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THE ANALYSIS OF SHAFT BREAKS ON ELECTRIC MOTORS COUPLED WITH RECIPROCATING COMPRESSORS

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ABSTRACT

The paper presents the analysis of repeated shaft breaks on 560 kW electric motors coupled with reciprocating hydrogen compressors in an oil refinery plant. Asynchronous electric motors and reciprocating compressors were coupled together using elastic couplings. Torsional vibration calculations, performed during design, did not indicate any problems in compressor set operation. Design changes performed after initial breaks didn't help to keep the plant in safe operation. The paper presents the performed systematic analysis, showing the pitfalls encountered in torsional vibration calculations performed in usual way as used by manufacturer. Detailed analysis has indicated real causes of the breaks, not only one but few of them, in very intriguing combination which had caused serious damages. The analysis was later applied to several manufacturers' solution proposals, helping to find the acceptable final solution. The paper indicates recommendations in performing torsional vibration analysis on such compressor drives.

1. INTRODUCTION

In the new hydrodesulphurization plant of the oil refinery a series of 8 shaft breaks on 560 kW electric motors coupled with reciprocating hydrogen compressors happened in a short period of only 4 years. Some of them occured in repeated manner on same compressors, as shown on the Figure 1. First 4 breaks happened during the warranty period and the manufacturer tried to avoid future breaks by changing the shaft design, but without any success.

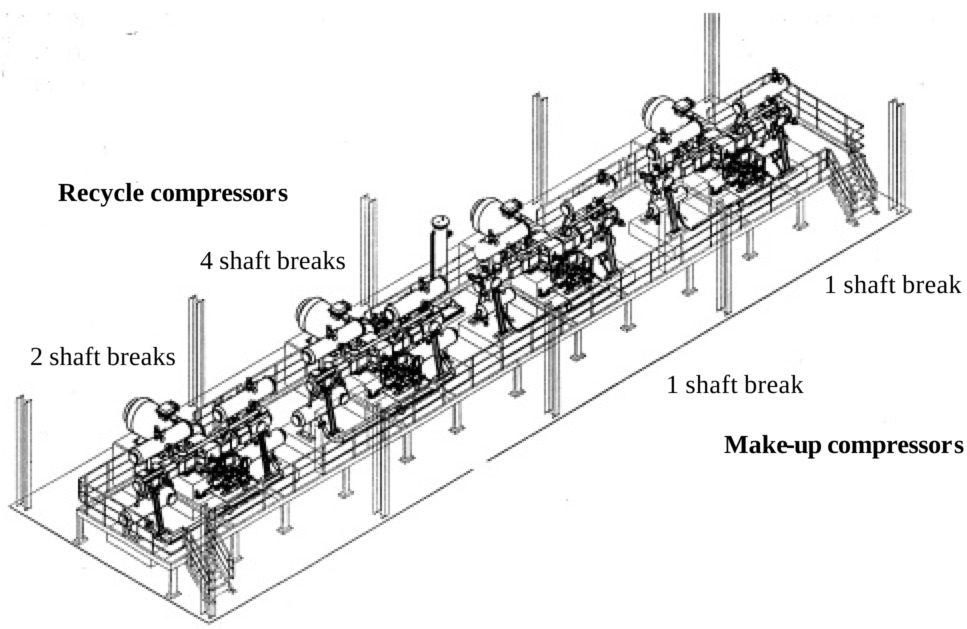

Figure 1 Shaft breaks on the compressor drives on the HDS plant in an oil refinery

Those shaft breaks have caused serious problems, namely increased reparation costs, cease of production and increased risk of explosion and fire. The working fluid is high purity hydrogen (more than 70%) with lethal concentration of hydrogen disulfide. After a serious damage caused by one of the shaft breaks and increased risk of future severe damages, the University of Rijeka has been engaged in solving the problem. At the same time the plant operational personnel has posed special attention to compressors operation regarding very high risk of accidents initiated by unexpected shaft breaks.

In preliminary analysis we noticed that technical documentation didn't contain any evidence of torsional vibrations calculation, what is normal in situations of coupled reciprocating piston engines and other rotating machinery. Asynchronous electric motor was coupled to the compressor flywheel using elastic coupling (Figure 2). The compressor was with 2 opposed cylinders, double acting reciprocating type, operating at 600 rpm.

