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NUMERICAL AND EXPERIMENTAL MODELLING OF CARRYING ELEMENTS OF HEAVY METALLURGICAL EQUIPMENTS

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Carrying structures of heavy metallurgical equipments are during their operation often exposed to extreme loading. The short-term overloading of the structure results to high stresses in locations of their concentrations. By repeating of these phenomena is decreased the life-time of the structure and eventually this leads to local failures in their carrying elements. In the paper are on examples described advantages of using numerical and experimental methods of mechanical system modelling that is exploited for identification of overloading in carrying elements of metallurgical equipments or for detection of damage causes.

Key words: metallurgical equipments, numerical and experimental modelling, stress concentration, frame lifetime

Numeričko i eksperimentalno modeliranje nosivih elemenata teške metalurške opreme. Nosivi elementi teške metalurške opreme tijekom eksploatacije često su izloženi ekstremnim opterećenjima. Njihova kratkotrajna preopterećenja izazivaju visoka naprezanja na mjestima koncentracije. Ponavljanje ove pojave izaziva skraćenje životnog vijeka konstrukcije i moguća lokalna oštećenja nosivih elemenata. U ovom članku, na dva primjera su prikazane prednosti primjene numeričkih i eksperimentalnih metoda modeliranja mehaničkog sustava u otkrivanju preopterećenja ili uzroka oštećenja nosivih elemenata metalurške opreme.

Ključne riječi: metalurška oprema, numeričko i eksperimentalno modeliranje, koncetracija naprezanja, životni vijek konstrukcije

INTRODUCTION

Carrying structures of technological equipments in metallurgical industry are during their operation exposed to intensive force influence. In order to increase productivity and volume of production, some parts of the technological equipments are modernized after certain period of operation and this leads to increased loading of carrying elements. On the other hand, decreasing of capital expenditures necessitates to use original parts of carrying structures. It results to increasing of stress levels in carrying elements, especially in locations of their concentration. These effects decrease the time of safe operation of technological equipments (their lifetime) and in some cases they initiate damage [1, 2]. For the stress and strain analysis of such carrying structures it is suitable to use the methods of numerical and experimental modelling. These methods allow both the prediction of residual life-time of carrying structures (on the base of known loading history) and the identification of causes of possible failures. In the paper is on two examples documented advantage of combination of numerical and experimental methods for the analysis of carrying elements of technological equipments in metallurgy.

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CARRYING FRAMES OF CASTING PEDESTAL

Continuous casting of slabs is at the present time the most spread method of steel production. During this process the liquid steel is casted from the pan to the container and successively it is distributed through number of jets to continuous production. Transportation of liquid steel from the converter to the container is provided by pans and casting pedestal (Figure 1). During operation of casting pedestal the joints consisting bolts and sunk keys were released. Actual state of pedestal was analyzed by analytical, numerical and experimental methods [3]. Main supporting part of casting pedestal is the traverse consisting of two beams jointed with middle part with arms for toothing by bolts and sunk keys (Figure 2). During the operation of casting pedestal are in hinges A, B or C, D given the pans by hinging supporting baskets.

Process of casting results to the following loading combinations of traverse: hinges A, B- full pan, hinges C, D- pan with rest of steel; hinges A, B- full pan, hinges C, D- without pan; hinges A, B- without pan, hinges C, D - pan with rest of steel. Due to the symmetry of pedestal are the hinges A, B and C, D interchangeable. For the solution was considered weight force for full pan 2 800

