Increasing service life and reliability of centrifugal separators by applying new materials

M Ya Khabibullin
Ufa State Petroleum Technological University, Branch of the University in the City of Oktyabrsky, 54a, Devonskaya str., Oktyabrsky, Republic of Bashkortostan, 452607, Russia

E-mail: m-hab@mail.ru

Abstract. This paper analyzes stress-strain behavior and bearing capacity of a rotor in a self-discharging separator, diameter 500 mm, manufactured from parts that underwent a hardening cycle. Three variants of the rotor were studied having the same design but differing in body materials: austenitic steels 06X17H13M3-ВД, ALIII-23-43-02 and austenitic-ferritic steel 10X26H5M, ALIII-23-24. Research results allow recommending austenitic-ferritic steel for manufacture of separator rotors, as they provide necessary margin of safety for the design.

1. Introduction
The current task of increasing useful life and reliability of centrifugal separators in conditions of corrosive and erosive action of processed media with high content of chlorine and abrasive inclusions defined a shift in rotor parts manufacture, from previously used high-strength low plasticity steels of austenitic-martensitic class to corrosion-resistant high plasticity austenitic and austenitic-ferritic steels. Use of high plasticity corrosion-resistant steels with high values of fracture viscosity prevents brittle fracturing of rotors. However, such steels have a low value of yield limit, thus defining a necessity of hardening the high-load rotor body components in the centrifugal forcefield. The hardening ensures elastic operation of rotor parts and creates fields of residual strains, partially compensating stresses appearing in their operational mode.

2. Results and discussion
This paper analyzed stress-strain behavior and bearing capacity of a rotor in a self-discharging separator, diameter 500 mm, manufactured from parts that underwent a hardening cycle. Three variants of the rotor were studied having the same design, but differing in body materials: austenitic steels 06X17H13M3-ВД, ALIII-23-43-02 and austenitic-ferritic steel 10X26H5M, ALIII-23-24 (Table 1).

Stress-strain behavior of the rotor was studied numerically with a finite element method for an axisymmetric problem [1-3]. When selecting a computational pattern of the design, the following assumptions were made: the rotor is an axisymmetrical design with the axisymmetrical load; initial stress is not taken into account; loads from process medium are taken upon the rotor body (not taking into account the internal piston).

Boundary conditions for the rotor were defined from analysis of interaction between the rotor hub and the shaft (the rotor is set onto the shaft with hub fixation in the axial direction); assemblies 1
and 2 (Figure 1 a) are fixed in the axial direction; assemblies at the interior surface of the hub may move freely. Boundary conditions at the contact surfaces are represented as a line where a joint movement is assumed in the axial direction and independent movements are assumed in the radial one (threaded joint is opening under the action of centrifugal forces). The conical cover and the threaded collar were assumed rigidly connected.

Table 1. Mechanical properties of rotor steels for the same design variant

| Rotor parts          | Steel            | $\sigma_r$, MPa | $\sigma_{0.2}$, MPa | $\delta$, % | $\psi$, % | $KCU$, MJ/m$^2$ | Parts hardening mode (rotational speed), rpm |
|----------------------|------------------|----------------|---------------------|------------|-----------|----------------|-------------------------------------------|
| Base Conic cover     | 10X26H5M         | 665            | 538                 | 28         | 56        | 1              | 7200–7800                                  |
| Base Conic cover     | AL III23-43-02   | 586            | 294                 | 75         | 75        | 3.19           | 6650–7800                                  |
| Base Conic cover     | AL III23-24      | 627            | 520                 | 28         | 55        | 0.75           | 6650–7800                                  |
| Base Conic cover     | 06X17H13M3-ВД    | 578            | 294                 | 70         | 80        | 3              | 6650–7800                                  |
| Base Conic cover     | 10X26H5M         | 666            | 538                 | 36         | 61        | 1.4            | 6650–7800                                  |

Note: Rotor threaded collars in all variants are made of 30XMA steel ($\sigma_r = 930$ MPa; $\sigma_{0.2} = 735$ MPa).

3. Experimental

Numerical implementation of the task for determining stress-strain behavior of the rotor was performed with a computer. Total number of finite elements was equal to 765. As a result, a matrix of the 1920th degree was obtained for a resolving system, with a band width of 336 and one right part. The system of equations is solved with the Gauss method. Time necessary for solution and printing out the results is equal to 96 minutes 07 seconds. The calculation resulted in obtaining displacement and strain components, as well as main stress values in the nodes. Strength estimation was performed from analysis of equivalent strains determined with the 4th strength theory (Figure 1, b).