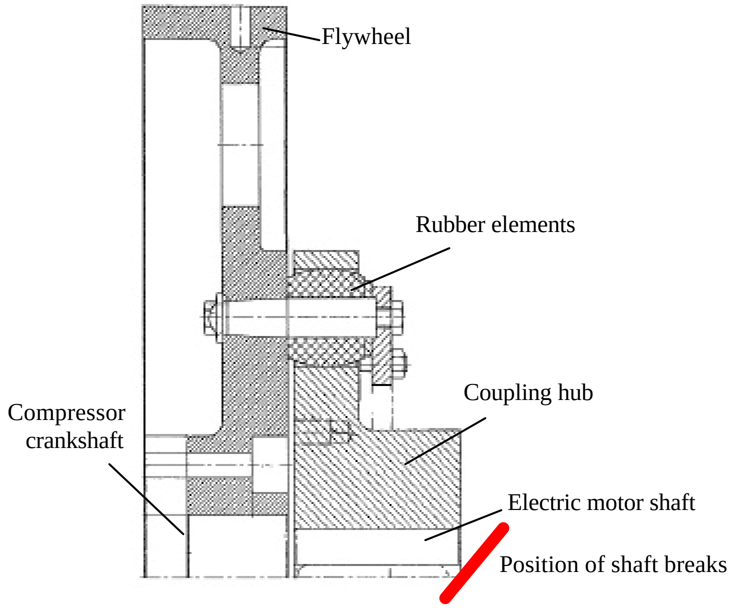

Figure 2 The original elastic coupling as delivered with compressors

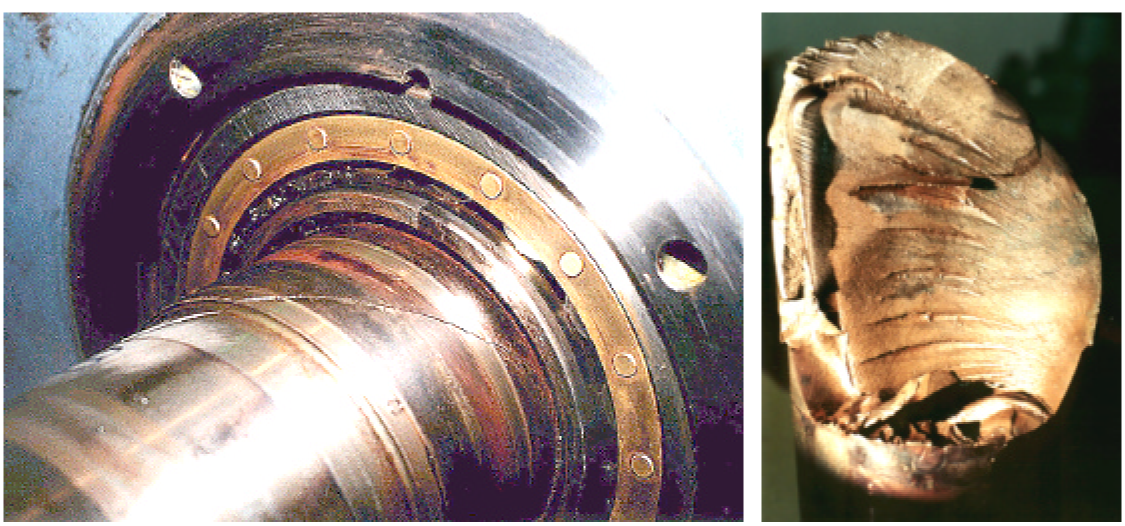

Figure 3 Examples of shaft breaks

As the driving end of the electric motor shaft was the weakest part of the system, all the breaks happened practically on the same place, all of them in usual torsional manner (Figure 3). After the first analysis three main problem areas were established: the electric motor shaft material properties, the elastic coupling and possible liquid accumulation. Later it was found that the elastic coupling has been the main source of related problems, and it is the matter of the presented paper.

