ON STRENGTH ANALYSIS OF ROTOR COUPLINGS

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It is proved that Birger's formula can be used to analyze the compliance of a flange and to design the HPR-IPR coupling of K-300-240 KhTZ turbine-generator set.

Keywords: turbine rotor; coupling assembly; flange compliance; pressure cone; Saint-Venant principle.

The troubles faced by the national power industry are well-known. These are the approaching end of the fleet life of major equipment and, unfortunately, poorer and poorer skills of operating and maintenance personnel.

To resolve the former problem, i.e., to extend the service life and improve the reliability and efficiency of equipment, it is necessary to formulate well-grounded recommendations on how to increase the load-carrying capacity of "weak" elements, to detect incipient faults, and to enhance the vibration reliability of whole units.

Some of the technological and design solutions recently introduced to improve equipment and to enhance its reliability are doubtful, or even dangerous. This problem is many-sided and should be studied separately.

Because of the poor skills of personnel, faults that were earlier considered exotic have now become sort of ordinary. The last year saw more than one events of finding oil in the central channel of HP rotors [1]. Previously, such statistics had been acquired for decades.

In the early 2012, one of the T-250 turbines was shut down because of intensive vibration. An inspection revealed 14 (out of 16) broken bolts of the HPR-IPR coupling. The HPR-IPR coupling of K-300-240 turbines made by the Kharkov Turbine Plant (KhTZ) is known to be sensitive to such faults because of specific design features (three-bearing HPR-IPR and balancing of the axial forces by the steam counter-flow in the HPC and IPC). But for the T-250 turbine-generator sets, this case is unique. The cause is simple in our opinion: gross violation of the coupling assembly procedure. All the bolts were broken in the template portion, i.e., by the torque, because of insufficient tightening during assembly.

In [2, 3], we performed a comprehensive analysis of the load-carrying capacity of bolts for couplings of three-bearing rotors using the HPR-IPR coupling of the K-300-240 KhTZ turbine-generator set as an example and showed that the fail-safe operation of this coupling can only be ensured by meeting the very strict assembly and repair requirements. The analysis was based on Birger's formula for flange compliance:

$$\lambda_f = \frac{1}{\pi d_h E_f \tan\alpha}\left[\ln\frac{(a_1+d_h)(a_1+2l_1\tan\alpha-d_h)}{(a_1-d_h)(a_1+2l_1\tan\alpha+d_h)}+\ln\frac{(a_2+d_h)(a_2+2l_2\tan\alpha-d_h)}{(a_2-d_h)(a_2+2l_2\tan\alpha+d_h)}\right], \qquad (1)$$

where $\tan\alpha = 0.4 - 0.6$ (found experimentally); a_1 and a_2 are the diameters of the bolt underhead and nut (washer) bearing faces; l_1 and l_2 are the thicknesses of the flanges of the coupling; d_h is the bolt hole diameter; E_f is the elastic modulus of the flange material.

Equation (1) has been used to design flanges for tightness for more than 50 years now. It can be found, in a somewhat modified form, in all editions of Kostyuk's Textbook *Dynamics and Strength of Turbomachines*.

However, the numerical analysis performed in [4] using COSMOS software (an application of SolidWorks 3D CAD system) cast some doubt on Birger's formula. The correctness of the reasoning and calculations behind this point of view is highly questionable. For example, Saint Venant's principle underlying Birger's formula is, in fact, disputed in [4]. Moreover, the calculated results contradict the initial data.

It was concluded there that "the standards for the design of flange joints in mechanical engineering should be revised" because Birger's formula leads to an error of more than 48% when used to calculate the sensitivity of the bolt to the external load as the basic indicator of flange load-carrying capacity.

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This was the reason why we decided to address the strength of three-bearing rotors.

Let us first consider the theoretical aspects of the design of flange joints.

If the boundary conditions for a problem in elasticity are prescribed so as to accurately describe the real distribution of forces, then the solution may appear very complicated. Therefore, use is often made of Saint Venant's principle which simplifies the boundary conditions and leads to a solution very accurately describing the real stress field over the almost entire body [5]. "Simple solutions... may be very accurate everywhere, except for the vicinity of the boundary" [6], i.e., in our case, along the generating line of the pressure cone in the flange body.

It is this principle that was used first by Bobarykov, who proposed to model a flange by an equivalent cylinder to calculate its compliance, and then by Birger, who replaced the cylinder with a "pressure cone" to refine the solution [7].

