FAILURE DIAGNOSIS OF THE GAS COMPRESSOR DIAPHRAGM VANE

Zbigniew KOZANECKI, Jakub ŁAGODZIŃSKI, Dorota KOZANECKA

Technical University of Łódź, Institute of Turbomachinery
Wólczańska 219/223, 93-005 Łódź, Poland, e-mail: zkozan@p.lodz.pl

Summary

The goal of this paper was to develop a methodology of explanation of diaphragm vane defects in a two–stage centrifugal compressor which had occurred in during operation of the machine. The methodology consisted of experimental investigations of the disassembled diaphragm, then a dynamical analysis using the finite element method on the *3D* diaphragm model, obtained by reversing engineering with an optical photogrammetric camera of the *TRITOP CMM* measuring system. The final purpose of this research work was to formulate the recommendations to avoid future problems.

Keywords: centrifugal compressor, diaphragm vane, damage, *3D* model.

DIAGNOSTYKA USZKODZEŃ ŁOPATEK KIEROWNICY SPRĘŻARKI PRZEPŁYWOWEJ

Streszczenie

Celem pracy było opracowanie metodyki pozwalającej na wyjaśnienie przyczyn uszkodzeń łopatek kierownicy dwustopniowej sprężarki promieniowej, które powstały w okresie eksploatacji maszyny. Zaproponowana metodyka obejmuje badania eksperymentalne oraz weryfikację modelu teoretycznego *3D* kierownicy, opracowanego przy wykorzystaniu nowoczesnego systemu optycznej metody skanowania w widmie światła białego TRITOP CMM. Końcowym efektem pracy było sformułowanie zaleceń modyfikacji konstrukcji maszyny dla eliminacji tego problemu.

Słowa kluczowe: sprężarka przepływowa, kierownica, łopatki, uszkodzenia, model *3D*.

1. INTRODUCTION

The origins of this paper were diaphragm vane defects, which had occurred after about one-year operation of a large-scale centrifugal compressor. The considered machine was a two–stage compressor working in the natural gas transport line. The subcomponent under analysis was a vaned diaphragm between the first and second stage. The diaphragm guides the gas to the second stage impeller inlet.

The main purpose of this research work was to develop a methodology of explanation of the damage (fig. 1) causes and to formulate the recommendations to avoid future problems.

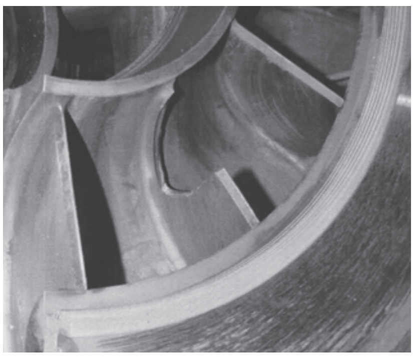


Fig. 1. View of the damaged diaphragm vane

The methodology consisted of experimental investigations of the disassembled diaphragm at the Institute laboratory, then a dynamical analysis using the finite element method on the *3D* diaphragm model.

2. THREE-DIMENSIONAL DIAPHRAGM MODEL

A *3D* model of the diaphragm was reconstructed using the reversed engineering method. The method is based on the continuous optical scanning of the marked points (fig. 2) with the *TRITOP* system. Afterwards, the measured point coordinates are transferred to the selected *CAD* system (fig. 3). The achievable retracing accuracy is about *0.02mm*.

Fig. 2. Diaphragm during the optical measurement

2. EXPERIMENTAL VERIFICATION OF THE CREATED MODEL

The investigations were conducted on the lower half of the diaphragm due to its less numerous operational damages than in the upper half. The vibrations were measured by a *Brüel – Kjaer 4375* type accelerometer with a *2635* type preamplifier. The acquisition and vibration data analysis was performed with a *LMS Pimento 5.1* analyzer. During the harmonic response analysis, diaphragm vanes were excited by a *LDS V201* type electrodynamics shaker powered by a *LDS PA25E* type power amplifier and by an impact hammer.

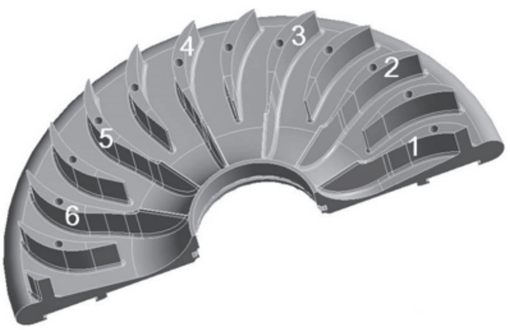


Fig. 3. Numbered vanes of the *3D* diaphragm model

Figure 4 shows a *FFT* response spectrum of each vane to the impact hammer. Table 1 contains an interpretation of all response spectra.