2. TORSIONAL VIBRATION CALCULATION

Properties of torsional rotating system determine the natural frequencies of the system. Each operation in the vicinity of those frequencies is leading to the danger of resonant vibrations and material damage. The usual torsional vibrations calculation covers the natural frequencies determination and the control of the system vibrations and shafting stresses in normal operation. This analysis is in usually performed using the Fourier analysis on complex perturbation to find out perturbing harmonics, continuing with solving the system responses for each harmonics and synthesizing the complete system response regarding the complex perturbation. This analysis is usually restricted to the steady state condition of the system operation. Such an analysis was performed by the compressor manufacturer. The results were all acceptable, but some of them doubtful.

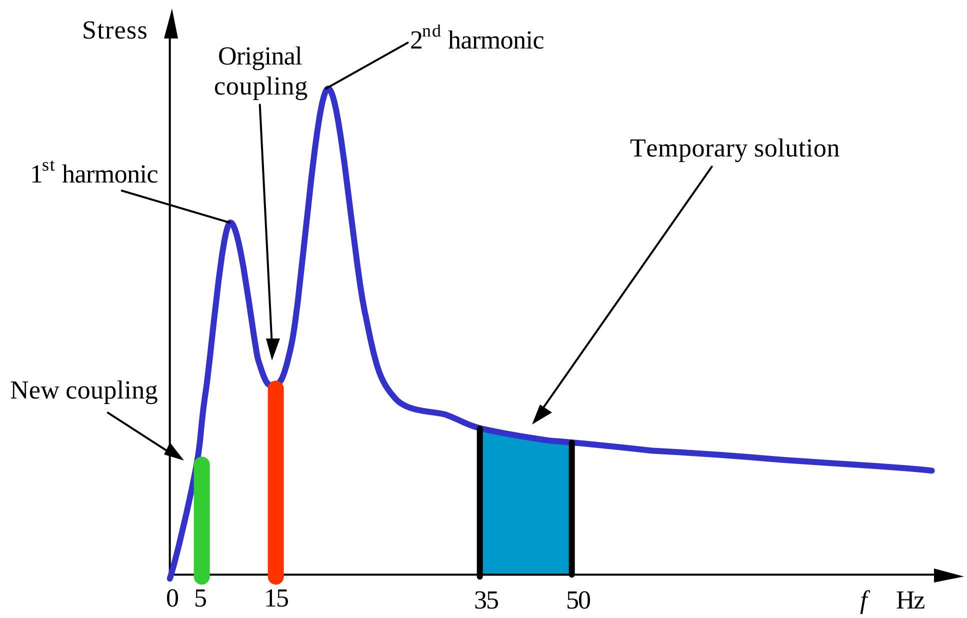

Figure 4 The shaft stress and natural frequency for system operation using various couplings

As the compressor was double acting, the second order perturbing harmonic was very evident, overriding other harmonics (at 20 Hz on the Figure 4). Placement of the first natural frequency using the original elastic coupling was arguable, while it was placed between two strong perturbing harmonics, the 1st and 2nd one. Any deviation in coupling properties is leading to the resonance either with the 1st or with the 2nd harmonic. Aside of that, the torsional stiffness of the coupling was highly nonlinear and many problems have arisen from that fact, while natural frequency depended on the vibration amplitude. Rubber elements used in the original coupling were of various properties. Examination of the rubber has evidenced the hardness deviations of $\pm$ 25% among the elements of the same set. Some calculations have shown that at one examined set of rubber elements the first natural frequency was very close to the 2nd perturbing harmonic.

Many of the rubber elements have been smashed after a short period, evidencing high shocks in operation. In quest for the cause of such shocks, attention was at first paid to possible operation in the vicinity of the resonance, but calculations didn't show that to be harmful in such a dramatic way. The answer has been found in the maintenance handbook for the applied elastic coupling. The criterion for the change of rubber elements was the measured angle of free rotation between two coupling hubs, indicating that there is clearance between elements in the torsional system. In the applied coupling, this clearance was found even with new rubber elements.