Physically, introducing the pressure cone means using not effective stresses (Fig. 1*a*), but stresses uniformly distributed over a cross-section of the cone with half apex angle being α (Fig. 1*b*).

This simplification and its validity may be demonstrated by Saint-Venant's experiment: two equal yet opposite forces acting on a rubber bar cause only its local deformation, the major portion of the bar length remaining undeformed.

Thus, the statement made in [4] that the cone is separated from the basic metal is wrong! This does not (and cannot) occur because the pressure cone is just an assumption that allows solving the problem in a simpler way and obtaining a quite accurate solution.

The error of the solution may be controlled by varying the angle α (or, to be exact, tan α). It should also be noted that in most cases, the value of tan α and, hence, the sensitivity of the bolt to the external load on the flange joint can be determined with high accuracy only experimentally [8]. This is why the handbook [8] provides extensive experimental data, which validate Birger's formula.

The finite-element method makes it much easier to validate analytic calculations.

In this connection, we used ANSYS software:

1. to validate Birger's formula by calculating the force exerted by a bolt of real design tightening a flange that has a relatively large diameter such that the pressure cone remains within the flange, the bolt hole being real as well;

2. to validate the formula by calculating a sector of the bolted coupling of the HP and IP rotors of K-300-240 KhTZ turbine-generator set.

The models of all parts are based on manufacturer drawings (HPR B-381-20-01 (order 14013), IPR B-783-20-01, nut M-382-24-02, washer M-382-24-03, bolt M-382-24-04a). The elastic modulus of all parts $E = 210{,}000$ MPa. The initial elongation of the bolt $\Delta_b^0 = 0.2$ mm.

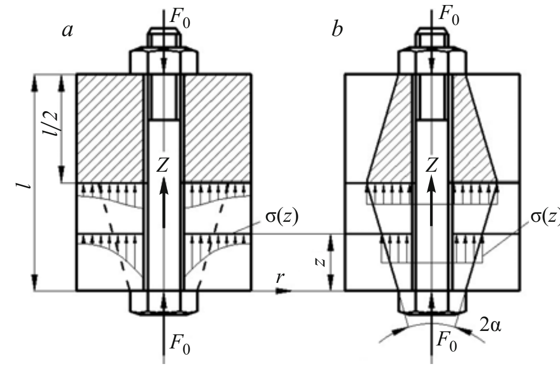

Fig. 1. Real (*a*) and Birger's (*b*) stress distribution in clamped parts.

The analytic solution predicts that the pre-tightened bolt will contract the flange by Δ_f, which can be found from the formula

$$\Delta_f = -\lambda Q_0', \tag{2}$$

where Q_0 is the equilibrium force responsible for the initial deformation of the flange and the bolt.

It is natural that the pretension of the bolt will be less by this amount:

$$\Delta_b' = \lambda_b Q_0' = \Delta_b^0 - \Delta_f = \Delta_b^0(1-\chi), \tag{3}$$

where λ_b is the compliance of the bolt (calculated by formulas from [8]); $\chi = \lambda_f/(\lambda_f + \lambda_b)$ is the sensitivity of a bolt in a flanged joint to the external load.

The value of χ depends on the value of tan α, which, as already mentioned, is found from theoretical and experimental data. As tan α is varied within the recommended range, the pretension of the bolt Δ_b varies from 0.1448 to 0.1547 mm and the coefficient χ varies from 276 to 0.226.

For the purpose of finite-element analysis, ANSYS solid models of the joints were first generated. Then they were meshed using the SOLID95 element to produce mapped meshes refined in the zones of stress concentration. For elements that do not transfer loads, free meshes were created.

The computed pretension of the bolt is 0.1455 mm ($\chi = 0.273$) in the former case (Fig. 2*a*) and 0.1420 mm ($\chi = 0.290$) in the latter case (Fig. 2*b*).

The values of bolt pretension and sensitivity differ in the two cases because the pressure cone is beyond the flange in the latter case (Fig. 2). This, naturally, somewhat reduces the stiffness of the flange (λ_f and χ increase, while the pretension of the bolt decreases).

If tan α = 0.4 (the exact value of this coefficient, as already mentioned, can be found only experimentally), then the numerical and analytic solutions are in very good agreement in case (i) and differ by 2% for bolt pretension and by 5% for bolt sensitivity in case (ii).

