Alternative seating of the drum in a separation line

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Abstract

Separation processes are used in order to separate substances in homogeneous and heterogeneous substance systems. Fabric systems are mostly the product of chemical transformation obtained by chemical or biochemical reaction in some apparatus or they may occur as a natural source (air, water, mineral deposits). They are further used for cleaning e.g. desulphurisation of gases from combustion or water purification. Continuous working filters are often part of the separation line. In this case, it is a drum rotary filter. The rotating drum has a filter partition on the side, which is outside. The drum is partially immersed in a slurry vessel and the filtrate passes through the forming cake, the filtering baffle, through the individual drum segments to the axis of rotation through the distributor head outwards. Formed cake is continuously washed, blown and removed. The solids content of the slurry should be relatively low (from 3% to 5%) because thanks to that sediment often does not have to be emptied (cleaned).

1 Introduction

The drum rotary filter operates with a horizontal axis of rotation and it is filled along an axis, which is near the bottom part of the drum. Emptying is manual and the drum holder is rigid by means of the two pressed pins according to Fig. 1

During operation of the device due to external and internal factors at the place where the pivot was attached to the drum, the pins were often broken. The demandingness of the work on the replacement of the new pin contributed to the consideration of applying the pins as a mechanically dismountable screw connection. In this case, as a screw connection there were used the pre-assembled screws. The bolt would be screwed through the flange to the front of the drum, as shown in Fig 2.