Once more, usual torsional vibration calculations have evidenced that the system will operate without problems. The reality was different. Pitfalls of the usual torsional vibration calculations are the following:

- there is no possibility to calculate the vibrations in the transient conditions (compressor start-up, shut-down, electric motor short circuit event etc.),
- there is no possibility to take into consideration possible discontinuities (clearance between loose parts etc.),
- there is very hard task to take in the consideration the nonlinearities in the coupling torsional stiffness,
- the driving torque of the electric motor is normally taken as the constant value (without any torque oscillations or changes due to the variable slip between the rotation speed and the frequency of the electromagnetic field caused by the rotor torsional vibrations),
- the momentum of inertia of the crank mechanism is often taken as medium value, rather than the angle dependent.

Above mentioned restrictions of the usual torsional vibration calculations were reason for authors to find out solutions which will be closer to the reality. All these mentioned restrictions in simulating the real conditions can be avoided in very simple and effective way using numerical simulations applied to the system motion differential equations, rather than by finding out analytical solutions. The system of the torsional system motion equations, written in concise form is:

$$[J]\{\ddot{j}\}+[D]\{\dot{j}\}+[K]\{j\}=\{T(j)\} \qquad (1)$$

where $[\ ]$ is the matrix, and $\{\ \}$ is vector notation, *J* is the momentum of inertia, *D* is viscous damping, *K* is torsional stiffness, *T* is the perturbing torque, j is the crank angle, $\dot{j}$ and $\ddot{j}$ are the first and second time derivatives. The same form is the basis also for the usual torsional vibration calculus.

The differential equation system (1) may be solved using combined single and double numerical integration marching through the time. A lot of developed integration procedures is available. The simplest integration method by Euler is also applicable in integrating these equations. The initial conditions must be set at the start of the integration process. Through the time there is the possibility to change the boundary conditions or any of the equations parameters (for example stiffness, moments of inertia, discontinuities etc.) between the time steps during calculation, regarding the system status or other imposed conditions.

To keep the evidence even on the smallest scale vibrations, the time step must be kept sufficiently small (10^{-5} s). The calculation is very fast regardless the used time step. A system shut-down, which lasts for the analyzed case in the reality about 30 seconds, can be numerically simulated using usual personal computer (Pentium 3) in less than 5 minutes, giving even the finest vibration details. There is no possibility to do this calculation using the usual torsional vibration calculations.

Numerical simulations of the steady state operation are very fast. They are conducted in the same manner as the transient operation simulations, leaving the damping to stabilize the system in steady state operation after the initialization from the assumed initial conditions.

3. NUMERICAL ANALYSIS OF THE ORIGINAL DESIGN

The first analysis was performed on the original design as delivered by the manufacturer. Just to find out the differences between the detailed and the usual calculation, there was assumed that the elastic coupling has no clearance between the system parts (as it was assumed by the manufacturer). The detailed simulation has taken into account the nonlinear stiffness of the rubber elements, the variation of the electric motor torque regarding the instantaneous rotation speed, the variations of the crank mechanism momentum of inertia as the function of the crank angle and instantaneous compressor torque as the function of the in-cylinder processes (without the use of perturbation harmonics).

The results of this simulation are presented in the Figure 5. The analysis has covered the start-up of the compressor set, the short operation in no-load condition and the application of the full load after 3 seconds from the start. The torque variations T_{EM} of the electric motor due to the variations of the rotor instantaneous speed n_{EM}. These rotor speed variations are presented at the topmost line in the diagram. Stresses in the electric motor shaft t_{EMS} and the compressor crankshaft t_{CS} are presented in the lower part of the diagram.

