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Determination of the Cause of Failure Based on Fracture Marks on Parts of a Complicated Transmission Mechanism

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Abstract. Device failure or a failure of its part can be caused by different factors. These include for example deficiency of assembly, operation conditions, or their closely related fatigue failure. In this paper, we deal with analyses of pivots fatigue failure and causes of the occurrence of this kind of failure in the transmission mechanism of a mixing device.

Introduction

A breakdown of equipment can result in the inoperability of the machine, which will have a significant effect on production capacity and so on the economic indicators of the operator. Damage to a machine part may be caused by a variety of factors (working technology, the environment, material properties, operation period, assembly errors and the like) which influence the lifetime, not least the fatigue life of the machine. Such a case was the breakdown of the transmission mechanism of a piece of technological equipment for which it was necessary to analyze the breakdown and its possible reasons, and to propose the necessary measures to be taken.

Breakdown, analysis of the reasons for the breakdown

The breakdown appeared on the gearbox of a mixing and dosing device whose output shaft, along with rotational movement, also performs a translational movement which is secured by means of a rather complex gearbox made up of a conical differential, excentres, a connecting rod and a so-called crosser. The damage, in the form of fatigue failure, was apparent in the "crosser" by the breaking off of both pins which are located on the sides (fig. 1).

Fig. 1 Damaged crosser with broken pins and caps

The operational fractures on the crosser pin were subjected to breakage analysis, on the basis of which the following facts were arrived at:

- On the crosser's left pin were clear marks of damage in the form of a fatigue fracture (fig. 2). From the figure it is clear that the fracture came from the side of the pin's rib stiffener and its initiation was in the place of the rounded transition from the crosser body to the pin (pin diameter 107 mm);
- the fracture area had conchoidal marks from the expansion of the fracture concentrated at the initiation point. The relatively rough area of the fatigue part of the fracture witnesses to intensive widening of the crack, leading to a sudden fracture. The opposite side of the fracture area showed characteristics of shearing (fig. 3);

Fig. 2 Fracture of the left crosser pin

Fig. 3 Structure of the fracture area of the left crosser pin

- from the relation of the size of the area of the fatigue part of the fracture and of the sudden fracture it can be concluded that the amplitude of transition burdening exceeded the mean tension value. The direction of the expansion of the fatigue part and the position of the breakage, as well as the character of the surface of the fracture, suggest strain from unsymmetrical bending in the plane of the pin axis and in the movement of the crosser. The unsymmetrical bending stress results from the differentiated pin pressure as the crosser moved forwards – supplying the material and return movement;
- significant marks of expansion of the fatigue fracture were not found in the fracture area of the right pin (fig. 4). The surface remained rough, with a small facture area. The fracture here took place more quickly than on the left pin. The initiation of the fracture, the expansion direction, the fracture position and also the character of burdening were similar as in the case of the left pin.

From the inspection and analysis of the fractures of the pins on the individual sides of the crosser, there resulted the conclusion that the occurrence of the fatigue cracks were the effect of probably unbalanced burdening and to a large degree oversizing when placing the caps on the individual pins.

Fig. 4 Structure of the fracture area of the right crosser pin

When considering the complex tension state of the given part of the equipment (the crosser pins), oversizing in the fitting of the caps on the individual pins cannot be neglected. The crosser pins, to improve the frictional relations, are equipped with pin caps. These caps are heat-moulded onto the pins and after cooling provide a radial pressure on the pins which can produce a concentration of stress in the pins' translational zone. The distribution of these pressures along the diameter of the pin is axis-symmetrical. The goal of the analysis of the strain state was to judge the intensity of the effects of these strains which are superimposed on the other components of the internal forces (mainly arising from the bend stress of the crosser pins) and which can, under certain circumstances, have a large influence on the fatigue lifetime of these places.

On the equipment is used tolerance scale of pins and caps H7/r6:

pin: Ø 107 r6, which means => $\phi 107^{+0,076}_{+0,019}$

cap: Ø 107 H7, which means => $\phi 107^{+0,035}_{+0,00}$

maximal sizing: *0,076* mm

minimal sizing: *0,019* mm

For determining the normal pressure between the cap and the pin, the following simplifying conditions were introduced:

- The maximum possible sizing while observing the above-stated production tolerances was taken into consideration;
- It was assumed that the shaft is sufficiently thick and will not deform after cooling.