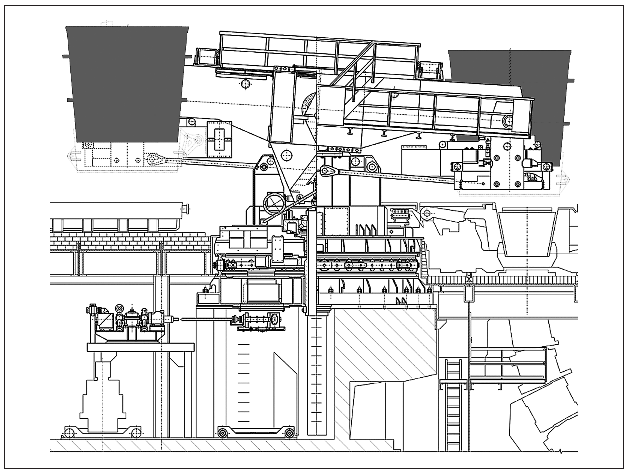

Figure 1. Casting pedestal with pans

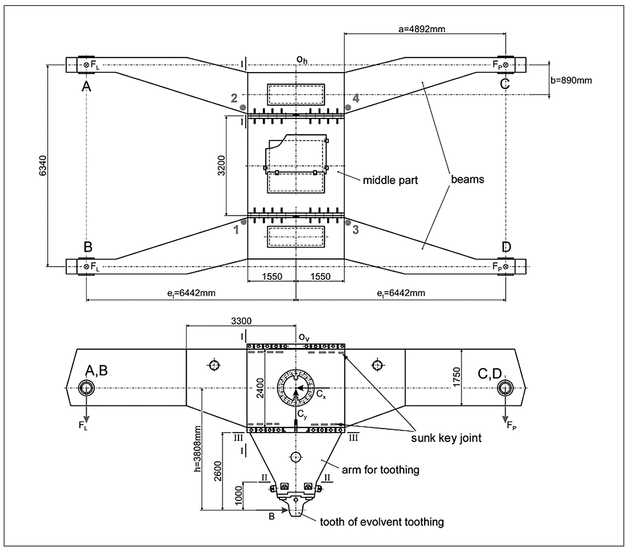

Figure 2. Traverse of casting pedestal

kN, weight force for empty pan 700 kN and weight force for pan with the rest of steel 1 200 kN.

Beams of traverse are jointed with the middle part by bolts and sunk keys (Figure 2), which are during operation of casting pedestal (as a result of alternating and irregular loading of its arms) released. Model of traverse (with respect to the symmetry was considered only one half – beam and middle part with arm of toothing) is given in Figure 3. Computation of traverse was realized for all load cases given above. In Figure 4 is shown the field of principal stresses σ_1 in location of bolts and sunk keys for loading of traverse by one full pan. It is seen shift of traverse beam in with respect to the middle part of joint.

For the strain gage measurement was used apparatus SPIDER 8. Strain gages were applied in ten locations of carrying structure of casting pedestal.

Further strain gages were applied on arms of toothing and on hinging supporting baskets. Measurement of time dependent strains was realized for simulated regimes and for operation of casting pedestal. Figure 5 shows stresses for one measurement during operation.