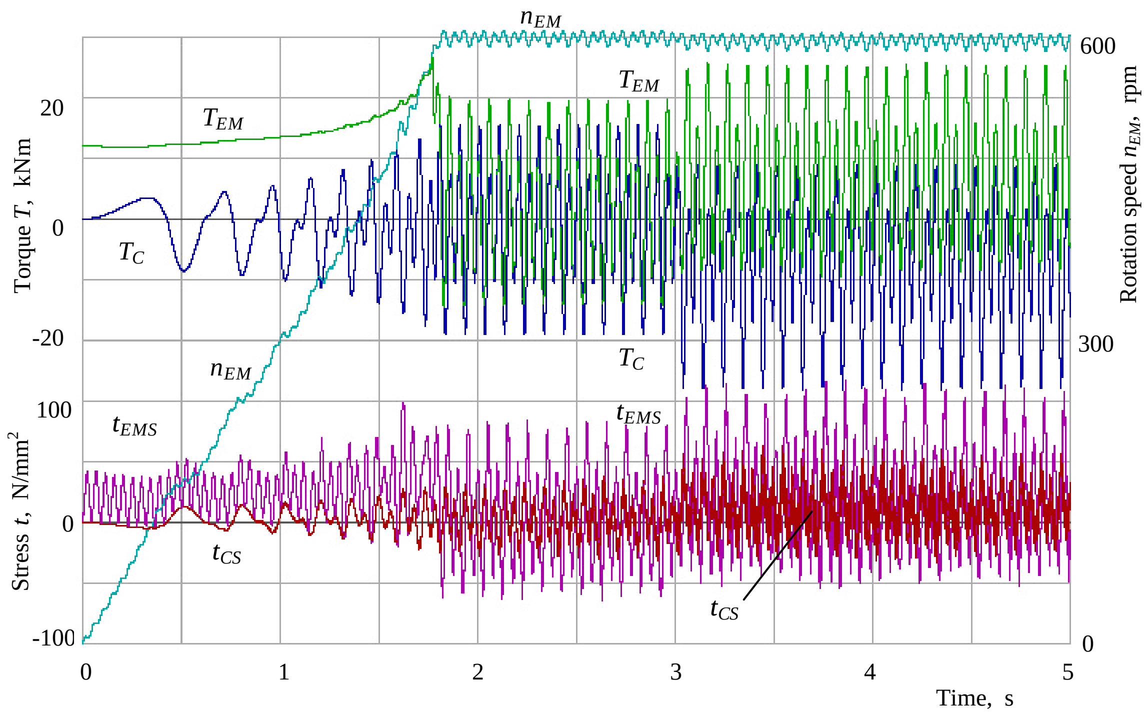

Figure 5 Results of the numerical simulations for the original design with elastic coupling without clearance between coupling parts

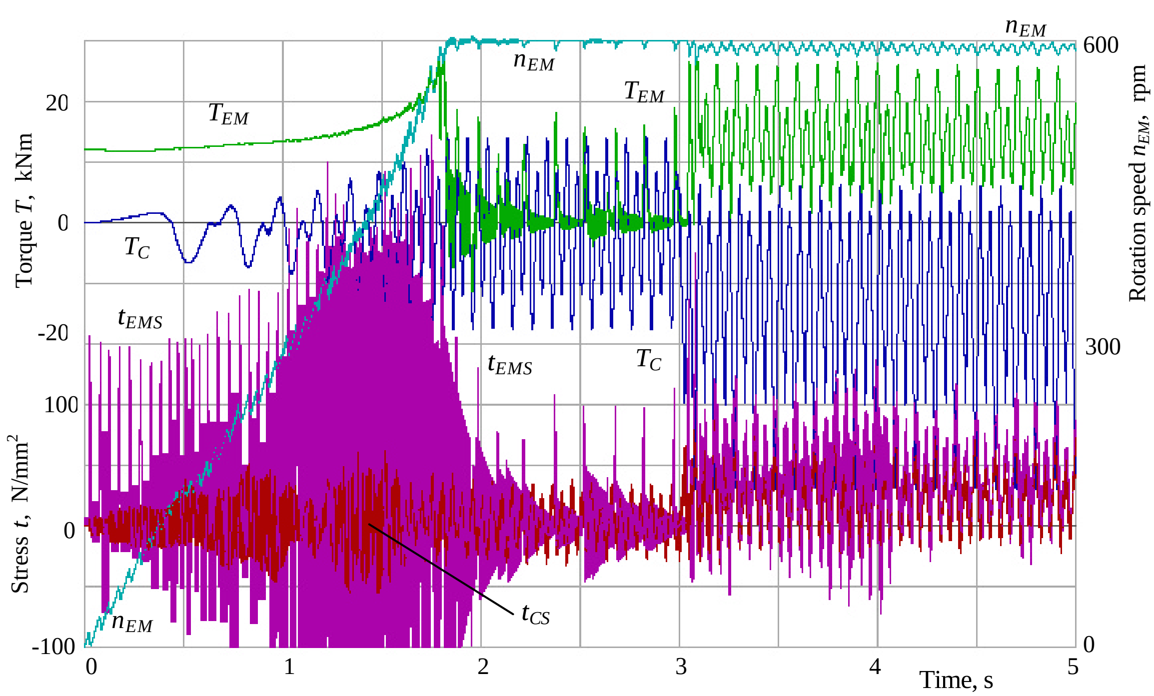

Figure 6 Results of the of the numerical simulations for the original design comprising elastic coupling with permitted clearance between coupling parts