For each vane, three natural frequencies, corresponding to three first mode shapes of each vane, were taken down.

Table 1. *FFT* response spectrum interpretation

Vane number	1	3	4	5
1st natural frequency *[Hz]*	1761	1750	1749	1778
2nd natural frequency *[Hz]*	3632	3453	3497	3528
3rd natural frequency *[Hz]*	4762	4449	4555	4509

During the next step, a *FFT* response spectrum for white noise excitation was measured. The purpose was to simulate vane excitation by a fluid flow during the operating conditions. The results of the spectrum response (fig. 5) show a fuzzy nature of the spectrum near the *6kHz* frequency band. Taking this into account, the diaphragm model was verified upon three first natural frequencies, in the bandwidth below *5kHz*.

The natural frequencies acquired from response spectra became a reference for the tuning process of the diaphragm model. By adjusting an algorithm and a structure of the model mesh and other parameters, the model was tuned up to the real object.

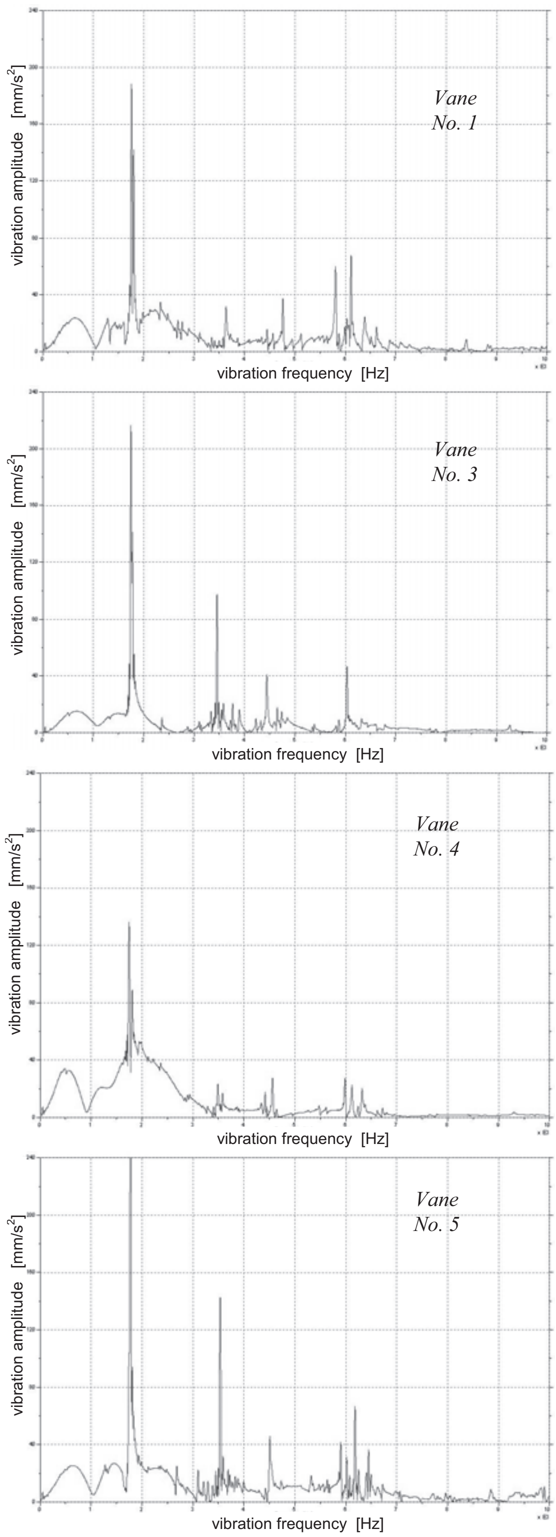


Fig. 4. *FFT* response spectrum in the *0-10 kHz* bandwidth of undamaged vanes.
The diaphragm cover is dismounted

Table 2 shows frequency differences between the tuned model and the real object. The disagreement did not exceed *3%*, which was considered as a good result.