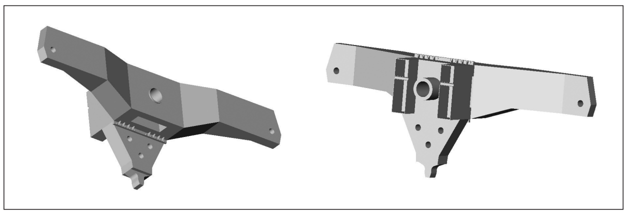

Figure 3. Computational model of casting pedestal traverse

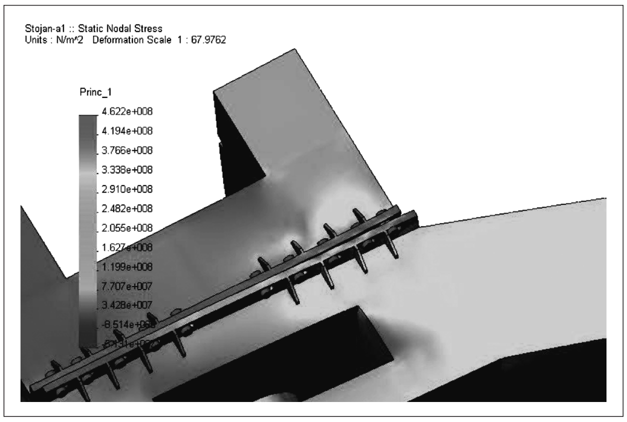

Figure 4. Principal stresses σ_1

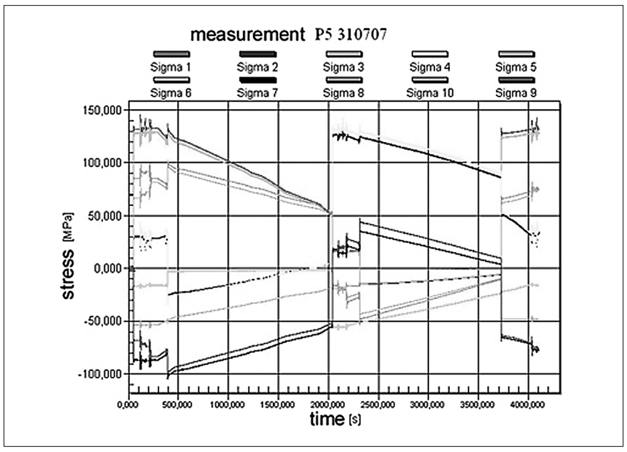

Figure 5. Time dependent stresses during operation

EXHAUST RADIAL FAN OF CONVERTOR

During the operation two of the same type radial fans one of them crashed. Numerical and experimental analysis have been realized for frame elements of crashed radial fan but also for second one that operated normally. The crashed fan was designed to exhauast convertor gas during operation of the convertor. It was a radial fan with a diameter of the rotation wheel of approximately 3 m, supported on two bearings and driven by an electric engine trouhgh the clutch (Figure 6). After several months´ operation with rotation frequency from 600 to 1400 rpm was the rotor damaged and consequently was damaged also whole fan. Figure 7 shows the crashed rotor and deformed covering sheet of the fan´s housing. On the basis of visual analysis of broken surfaces was found out that the first damaged element of fan was its rotating wheel. In the paper are given the results of nu-