Under the stated assumptions the relative elongation in the cap can be expressed by the relation:

$$\varepsilon = \frac{\pi\left(D + \Delta D\right) - \pi D}{\pi D} = \frac{\Delta D}{D} = \frac{\Delta R}{R} \quad [-] \quad (1)$$

where:

ε – is relative elongation [-]

D – je the diameter of the shaft [mm]

ΔD – is the size of the tolerance [mm]

By means of Hook’s law for uniaxial strain we then get:

$$\sigma = \frac{\Delta D}{D} E \qquad \text{[MPa]} \qquad (2)$$

where:

E – is Young’s modulus of compression [MPa]

After adjustment, the normal pressure p (at maximum sizing) we can calculate with the relation:

$$p = \frac{\Delta D E t}{D^2} \qquad \text{[MPa]} \qquad (3)$$

After substitution of numerical values, the normal pressure derived from maximum sizing equalled $p = 41$ MPa. The concentration of stress in the transfer area from the pin to the thick part of the crosser was sought numerically by the finite elements method.

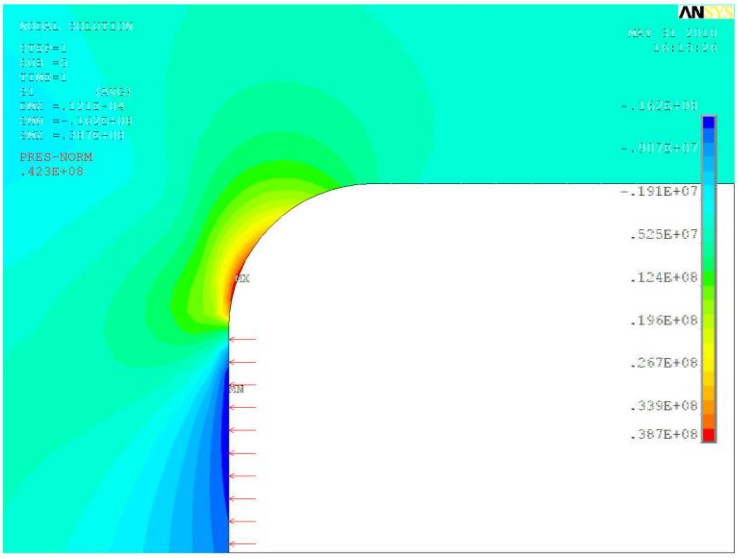

Fig. 5 Distribution of main stress on transition from pin to thick part of crosser

From the distribution of stress (fig. 5) it clearly follows that in the locality of maximum bending stress (where the fatigue crack initiated on the pins) there occurred significant stress from the moulded steel cap. Since these stresses are tensile (positive), with each rotation of the connecting rod they superimpose on the bending and increase the amplitude stress value.

Other significant factors affecting the fatigue lifetime of the equipment are assembly deviations of the excentres, looseness in the axial bearings, and similar.

As was stated above, the analyzed gearbox of the mixing equipment had to provide translational movement of the mixing worm in addition to its rotational motion. This property was achieved by the usage conception of two excentrically positioned connecting rods. If both excentres are synchronized completely accurately, they achieve closure at the very same moment, and so provide the necessary equal horizontal movement of both of the pins. Since the failures occurred just on the stated pins and after large-scale general maintenance (during the repairs the whole gearbox was disassembled and put together anew), it could be assumed that the excentres are not synchronized. If the extremes of the excentres are slewed against each other horizontal movement on the feeder pins will not be equal, and one side will be more stressed. For discovery of this fact, measuring was performed on the connecting rods wherein strain-gauge sensors were placed on the left and right sides of the upper flanges in the proximity of the crosser pins (fig. 6).