Table 2. Experimental and model natural frequencies of diaphragm vane *No.1*

Mode shape	f_r (real) *[Hz]*	f_m (model) *[Hz]*	$\Delta f = \frac{\Delta f}{f_{rz}} \cdot 100\%$
1st	1761	1782	1,2
2nd	3632	3615	0,5
3rd	4762	4875	2,4

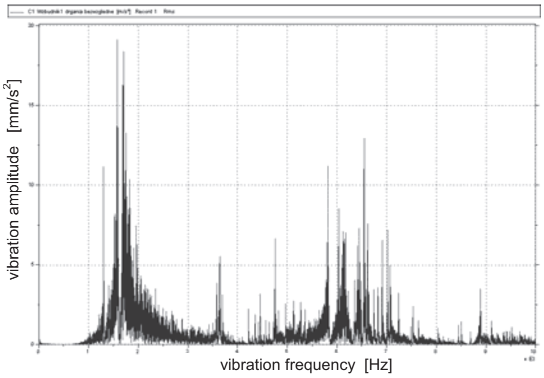
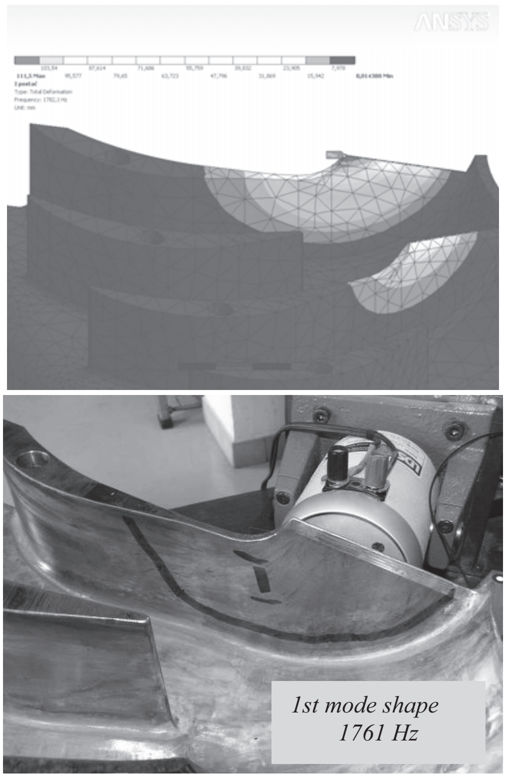


Fig. 5. Diaphragm vane No. 1 response spectrum to the white noise excitation in the *1 – 10 kHz* frequency bandwidth

Figure 6 shows theoretical and experimental natural mode shapes of single vane vibrations obtained respectively during calculations and experiments.