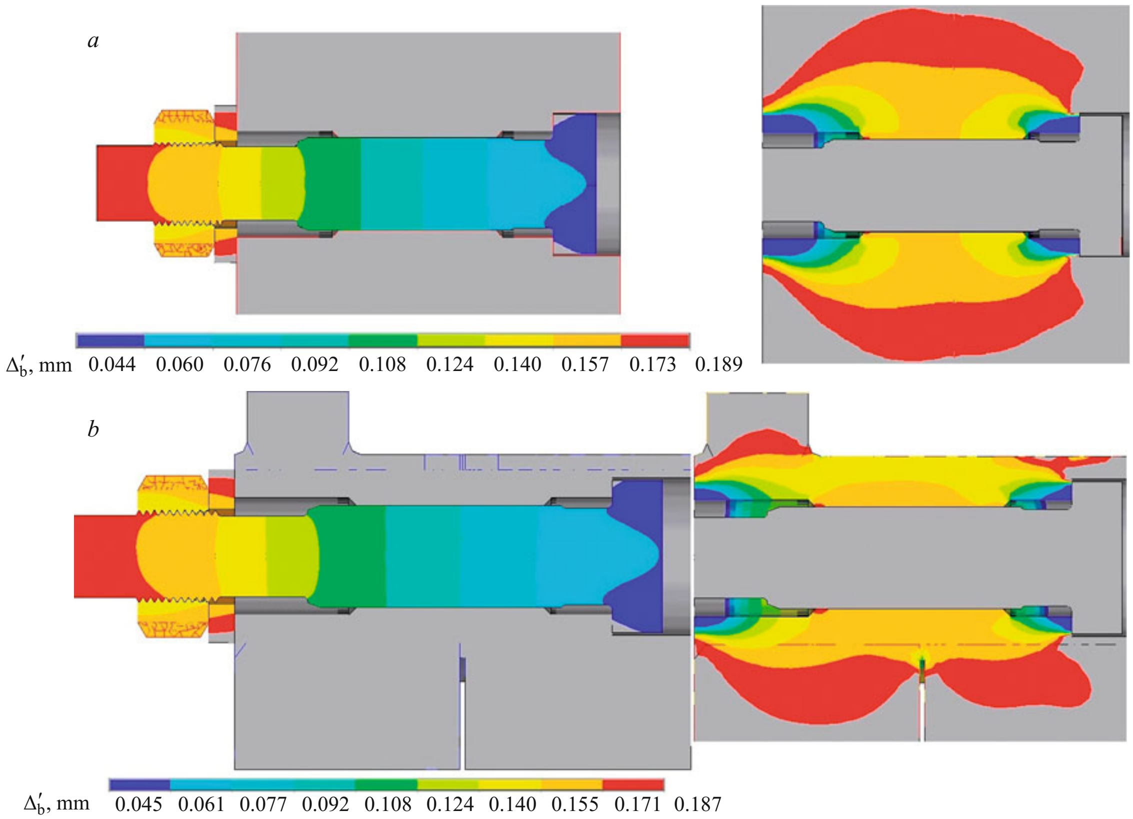

Fig. 2. Deformation of a bolt and pressure cone in a flange elongated by 0.2 mm: *a*, pressure cone is within the flange; *b*, pressure cone is beyond the flange.

The error 48% of the coefficient χ reported in [4] is due, in our opinion, to gross miscalculations. This is also confirmed by an analysis of the results of [4]. For example, the initial load on the bolt used in [4] was a concentrated force of 25 tonf, which means that the stress in the waisted portion (∅38 mm) of the bolt under its head should be about 215 MPa. In [4], however, it was stated that this stress is no higher than 65 MPa, even in the zones of stress concentration.

Incidentally, we also used $\tan\alpha = 0.4$ in [2, 3], but on the ground that with such a value, the coefficient χ is greater and, hence, the stress state of the bolts is worse.

In conclusion, we would like to point out that the HPR-IPR coupling of K-300-240 KhTZ turbine-generator set should be assembled in compliance with the requirements [9] with the amendments from [2].

CONCLUSIONS

Birger’s formula for flange compliance is based on one of the major principles of elasticity theory. The validity of this formula is supported by successful long-term use and numerous experimental data. Naturally, it can be improved by refining the boundary conditions and complicating the solution, but to say that the formula is incorrect is inadmissible.

Our numerical analysis has confirmed the validity of Birger’s formula for designing flange joints in general and the HPR-IPR coupling of K-300-240 KhTZ turbine-generator set in particular.

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