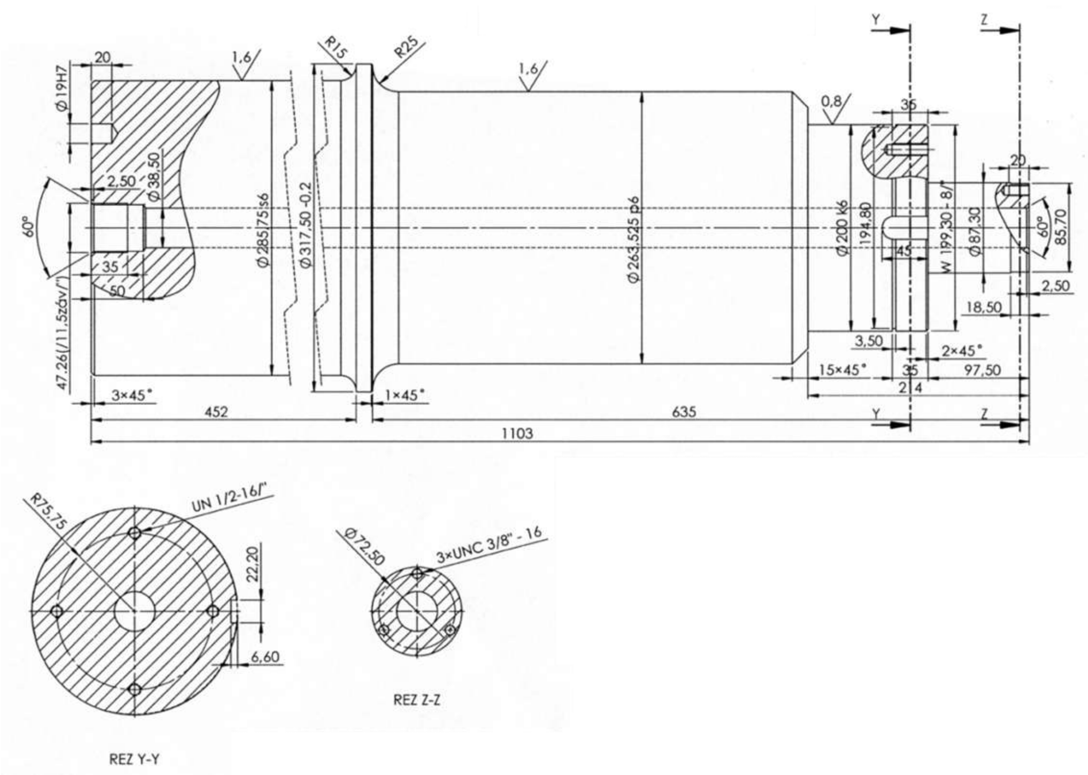

Fig. 1 Drum holder bar

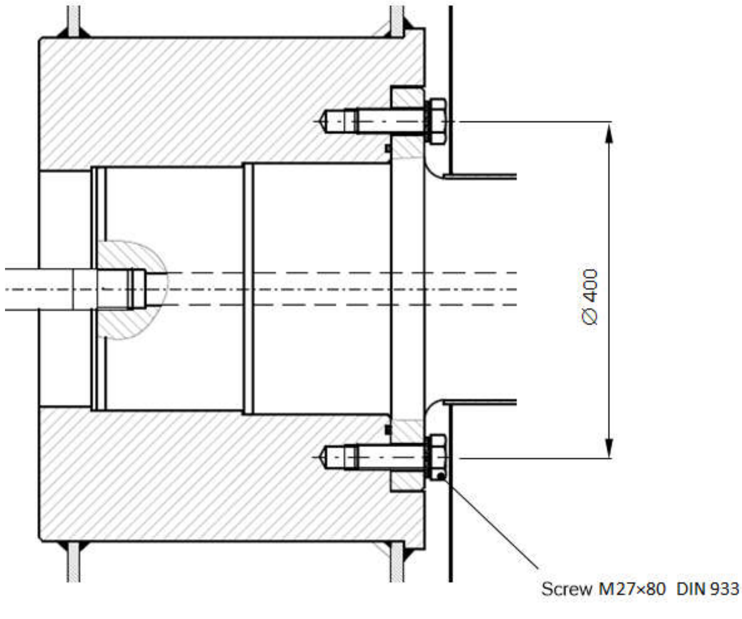

Fig. 2 Mounting of the pin on the front side of the drum

Advantage or the disadvantage of such a solution was verified by the performed calculation and FEM analysis. The creation of a computational model for FEM analysis was based on the work of various authors. Bocko et al. [1] analyze the impact of screw connections on stress and deformation as a whole. It indicates that the type of used calculation model depends on the desired calculation goal. In the case of a strength test, the joint recommends using a model to determine axial force, transverse force and bending moment, or to create a sufficiently accurate model for voltage-based evaluation. The calculation shows 5 types of calculation models. Krištofovič [2] works on the dynamic interaction of the liquid with the tank, which leads to complex motion equations and indicates the simplification of the hydrodynamic side of the problem. Koves [3] in his contribution provides a method for sizing flanges and screws. The optimization of the shape of the joint is discussed by [4], where it states that the optimized shape has a positive effect on the maximum stress which results in a more even distribution of the load along the threads in the engagement. The problem of the sensitivity of the torsion stiffness of the bolt is analyzed in [5], for example and with changes in the dynamics of the rotor when loosening the screws it is analyzed in [6]. The effects of clamping forces on fatigue life of screw connections are examined by [7].

2 Computational model - static analysis using FEM

The object of the analysis is the pin consisting of material Fe 510 (11 523.1), which has a value of f_y (Re) = 345 MPa, the strength of f_u (Rm) = 490 - 630 MPa [9].

The pin is designed in two cases of hub connection, both as a pressed joint and as a screw connection after design modifications.

The input data for the calculation of the reactions were taken from a static calculation [8] where a total load of 1 452 kN was assumed to consist of the drum's own weight and the weight of the load plus the dynamic coefficients. The test pin is located at reaction site B, Fig. 3.

