differ greatly.

To investigate vibrations of such a complex mechanical system, we used a three-dimensional finite element model of shafting, which consisted of 50,000 finite elements (Fig. 2). The main attention was given to the investigation of shafting vibrations as a result of SC since this regime is, as our investigations showed, most hazardous. From the standpoint of mechanical loading, SC results in a strong reactionary-torque spike of short duration (acts in the direction opposite to the sense of rotor rotation).

The nominal torques at the high-, medium-, and low-pressure stages were taken from the turbine operating conditions: $M_h = 0.196$ MN · m, $M_m = 0.291$ MN · m, and $M_l = 0.163$ MN · m. The reactionary torque M_r acting on the turbogenerator side was taken to be 3 and 6 nominal torques (M_{nom}):

$$M_r = 3M_{nom}, \qquad M_r = 6M_{nom}, \tag{1}$$

where $M_{nom} = M_h + M_m + M_l$.

Figure 3 shows some results of calculations of the first cycles of torsional vibrations of shafting under SC. In this particular case, the reactionary torque amplitude $M_r = 3M_l$. As is evident, the force action in question is really able to excite considerable torsional vibrations of shafting with high cycle asymmetry. In this case, mainly vibrations of the first torsional mode are excited; higher vibration modes manifest themselves only slightly (in Fig. 3, the second vibration mode of small amplitude can be seen). The SC duration has a considerable effect on vibrations. In our case. SC of longer duration caused shafting vibrations of smaller amplitude. At first sight this result seems paradoxical. However, spacial investigations, which were carried out on a system with one degree of freedom using a program worked out earlier [5], show that the amplitude of vibration of the mechanical system as a result of action of SC depends not so much on its duration as on the ratio of SC duration (T_{SC}) to the natural period of the system (T_{NP}). The maximum of vibration amplitude is reached at

$$T_{SC}/T_{NP} = (0.5 + n), \tag{2}$$

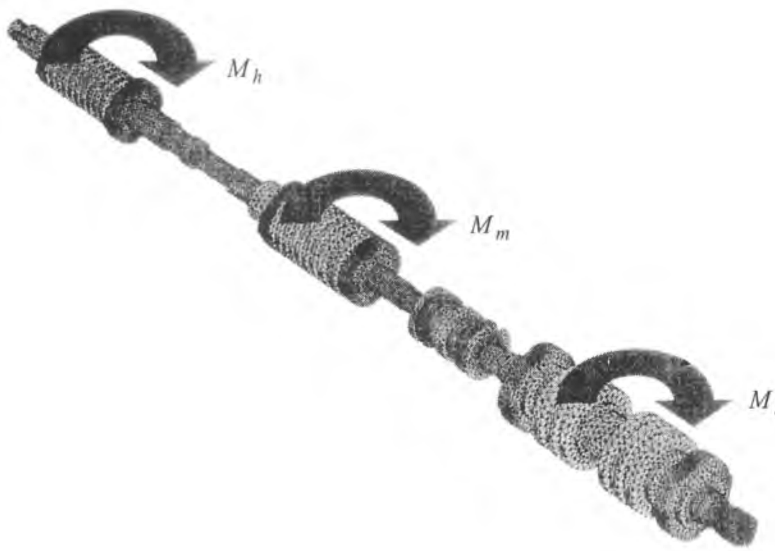

Fig. 2. Finite element model of the shafting of K-200-130 steam turbine.