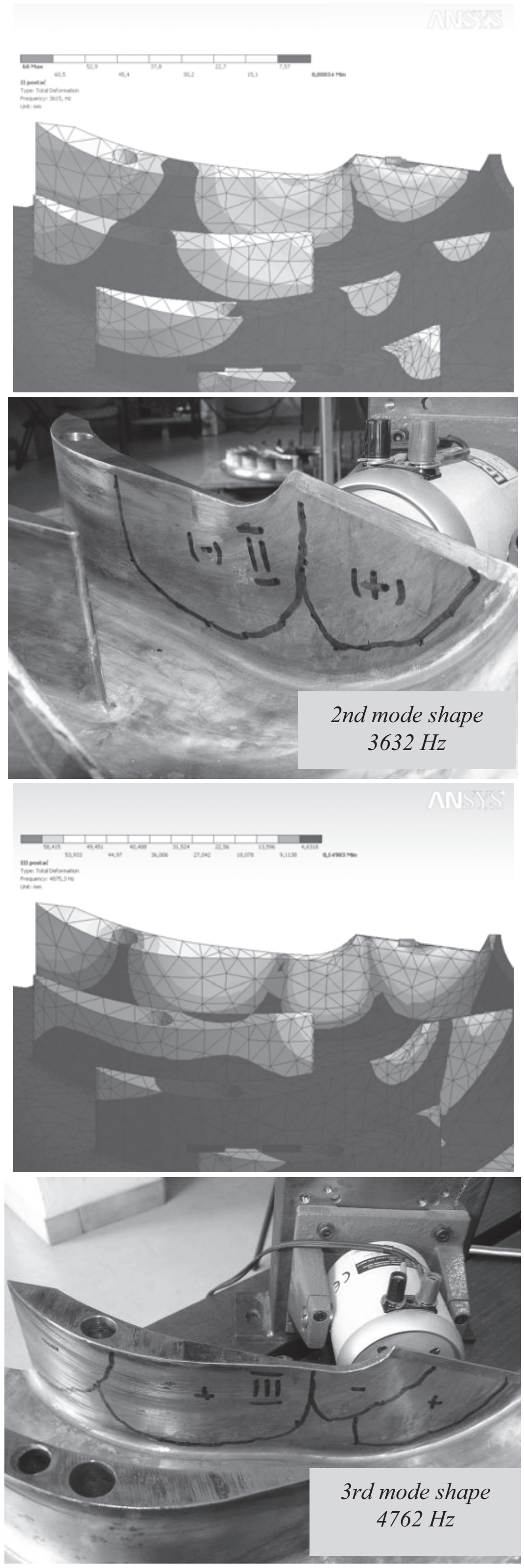


Fig. 6. Experimental and model natural frequencies of diaphragm vane *No.1.* in response to the white noise excitation

In the next stage, the diaphragm was fastened to the cover (fig. 7) and the impact response spectrum analysis was performed in this state.

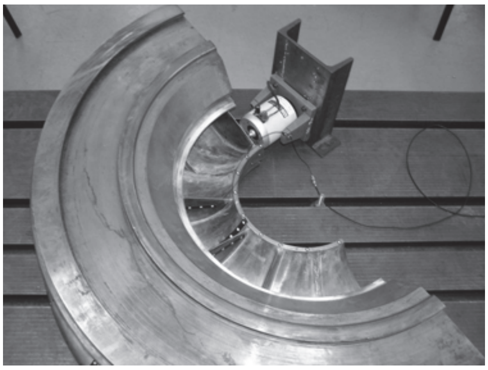


Fig. 7. Diaphragm fixed to the cover. A *LDS V201* type electrodynamics shaker is visible

Higher foundation stiffness caused an increase of natural frequencies with the fastened cover (fig.8).

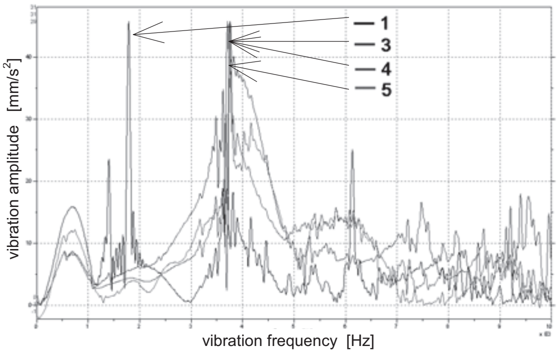


Fig. 8. *FFT* spectrum responses of vanes *No. 1,3,4* and *5* with the fastened cover

From the viewpoint of the stiffness increase of the fixed structure, an interaction between vibrations of the vanes and the diaphragm back plate was considered. Some results are shown in fig. 9.

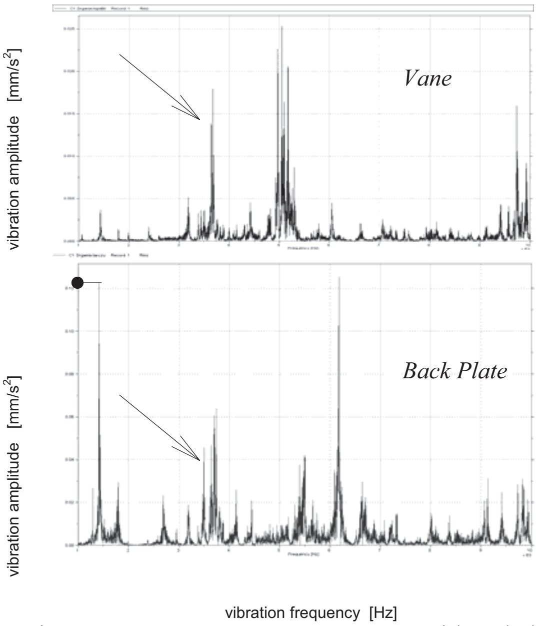


Fig. 9. Vane *FFT* spectrum response with a shaker whipping up the back plate and vice versa. The frequency interference is marked by arrows

The obtained results are shown in table 3.

Table 3. Interpretation of the diaphragm vanes response spectrum with the cover fastened

Vane number	1	3	4	5
Measured frequency	*1801Hz*	*3705Hz*	*3748Hz*	*3766Hz*

The band near frequency of *3700Hz* (marked by arrows in fig. 9) reveals an interference of the back plate and vane vibrations. It is likely that these natural frequencies amplify one another.

The tuning datum point of the diaphragm model with the fixed cover was to establish the contact surface between the mentioned elements.