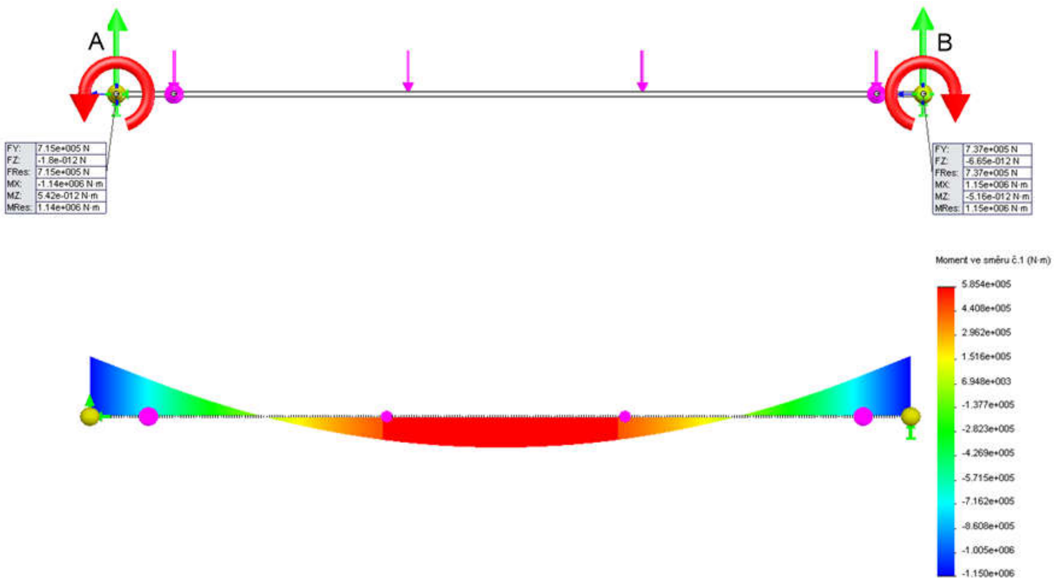

Fig. 3 The magnitude of the reactions at the locations of the bearing supports

3 Pin fastening made as a moulded joint

In Fig. 1 is shown a pin whose connection to the face of the drum is made by pressing (used housing H7 / s6). The following tolerance values [11] are used for the used storage:

$$Ø285{,}75\ H7^{+0{,}052}_{0}$$

$$Ø285{,}75\ s6^{+0{,}202}_{+0{,}170}$$

On the pressed joint with a hollow pin, according to [10], the relative pressures p0 - in the hole of the hollow pin, p1 - in the contact surface at the radius r1 and the relative pressure p2 - act on the surface of the hub at the radius r2.

Simplified 3D pin model was created for simulation purposes. The boundary conditions, applied loads, and a finite element network have been defined on the model.

The result of the simulation is the total stress distribution in the pins (Figure 4), which does not exceed the value of the tread of the material.

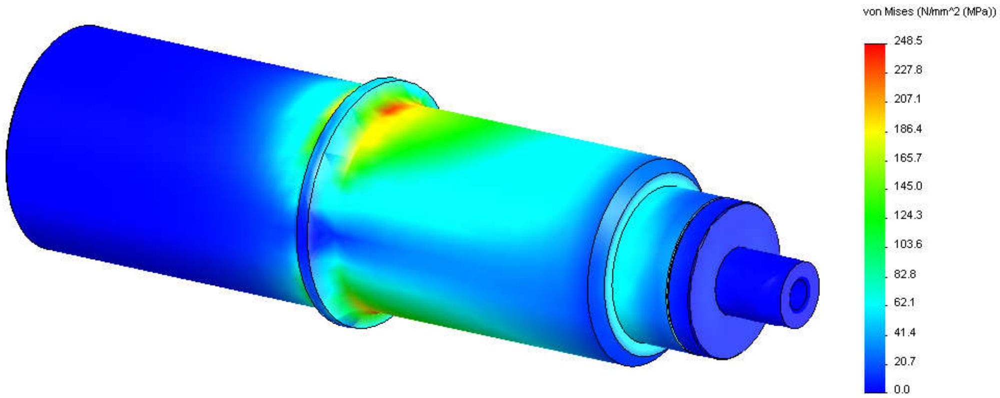

Fig. 4 Total tension distribution in the pin

4 Pin bolt made as a screw connection

A design was made on the pin, which consisted of changing the pin attachment to the front wall of the press from the pressed joint to the screw connection. The drawing of the pin adjustment is shown in Fig. 5.