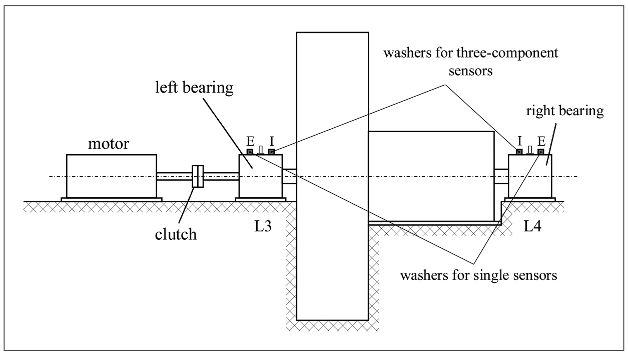

Figure 6. Radial fan

Figure 7. Damaged rotating wheel and covering steel sheet

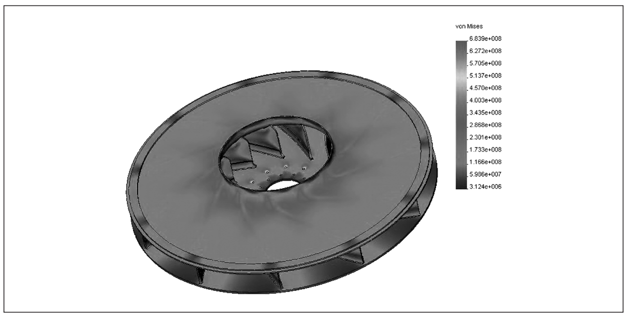

Figure 8. Strains and equivalent stresses in rotating wheel

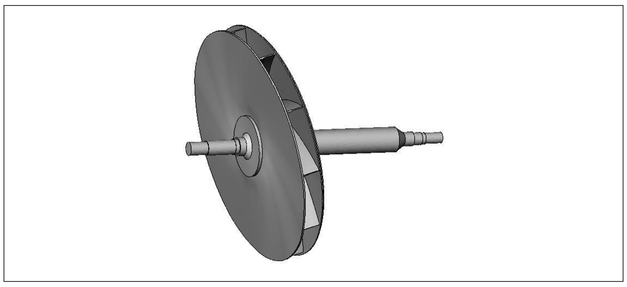

Figure 9. The first eigenshape of rotating wheel with shaft

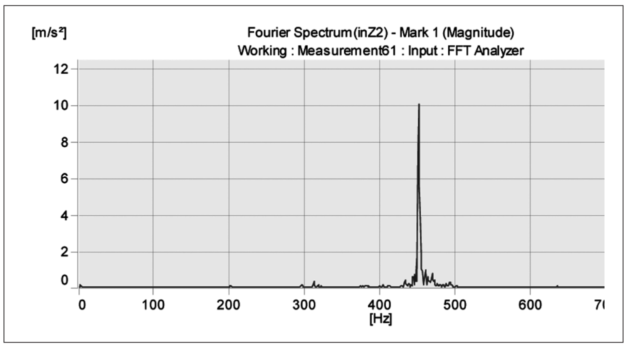

Figure 10. Accelerations dependent on frequencies bearing L4, 1100 rpm

merical and experimental analysis with the aim to determine the reasons of failure of individual parts of radial fan.

The finite element method was used for determination of stresses in the fan´s of rotating wheel as well as for computation of its eigenshapes and eigenfrequencies. Figure 8 shows equivalent stresses in rotating wheel for frequency 1400 rpm. The shaft was modelled for computation of eigenshapes and eigenfrequencies of rotating wheel. Figure 9 shows first eigenshape of vibration.

Vibrodiagnostic measurements were realized during operation of the fan of the same type and used for the same purposes, as the damaged one. There were applied sensors of acceleration of the type 4506 B.

Location of supports for the fixation of sensors is shown in Figure 6. The sensors had their x axes in axial direction of bearings, axes y in horizontal radial direction and axes z in vertical radial direction of bearings. Maximal values of acceleration amplitudes were detected in the right bearing of fan (L4) during normal operation (1100 rpm). Figure 10 shows the charts of acceleration in z direction with respect to frequency for such a measurement. Experiences has shown that for broad band vibration is the best measure of dangerousness and harmfulness of vibration the effective value of vibration velocity v_{ef} because to the certain value of velocity corresponds certain value of energy [4, 5].

Standard STN ISO 10816-1 [6] categorizes fan in question to Class III with the following band limits: Level A – $v_{ef} \leq 1{,}8$ mm/s (new machines); Level B – $v_{ef} =$ 2,81 – 4,5 mm/s (common operation without time limit); Level C – v_{ef} = 7,1 – 11,2 mm/s (not suitable for long-time operation – limited operation); Level D – $v_{ef} \geq$ 18 mm/s (not allowed values). Maximal acceleration and velocity amplitudes were, according to the measurements, for frequencies 450 – 460 Hz, maximal effective value of velocity reached 4,5 mm/s^2 and this shift operation to the boundaries of the Level C that defines limited operation of fan.

CONCLUSION

In the paper is on two examples documented advantage of combination of numerical and experimental methods for the analysis of carrying elements of technological equipments in metallurgy.

From the analysis of supporing parts of casting pedestal concluded that joint of traverse beam with the middle part is problematic. Sunk keys transfer 80 % of loading. Their releasing follows to danger of failure of the whole joint. Analysis by the finite element method proves that the most dangerous loading case is during manipulation with one full pan. In order to prevent sunk keys releasing during operation of casting pedestal it was suggested to fix the sunk keys by bolts with lock. At the same time it was recommended to increase prestress in bolted joints (with possible increased diameter of bolts). For the increasing of life time of pedestal was recommended to exclude casting pedestal from regime – one side of pedestal with full pan, second one without pan.

In order to find reason of radial fan failure there was accomplished analysis of eigenshapes, corresponding eigenfrequen-cies and stresses by the finite element method as well as by measurements of accelerations in ventilator bearings. For certain rotation frequencies and certain operation regimes was found out that velocity amplitudes reach such values that according to theory of effective velocities and thus massiveness of vibration, ventilator cannot work in unlimited operation although he was constructed for incessant operation.

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