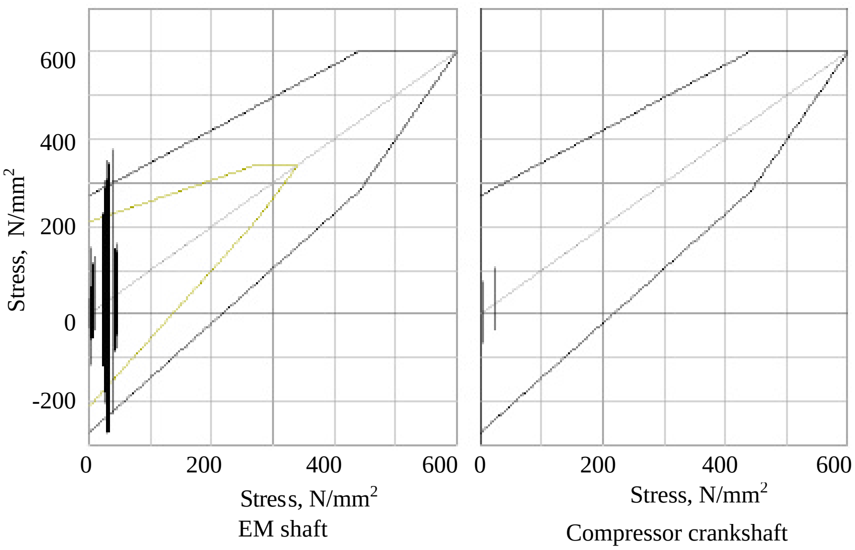

Figure 7 Smith diagrams for the electric motor shaft and compressor crankshaft for vibrations according to Figure 6

Figure 6 presents the results for the same original design, but with the clearance between parts of the elastic coupling. If we compare the stresses in the electric motor shaft there is evident that the breaks are inevitable. The same is also obvious from the Smith diagram given in the Figure 7 for the considered shaft material. The stresses for the electric motor shaft are out of the permitted area, so the breaks are to be expected after certain time of operation. As it is evident from the results, the majority of problems originates from the transients during the start of the compressor set. These transients are not covered by usual torsional vibration calculations.

After this analysis it was evident that the existing clearance has to be eliminated. As it can be seen from the Figure 4, there is also the necessity to set the system outside the dangerous area between the two main perturbing harmonics. To solve the problem, much softer elastic coupling, giving the very low first natural frequency, which is lower then the frequency of the first perturbing harmonic has to be applied.

To override the time necessary for the realization of this new solution, the temporary solution was applied, enabling the safe oil refinery operation. The temporary solution was realized using modified rubber elements instead of the original ones. These elements were sized to eliminate any possible clearance even in the hardest operation conditions. Temporary solution was also checked using the numerical simulations. The results are presented in the Figure 8.

As it can be seen, the stress in the electric motor shaft t_{EMS} (lower than 100 N/mm^2) was no longer problem, and the conditions for the safe operation of the compressor sets have been achieved. Due to the various circumstances and delays in the realization of new elastic couplings, the temporary solution has enabled the safe and reliable operation of the entire group of compressor sets during four years. After application of this very simple and cheap solution, no shaft breaks happened.

The reason why this solution has been considered only as temporary was the constant danger of possible electric motor short circuit event. As the elastic coupling was torsionaly very rigid (which was necessary to skip out the first natural frequency from the area of the 1st and 2nd perturbation harmonic), any instant increase of the torsional torque could be harmful for the electric motor shaft with immediate shaft break as a consequence.