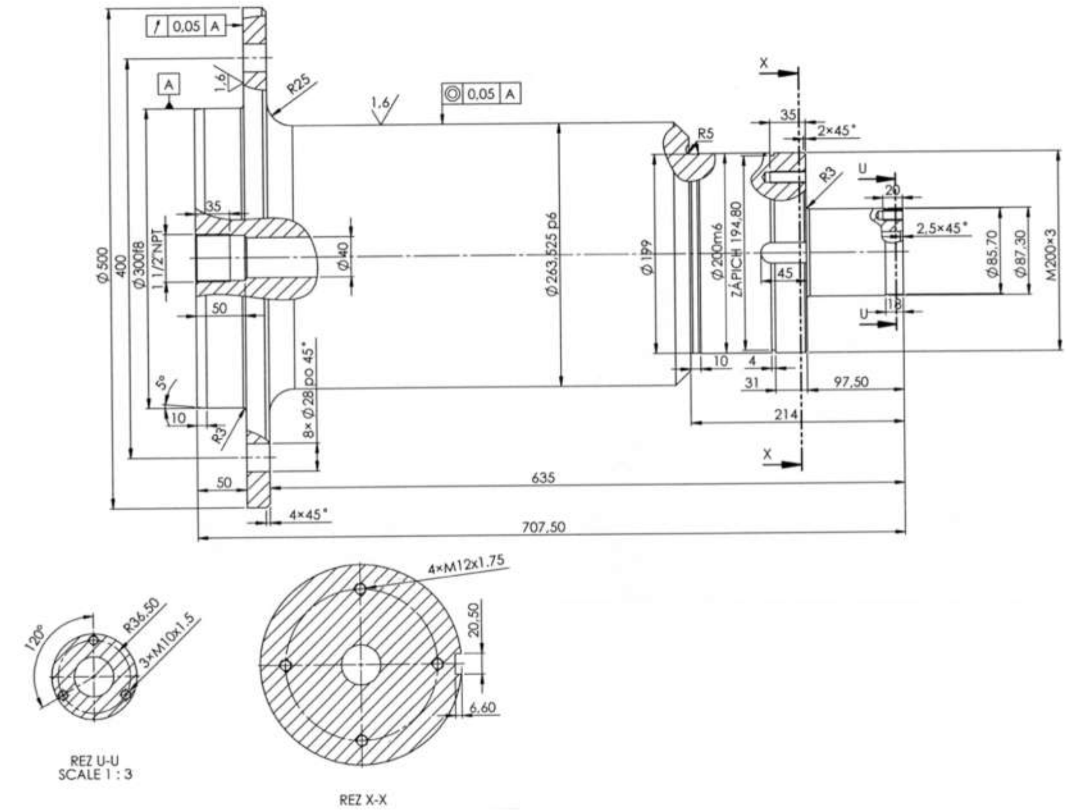

Fig. 5 Drawing of a modified pin

In this case, the pin attachment to the front of the drum is realized by means of 8 screws M27. Analogously, the pattern was applied to the model, with the marginal conditions adjusted for the new drum connection. The simulation result is shown in Fig. 6.