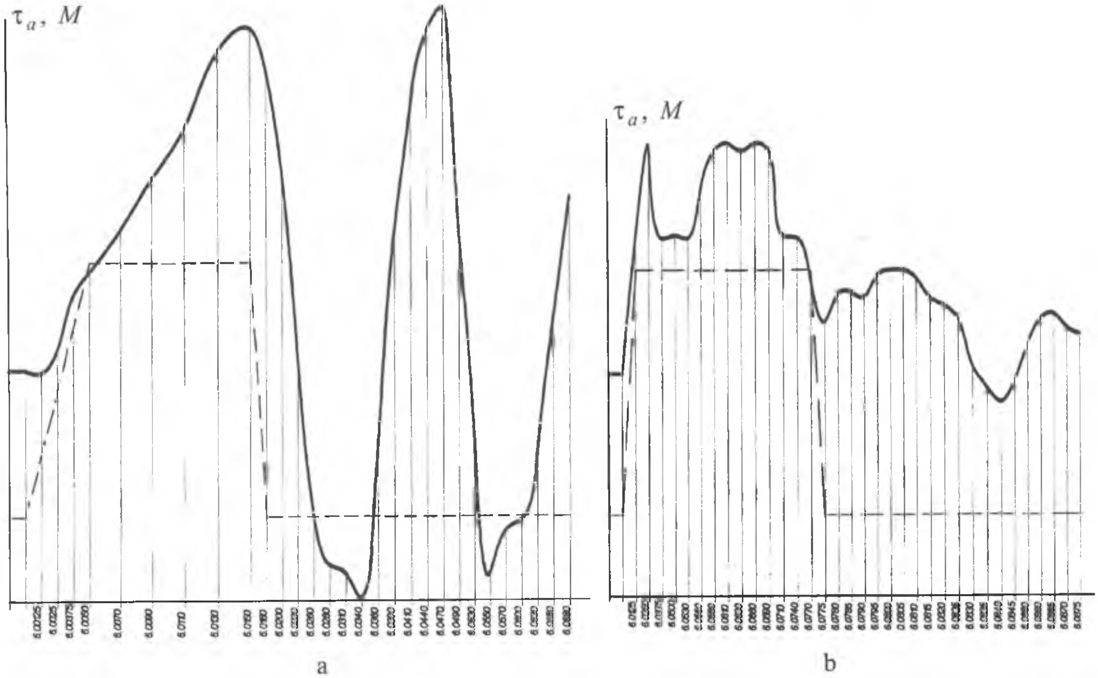

Fig. 3. Torsional vibrations (τ_a) of the shafting of K-200-130 steam turbine at the SC duration $T_{SC} = 0.015$ (a) and 0.078 s (b). (The dashed lines denote the time variation of torque M.)

and the minimum at

$$T_{SC} / T_{NP} = n, \tag{3}$$

where n is an integer.

As is seen from Fig. 3, in the former case, the ratio $T_{SC} / T_{NP} \approx 0.6$, i.e., is close to 0.5, therefore the torsional vibration amplitudes of shafting are close to the maximum possible value; in the latter case, $T_{SC} / T_{NP} \approx 2.9$, i.e., is close to 3, therefore SC is able to excite only vibrations of relatively small amplitude.

Thus, there is an additional uncertainty factor in the problem of torsional vibrations of steam turbine shafting as a result of SC. Under real conditions, the SC duration is a random and un predictable quantity. It follows that these loading conditions can both cause considerable and dangerous shafting vibrations and not excite them at all.

The third point is the estimation of the degree of fatigue damage of shaftings as a result of SC. This estimation requires, in addition to results of investigations of their torsional vibrations, data on the torsional cyclic strength of rotor steel with allowance for the influence of operation factors: temperature, cycle asymmetry, scale factor and the form of stressed state.

Since there are no such data at present, a rough estimation of them has been performed on the basis of a fatigue curve for 25Kh1MFA rotor steel in case of symmetrical tension–compression [6]. The curve was approximated by a power function of the form:

$$\sigma_a = \eta_0 + \eta_p (N)^c + \eta_u (N)^b, \tag{4}$$

where η_0, η_p, and η_u are the coefficients of the function ($\eta_0 = 175.0$, $\eta_p = 2669.4162$, and $\eta_u = 156{,}274.5216$), c and b are exponents ($c = -0.3114$ and $b = -0.8348$), and N is cycle life.