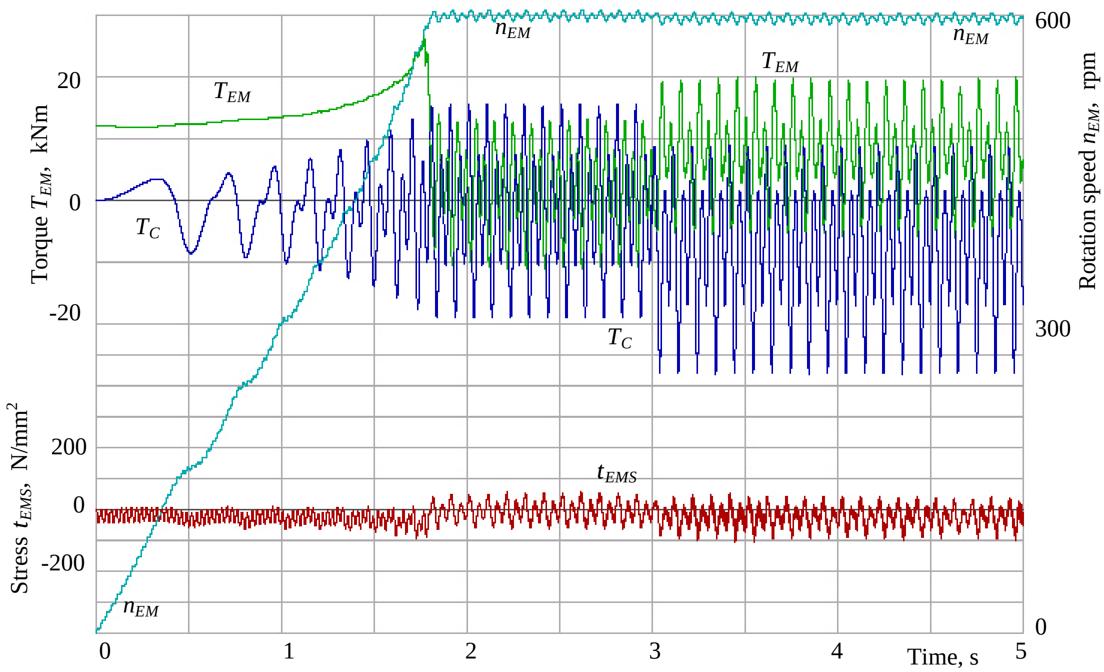

Figure 8 Results for the analysis of the system with the temporary solution of the elastic coupling

4. NUMERICAL ANALYSIS OF THE NEW ELASTIC COUPLING

New elastic coupling was selected to achieve very low natural frequency of 6.7 Hz for the compressor set rotating system. Lower frequency seemed to be favorable at the first glance, but problems in the shut-down transient were encountered.

Due to the existing space restrictions, new flywheel was designed to enable the application of the new coupling. The properties of the new flywheel (mass and the momentum of inertia) are the same as for old one, but the geometry was changed. Application of the new coupling has solved many problems regarding the stress in electric motor shaft. The shaft is now protected also for the case of the electric motor short circuit event.

The new solution has solved a lot of problems which have been detected in original compressor set design. Many of these problems have been avoided by using the new coupling. Beside of the solved problems, new problems have emerged, as detected by using numerical simulations on the solution proposals from the side of the components contractor. One of them was the normal shut-down of the compressor set.

As the first natural frequency is in the new solution lower than the 1st perturbing harmonic frequency, the compressor set is now operating in the vibroisolation mode. The system response to forced torsional vibrations is very low, giving very small stress amplitudes in the electric motor shaft.

When the compressor set is shut-down, the compressor is unloaded. As the kinetic energy of the entire rotating system is high, the stopping time is very long. It lasts more than 30 seconds, sometimes more than 60 seconds. During this time the rotation speed decreases slowly. When the speed frequency dropps to the first natural frequency, the vibration resonance is achieved. As the time of residence in this area is relatively long, very high vibrational amplitudes can be achieved, thus resulting in dangerous shaft stresses. Not only the shaft stress was very high but the coupling rubber elements deformations are also large. That increases the danger of the rubber elements rupture and damage of the elastic coupling. It is desirable to shorten the shut-down event, operating with the compressor as in the case of the emergency stop under the load.