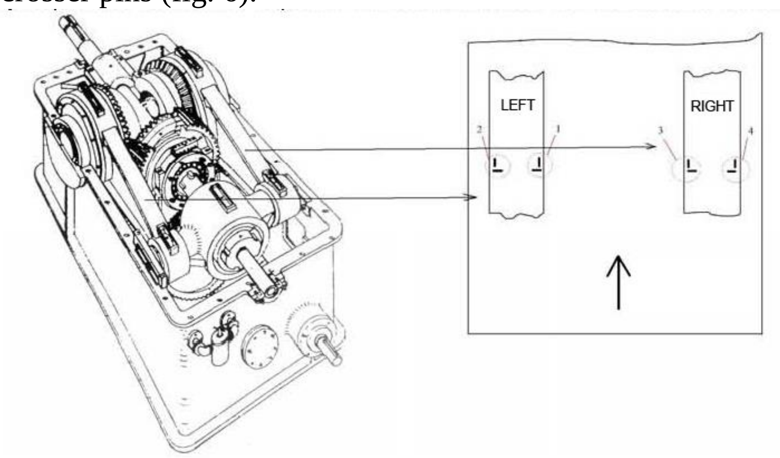

Fig. 6 Location of strain gauges on connecting rods

From the measured values it was clearly demonstrable that the assumption of inaccurate matching of the two excentres was confirmed. From the measured record (fig. 7) it was clear that in the movements of the mixing worm to the stroke (movement of the material) there was manifested on the left interior side about twice as much burdening as on the right exterior side, which clearly caused the bending stress on the crosser pins.

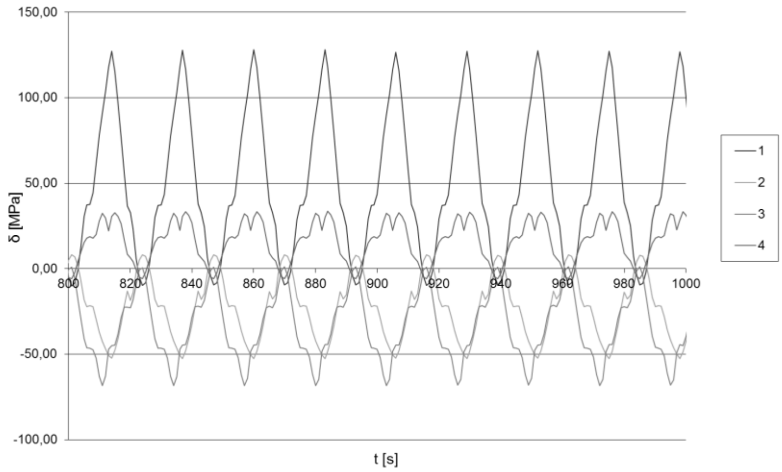

Fig. 7 Detail from the measured record

Maximum stress measured in individual places:

1. + 136,9 MPa
2. - 57,1 MPa
3. - 71,4 MPa
4. + 37,1 MPa

The stress values measured on the left connecting rod were transferred to the crosser pin and the stresses causing tensile and bending stress were separated:

$\sigma_t = (\sigma_{max} + \sigma_{min})/2 = 136{,}9 – 57{,}1 = 39{,}9\ MPa$ [MPa] (4)

$\sigma_o = (\sigma_{max} – \sigma_{min})/2 = 136{,}9 + 57{,}1 = 97\ MPa$ [MPa] (5)

The bending stress determined in this way caused the bending moment:

$$M_o = \sigma_o . W_{o,\,pin} = 13{,}178.10^6\ Nmm \qquad [\text{N.mm}] \qquad (6)$$

It is assumed that the bending across the sliding position of the junction of the connecting rod and the crosser is transferred onto the pin, where it causes the stress:

$$\sigma_{o,pin} = M_o / W_{o,pin} =$$

$$13{,}178.10^6 / 0{,}1.107^3 = 107{,}57\ MPa \qquad [\text{MPa}] \qquad (7)$$

With consideration of the concentration of the stress from the reduction of the diameter and rounding in the transition from the crosser body to the pin, which is expressed by the coefficient of the effect of the stress *α,* the value in the critical place on the pin is:

$$\sigma_{max} = \alpha .\ \sigma_{o,pin} = 1{,}9.\ 107{,}57 = 204{,}38\ MPa \qquad [\text{MPa}] \qquad (8)$$

It is also necessary to add on the additional stress from the moulded cap, *41* MPa, from which it follows that the maximum stress will be $\sigma_{max} = 245{,}38$ MPa.
The value of this stress represents the lower border of the potential stress, and its value approaches the boundary fatigue of the material used. It should however be emphasized that these calculations are only approximate and cannot exclude inaccuracies that can significantly affect the results.

Conclusion

The results of the measurements show that the premature breakdown on the crosser pins in the gearbox equipment was caused by added pressure, whose origin was not burdening from the working technology of mixing, but more probably due to geometric or assembly inaccuracies in the makeup of the gearbox equipment.

In the case of need it is possible to determine from the measured stress values on the connecting rods the load power of the movement of the mixer's shaft, which equals the differential of the stress on the tensile components in the measured places on the connecting rods.

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