The effect of the form of stressed state on the cyclic strength of 25Kh1MFA low-alloy carbon steel was taken into account on the basis of the second and fourth theories of strength since the ratio of torsional and tension-and-compression fatigue limits ($\kappa = \tau_{-1}/\sigma_{-1}$) for carbon steels is known [7] to be best described by the second theory of strength and for alloy steels by the fourth theory of strength:

$$\tau_a = K_{sf} K_a \kappa_{\text{II-IV}} (\eta_0 + \eta_p (N)^c + \eta_u (N)^b). \tag{5}$$

The coefficients taking into account the effect of the form of stressed state on the cyclic strength of steel were taken as $\kappa_{\text{II}} = 0.794$ and $\kappa_{\text{IV}} = 0.577$. The effect of mean cycle stress on the cyclic strength of steel was taken into account with the aid of K_a coefficients, which were determined from appropriate Goodman and Herber limiting amplitude diagrams:

$$K_a = \frac{\tau_{-1} - \psi_\tau \tau_m}{\tau_{-1}}, \tag{6}$$

$$K_a = \frac{\sqrt{\tau_{-1}^2 - \kappa^2 \psi_\tau^2 \tau_m^2}}{\tau_{-1}}. \tag{7}$$

The influence of scale factor was estimated with the aid of the scale factor influence coefficient K_{sf}, which was determined from the data presented in [8]. When calculating the cyclic strength of the shaft, whose diameter was over 300 mm, the coefficient K_{sf} was taken to be 0.58.

The short-circuit conditions were modeled by reactionary torque spike of different duration and amplitude. Investigations of vibrations resulting from such loading show that the highest tangential stresses occur on the shafting of K-200-130 turbine in two its sections: in the coupling zone between the medium-pressure and low-pressure rotors and in the shafting-to-turbogenerator joint zone. The mean cycle stresses in the case of first-mode torsional vibrations of shafting $\tau_m = 82$ and 59 MPa.

The action of SC on the mechanical system gave rise to a damped vibration process. In the calculations, linear viscous damping was assumed, therefore the damped vibration process is described by an exponential function of the form:

$$\tau = \tau_a e^{-\delta f t} \sin \omega t, \tag{8}$$

where τ_a is the initial stress amplitude of the damped process, δ is a logarithmic vibration decrement, which characterizes the rate of free-vibration damping, f and ω are the frequency and circular frequency of free vibrations of shafting respectively, t is time.

TABLE 1. Degree of Cyclic Damage of the Shafting of K-200-13-3 Turbine of the Thermal Power Station at Kurakhovo in the Coupling Zone between the Medium- and Low-pressure Rotors as a Result of One SC

τ_a, MPa	δ, %	Degree of damage, Π			
		Second theory (Goodman)	Fourth theory (Goodman)	Second theory (Herber)	Fourth theory (Herber)
T_{SC} = 0.015 s (T_{SC} = 0.56 T_{NP})					
128	1.0	0.0010	0.0073	0.0002	0.0021
214	0.8	0.0123	0.0424	0.0056	0.0185
	1.0	0.0098	0.0339	0.0045	0.0148
T_{SC} = 0.079 s (T_{SC} = 2.9 T_{NP})					
38	1.0	0	0	0	0
93	1.0	0.0001	0.0019	0	0.0003

Note. All data are given for a mean cycle stress (τ_m) of 82 MPa.

The amplitude of the *i*th cycle of the damped process is determined from the formula:

$$\tau_{ai} = \tau_a e^{-\delta(1+4i)/4}. \tag{9}$$

In the calculation of the cyclic strength of the shaft we used the linear theory of damage summation (Palmgren–Miner hypothesis):

$$\sum_{i=1}^{s} \frac{n_i}{N_{fi}} = 1, \tag{10}$$

where n_i is the number of loading cycles with stress amplitude τ_{ai}, N_{fi} is cycle life in the case of cyclic loading with stress amplitude τ_{ai}, and s is the number of loading levels (blocks).

In this work, summation of damages in each deformation cycle has been effected, i.e., the number of loading blocks, s, is equal to the number of damaging loading cycles. The degree of material damage is estimated by the parameter Π:

$$\Pi = \sum_{i=1}^{s} \Pi_i, \tag{11}$$

where Π_i is the degree of material damage in one vibration cycle.

It is evident that the vibration damping level of a mechanical system has a great effect on the number of damaging loading cycles. In the general case, vibrational energy dissipation for such system as shafting under operating conditions is caused by energy loss in the material, structural energy dissipation and air damping, which is associated with interaction between vibrating elements and the steam-air medium.