The largest strain values are found in nodes 148 and 336, amounting to 195 and 160.7 MPa, respectively. Strains in the constructive concentrator zones were determined by means of correcting calculated nominal strains $\sigma_{nom}$ taking into account experimental concentration coefficients $\alpha$, obtained by an optical polarization method [4,5]: $\sigma_m = \alpha \sigma_{nom}$.

Analysis of elastic strains has shown that maximum local strains arise at the cutout of the stopper opening in the $I$–$I$ section and in the transition zone with the radius of 5 mm, amounting to 403 and 380 MPa, respectively; it is higher than the yield strength of the steels used for manufacture of the rotor in variants II and III. In this case, formation of local zones of flowage results in redistribution of strain and elastic-plastic deformation in the concentration zones increases disproportionately to the external loads. At that, strain concentration coefficients $\alpha_\sigma$ decrease, while deformation concentration coefficients $\alpha_\varepsilon$ increase. Coefficients $\alpha_\sigma$ and $\alpha_\varepsilon$ in the elastic-plastic area may be calculated with the well-known formulas [6–9]:

$$\alpha_\sigma = 1 + \left(\alpha_{\sigma \, str} - 1\right) \frac{E_\varepsilon}{E};$$  \hspace{1cm} (1)

$$\frac{\alpha_\sigma \alpha_\varepsilon}{(\alpha_{\sigma \, str} - 1)^2} = 1,$$  \hspace{1cm} (2)

where $\alpha_{\sigma \, str}$ is a strain concentration coefficient for the elastic area; $E$, $E_\varepsilon$ are secant moduli for nominal and maximal deformation in the concentration zone, respectively (determined from material’s tensile diagram).
Such estimation of strains in the local zones allow for a more accurate account of actual behavior of the structure in the elastic-plastic area, as elastic calculations result in overestimated strains in the concentration zones. The rotors’ bearing capacity was studied at an acceleration bench VRD-1500 [10].

Limiting rotational speed was determined, causing depletion of the material’s elastic behavior and appearance of plastic deformations $f_{0,2}$, as well as depletion of the structure’s load bearing capacity $f_p$. The rotors were step-loaded. The first-stage mode was set on the basis of operating mode with account for correcting coefficient to provide equivalent load onto rotor parts from its own weight in the absence of processing medium pressure, as due to a design feature of the self-discharge separator, keeping the processing medium in the rotor during the tests was impossible without relevant hydraulic systems. Besides, plastic deformation of parts makes it impossible to ensure tightness of rotor’s seals. The equivalent rotary velocity of the rotor was calculated with the formula

$$f_e = kf_f,$$

where $k = \sqrt{1 + \frac{R\psi \rho_m}{2s\rho}}$ is a coefficient accounting for the absence of the medium; $R$ is an interior radius of the rotor; $\psi = (R^2 - R_0^2)/R^2$ is the rotor’s coefficient of charge; $R_0$ is a radius of free liquid surface; $\rho$, $\rho_m$ is the density of rotor material and medium, respectively; $s$ is the rotor wall thickness.

The following load steps were defined from the experimental data. Rotary velocity $f_{0,2}$, providing limit behavior of the rotor structure with its repeated plastic deformation (from its initial elastic state caused by prepeening at a residual deformation $\varepsilon_{res} = (0.2 - 0.5\%)$ to $\varepsilon_{res} = 0.2\%$, and rotary velocity $f_r$, causing destruction of the rotor were determined with account for specific strength of the material and Newton’s mechanical similarity criterion $Ne$ with the formulas

$$f_{0,2} = \frac{30}{\pi R} \sqrt{\frac{\sigma_{0,2}}{\rho Ne}};$$
$$f_r = \frac{30}{\pi R} \sqrt{\frac{\sigma_r}{\rho Ne}}.$$

During the step-loading, residual deformation was determined by strain-gaging; additionally, parts diameters were measured with a micrometer to an accuracy of ±0.01 mm. Resistive strain gages were glued near the concentrators’ cutouts: openings for cover stopper with respect to the base, unloading apertures, openings in the clamping rings, as well as to the butt-end of the base and the clamping ring. Results of testing the top butt-end surface of the rotor base are shown in Figure 2 and Table 2.