First solution proposals have used coupling designs with limited twist angle, lower than the simulated coupling response. When applied, this coupling will not survive the shut-down procedure, although giving very favorable results in steady state operation. To eliminate this danger of large vibrations in resonance during the long shut-down period, the stiffer coupling design with high damping rate was recommended and selected. The material damping impose limits to the vibration amplitude maintaining the shaft stress within the permissible stress limits when passing through the resonance area. As the kinetic energy is converted through the damping into the heat, the heat load of the rubber elements has to be considered for the entire transient. Excessive temperatures may deteriorate the rubber properties.

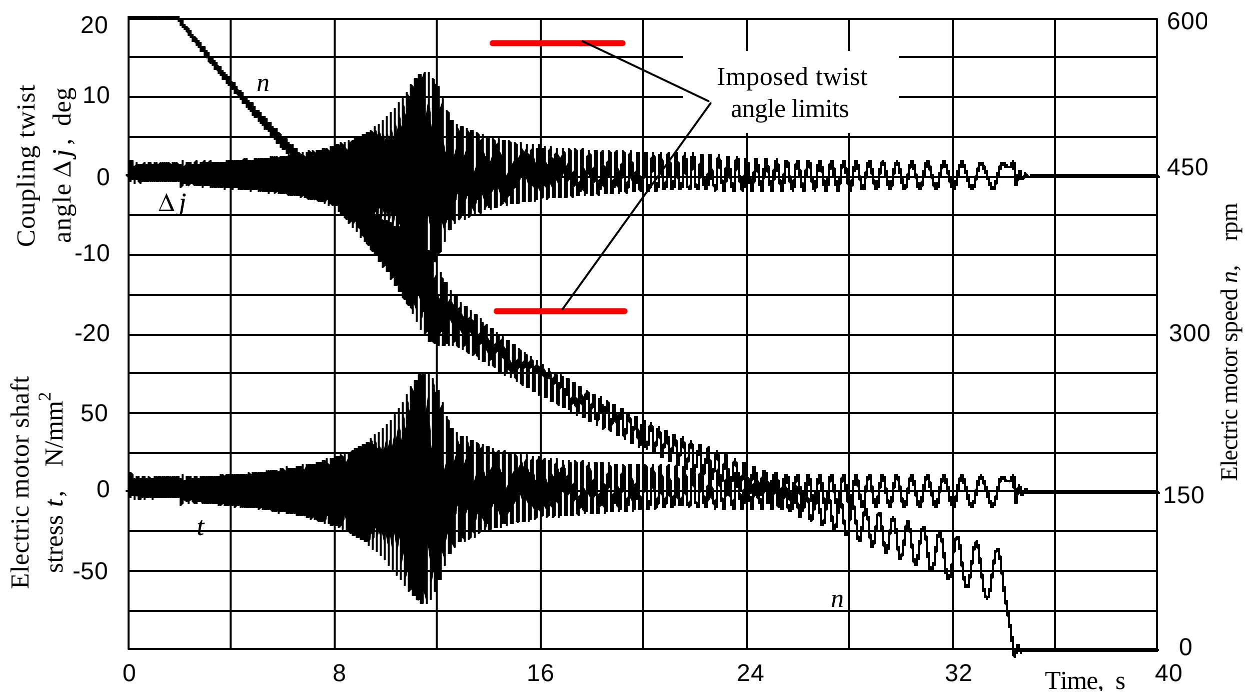

Figure 9 Shut-down simulation for the compressor set with new elastic coupling

The Figure 9 presents the results of the numerical simulation for the shut-down of the compressor set with new elastic coupling. The increased vibration amplitudes during the passing through the resonance area are obvious. It is evident that the maximum coupling deflection angle is lower than permissible (indicated by lines). The maximum stress amplitudes for the electric motor shaft are well under 80 N/mm^2 in this transient, giving very safe operation conditions.

5. CONCLUSIONS

The paper has indicated the possibilities of the torsional vibration analysis by using the numerical integration of the differential equations set for the system motion. The performed analysis has pointed out the importance of the simulation of transient events in finding out the design solutions to enable safe operation of the entire system. By using the presented analysis the existing problem was successfully solved, thus increasing the safety and reliability of the entire system operation, decreasing the cost caused by production losses and reparation costs.