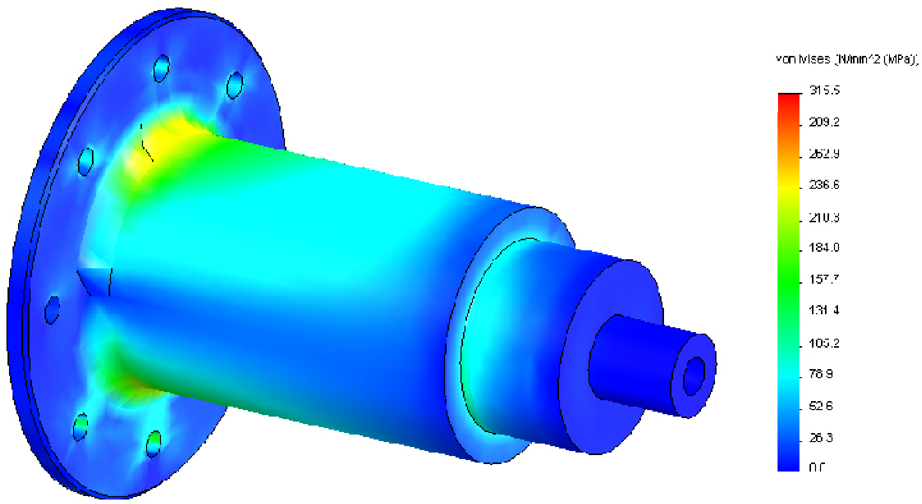

Fig. 6 Total tension distribution in the adjusted pins

From the Fig. 6 it can be seen that even in this case the maximum voltage does not exceed the value of the slump of the material.

5 Calculation of torque and shear force on screws

The pin to the drum is bolted by means of 8 screws M27 x 80 DIN 933 with the diameter 400 mm. The most unfavourable situation may occur in the case of locking of the bearing on the pin, which results in a maximum load on the pin at the point of attachment to the drum

The torque required for the analysis was calculated from the engine power and the drum speed. The engine power is P = 30 kW, drum speed is $n_b = 2\ min^{-1}$.

$$M_k = \frac{P}{2.\pi.n_b} \tag{1}$$

$$M_k = 143{,}3\ kNm$$

The cutting force applied to the bolts at the point of attachment of the pin to the drum is determined according to the following equation:

$$F_c = \frac{2.M_k}{D_S} \tag{2}$$

where D_s is the diameter of the bolt attachment.

$$F_c = 716{,}5\ kN.$$

The force acting on one screw (Figure 7) is given as a proportion of the total shear force and number of screws (3) used.

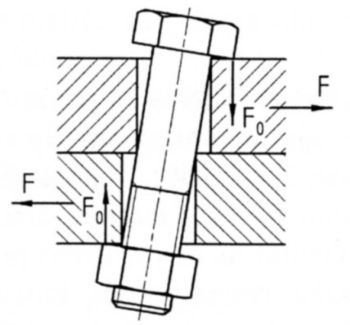

Fig. 7 Dynamically loaded coupling

F - the shear force, F_0 - the preload obtained by tightening the screw

$$F = F_c / 8 = 89{,}6\ kN. \tag{3}$$

The recommended tightening torque for screws M27 DIN 933 of the strength class 8.8 (without pre-tensioning) is, according to the manufacturer, $F_0 = 109$ Nm.

Screw parameters M27 x 80 DIN 933:

thread pitch p = 3 mm,
mean diameter $d_2 = 25{,}051$ mm,

small diameter d_3 = 23,319 mm,
cross-section of the screw core A = 459 mm^2.

According to [12], for the strength class of screws 8.8, the following nominal values of the fluctuation factor f_{yb} = 640 N/mm^2 and tensile strengths f_{ub} = 800 N/mm^2.

The resistance of the shear bolt for a single shear plane is determined according to the relationship (4) as follows:

$$F_{v,Rd} = \frac{\alpha_v \cdot f_{ub} \cdot A}{\gamma_{M2}} \tag{4}$$

where α_v = 0,6 for strength classes 4.6, 5.6 a 8.8,
γ_{M2} - is the partial confidence factor = 1,25

$F_{v,Rd}$ = 176 256 N = 176,3 kN.

For the resistance of the cutting screw, the condition must be met:

$$F \leq F_{v,Rd}$$
$$89{,}6 \text{ kN} \leq 176{,}3 \text{ kN}$$

6 Shear resistance of screw for one shear plane using the FEM

For the loads weighing on one bolt, finite element analysis for shear bolt resistance was performed (Figure 8).