Analysis of the damping properties of a number of low-alloy carbon steels shows that the logarithmic decrement of torsional vibrations of 25Kh1MFA steel can hardly be under 0.8%. Therefore, the lower limit of the damping characteristic δ was taken in calculations to be 0.8, the upper limit of this quantity, which takes into account structural energy dissipation and air damping as well, being 1%.

The degree of cyclic damage of shafting as a result of SC was calculated at two reactionary-to-nominal torque ratios ($M_r/M_{nom} = 3$ and $M_r/M_{nom} = 6$) and at two SC durations ($T_{SC} = 0.0155$ and 0.078 s).

Table 1 lists the results of calculations of the degree of cyclic damage of the most stressed shafting zone, i.e., the zone between the medium-pressure and low-pressure rotors, in the case of first-mode torsional vibrations. As can be seen, in most cases of loading considered and vibrating system parameters, a cyclic damage occurs in the shafting material, the highest level of which is observed at the SC duration $T_{SC} = 0.56 T_{NP}$ with a reactionary torque amplitude that is equal to six nominal ones. In this case, the degree of cyclic damage of rotor steel Π = 0.56–4.24% according to different theories of strength.

The spread of values of the degree of cyclic damage, determined according to different theories, reaches an order of magnitude. This spread can be appreciably reduced, and the accuracy of prediction can be thereby increased through using directly data on the torsion fatigue properties of 25Kh1MFA steel. In other cases of loading considered, the degree of cyclic damage of material is much lower; at $T_{SC} = 2.9 T_{NP}$ with a reactionary torque amplitude that is equal to three nominal ones, it does not occur.

Noteworthy is that the calculations of the degree of cyclic damage of shafting have been performed for one SC. The degree of damage during all service time of turbine unit is determined as the sum of degrees of shafting damage as a result of all SCs on turbogenerator.

CONCLUSIONS

1. Analysis of torsional vibrations of the shafting of K-200-13-3 steam turbine, which occur as a result of SC on the turbogenerator, shows that in shafting elements there may be a fatigue damage, the level of which is determined in the general case by the torsional cyclic strength of the shafting material, reactionary torque parameters and vibration damping level in mechanical system.

2. Taking into account the potential danger of the phenomenon and considerable uncertainty of loading parameters, it is necessary to organize in the future continuous monitoring of torque variation over shafting and to create on this basis an automated system for the estimation of the degree of cyclic damage of shafting by the action of operating dynamic torsional loads.

REFERENCES

1. L. D. Kramer and D. D. Randolph, "Analysis of the Tennessee valley authority, Gallatin unit N2 turbine rotor burst," in: ASME-MPC Symp. on *Creep-Fatigue Interaction* (1976), p. 1.
2. I. Sh. Zagretdinov, A. G. Kostyuk, A. D. Trukhnii, and P. R. Dolzhanskii, "Failure of the 300 MW turbine unit of the state district power station at Kashira: causes, consequences and conclusions," *Teploėnergetika*, No. 5, 5–15 (2004).
3. F. M. Detinko, G. A. Zagorodnaya, and V. M. Fastovskii, *Strength and Vibrations of Electrical Machines* [in Russian], Ėnergiya, Leningrad (1969).
4. I. A. Glebov, E. Ya. Kazovskii, Ė. E. Ostroumov, and G. E. Rubisov, "Torques on turbine unit shaft on clearing short circuits," *Ėlektrichestvo*, No. 2, 22–26 (1978).
5. A. P. Bovsunovskii, "Numerical study of vibrations of a nonlinear mechanical system simulating a cracked body," *Strength Mater.*, **31**, No. 6, 571–581 (1999).
6. *RTM 108.021.103-85. Parts of Stationary Steam Turbines. Low-Cycle Fatigue Design* [in Russian], NPO TsKTI, Moscow (1985).
7. V. T. Troshchenko and L. A. Sosnovskii, *Fatigue Resistance of Metals and Alloys. A Handbook* [in Russian], Naukova Dumka, Kiev (1987).
8. I. V. Kudryavtsev and N. E. Naumchenkov, "Fatigue strength characteristics of steel 25KhNZMFA in relation to absolute dimensions and stress concentrations," *Strength Mater.*, **10**, No. 4, 386–391 (1978).