In all variants, depletion of the rotor’s load-bearing capacity happened due to plastic deformation of parts and disbalancing of the rotors. There were no residual deformations in the clamping ring. After the disbalancing, only rotor bases were tested, until their destruction from centrifugal inertia forces of their own mass.
Figure 1. Nodalization diagram of a rotor (a) and equivalent strain curves (b) (MPa)

Figure 2. Changes in residual deformations ε at the top butt-end surface of the rotor base during its loading: 1, 2, 3 – for variants I, II and III, respectively.
Strength margin for the failure limit state $n_{0,2}$, depletion of load-bearing capacity $n_p$ and destruction $n_r$ were calculated with the equations

$$n_{0,2} = \left(\frac{f_0}{f_e}\right)^2;$$

$$n_p = \left(\frac{f_p}{f_e}\right)^2;$$

$$n_r = \left(\frac{f_r}{f_e}\right)^2.$$

### Table 2. Results from studying the load-bearing capacity of separator rotors

| Loading mode (rotary frequency), rpm | I     | II    | III   |
|-------------------------------------|-------|-------|-------|
| $f_e$                               | 6700  | 6700  | 6700  |
| $f_{0,2}$                           | 9500  | 7125  | –     |
| $f_p$                               | 10300 | 7750  | 6750  |
| $f_r$                               | 11100 | 9140  | 7700  |
| Marginal strength coefficient       |       |       |       |
| $n_{0,2}$                           | 2.03  | 1.14  | –     |
| $n_p$                               | 2.36  | 1.34  | 1.015 |
| $n_r$                               | 2.74  | 1.86  | 1.32  |

### 4. Conclusion

Research results allow recommending austenitic-ferritic steel for manufacture of separator rotors, as they provide necessary margin of safety of the design.

### References

[1] Khabibullin M Ya and Suleimanov R I 2018 Selection of optimal design of a universal device for nonstationary pulse pumping of liquid in a reservoir pressure maintenance system *Chemical and Petroleum Engineering* **54**(3-4) 225-232 DOI: 10.1007/s10556-018-0467-2

[2] Haoran Zh, Yongtu L and Xingyuan Zh 2017 Sensitivity analysis and optimal operation control for large-scale waterflooding pipeline network of oilfield *Journal of Petroleum Science and Engineering* **154** 38-48

[3] Khabibullin M Ya, Suleimanov R I, Sidorkin D I and Arslanov I G 2017 Parameters of damping of vibrations of tubing string in the operation of bottomhole pulse devices *Chemical and Petroleum Engineering* **53**(5-6) 378-384 DOI: 10.1007/s10556-017-0350-6

[4] Korn G A and Korn T M 1984 *Mathematical Handbook for Scientists and Engineers: Definitions, Theorems, and Formulas for Reference and Review* (Moscow: Nauka)

[5] Malyarenko A M, Bogdan V A, Kotenev Yu A, Mukhametshin V Sh and Umetbaev V G 2019 Wettability and formation conditions of reservoirs *IOP Conference Series: Earth and Environmental Science* (IPDME 2019 – Int. Workshop on Innovations and Prospects of Development of Mining Machinery and Electrical Engineering) **378**(1) 012040 DOI: 10.1088/1755-1315/378/1/012040

[6] Welsh E 1987 *Borehole Coupling in Porous Media, PhD Thesis* (Colorado School of Mines, Golden, Colo)

[7] Khabibullin M Ya 2018 Impulse non-stationary flooding of hydrocarbon deposits *Advances in Engineering Research (AER)* (Int. Conf. "Actual issues of mechanical engineering" (AIME 2018)) **157** 266-271 DOI: 10.2991/aime-18.2018.51

[8] Sun W and Mun-Hong H 2017 Forecasting and uncertainty quantification for naturally fractured reservoirs using a new data-space inversion procedure *European Assoc Geoscientists & Engineers Computational geosciences 15th Conf. on the Mathematics of Oil Recovery (ECMOR)* (Amsterdam, Netherlands) **21**(5-6) 1443-1458

[9] Polyakov V N, Chizhov A P, Kotenev Yu A and Mukhametshin V Sh 2019 Results of system drilling techniques and completion of oil and gas wells *IOP Conference Series: Earth and Environmental Science* (IPDME 2019 – Int. Workshop on Innovations and Prospects of
[10] Mukhametshin V V and Kuleshova L S 2019 Prediction of production well flow rates using survey data *IOP Conference Series: Earth and Environmental Science (IPDME 2019 – Int. Workshop on Innovations and Prospects of Development of Mining Machinery and Electrical Engineering)* 378(1) 012116 DOI: 10.1088/1755-1315/378/1/012116