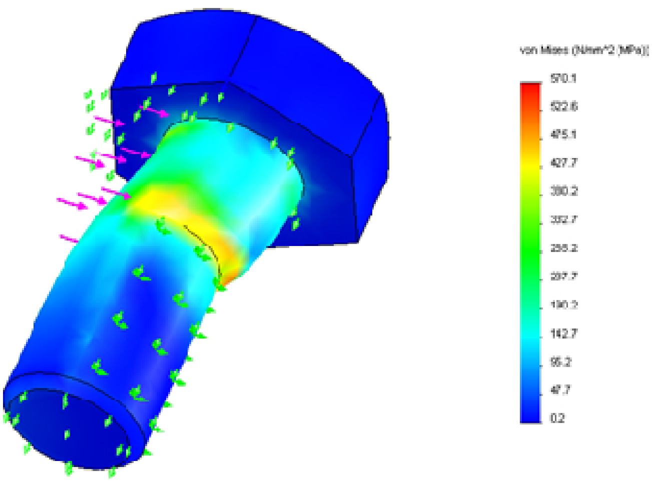

Fig. 8 Result of analysis for cutting screw resistance

According to the results of the analysis, the yield curve of the material was not exceeded, which confirmed the calculation of the cutting screw resistance described above.

The most unfavourable case of pivot loading is assumed to be an extreme load case when the bearing on the pins under investigation is completely blocked, thereby resulting in a combined load of force from the drum's own gravity to the drum load in conjunction with the maximum torque transmitted from the propulsion system.
The simulation model of this extreme load condition and the simulation results are shown in Fig. 9.

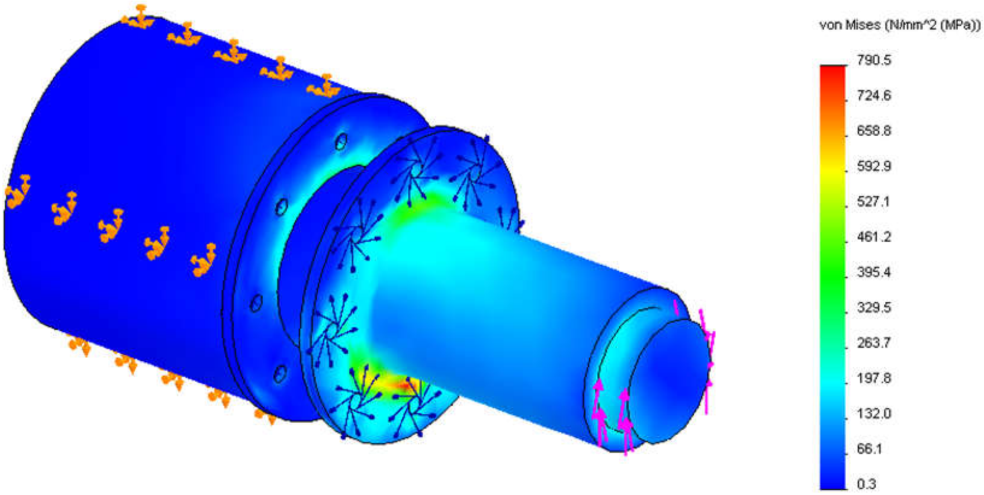

Fig. 9 Simulation results

For more accurate results, it is necessary to create a simulation of the load on the whole system, since the partial simulation for the pin only represents a rough approximation of the boundary conditions, causing uncertainties in the resulting voltage values.

7 Summary of results and conclusion

Based on the computational simulations for extreme load conditions, the analysis showed that the proposed adjustment of the pin in terms of stress is not as advantageous as, for the same load conditions, it shows 27% higher stresses but not exceeding the allowable values. However, considering the costs associated with the replacement of the damaged pin and the total line shutdown time and then despite this aspect, it is advantageous to use the proposed adjustment of the pin for the screw connection.

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