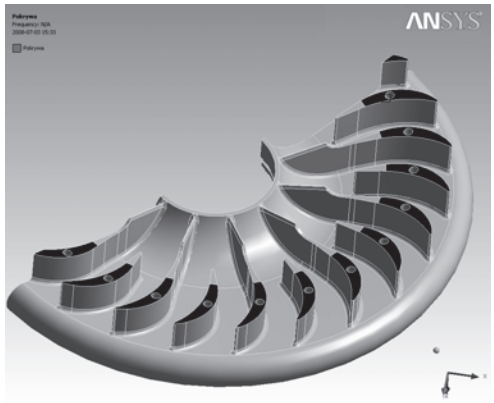


Fig. 10. *3D* model of the diaphragm with the fixed cover. The hypothetical contact surfaces are marked in black

Figure 10 shows a theoretical model of the diaphragm with the fixed cover, where the assumed contact surface is marked in black. The contact surface in the real object depends on the assembly technique, surface roughness and was a variable parameter for the tuning process.

4. NUMERICAL APPROACH

In the next stage a numerical modal analysis of the tuned model was performed and a solution was obtained. In fig. 11 mode shapes of the calculated model are shown. They are related to the measured real object natural frequencies, respectively *1410Hz* and *1801Hz*. A comparison of natural frequencies calculated from the model and measured on the real object is contained in table 4.

Table 4. Natural frequencies of the diaphragm vane with the fixed cover

Mode shape	f_r (measured) *[Hz]*	f_m (calculated) *[Hz]*	$\Delta f = \frac{\Delta f}{f_r} \cdot 100\%$
1st	1410	1328	5
2nd	1801	1819	1

Figure 11 shows theoretical mode shapes of the covered diaphragm vibration obtained during calculation in the frequency band below *2kHz*.

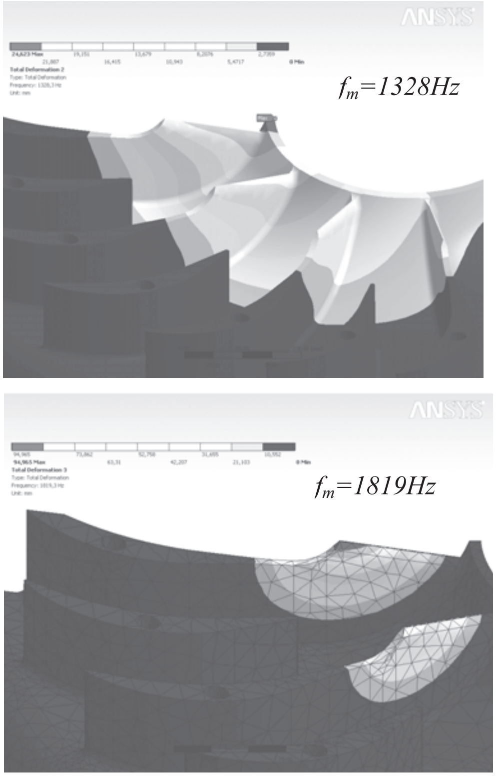
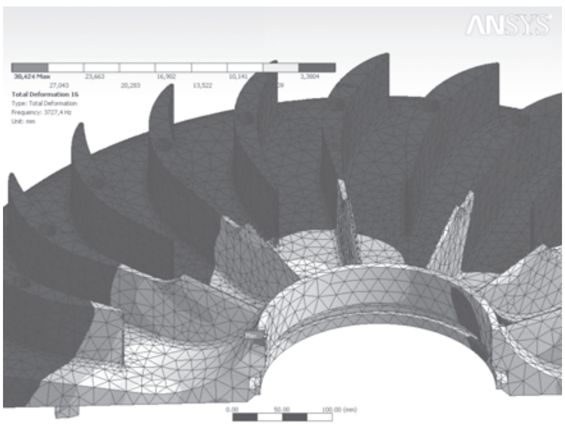


Fig. 11. Mode shapes of the diaphragm with the fixed cover in the frequency band below *2kHz*

The mode shape related to the frequency of *1328Hz* (fig. 11) is a complex mode of the back plate (marked by point in fig. 9), which definitely has an influence on vibrations of vane *No.1*.

Natural frequencies occurring between *2÷5kHz* are related to complex phenomena of the mutual vibration interference among the back plate and the vanes. A few natural frequencies overlap in this range. The white noise excitation response spectrum has a higher resolution, so it allowed for reducing the overlapping bandwidth to *3.5÷4kHz*. Calculated mode shapes from this range are shown in fig. 12.



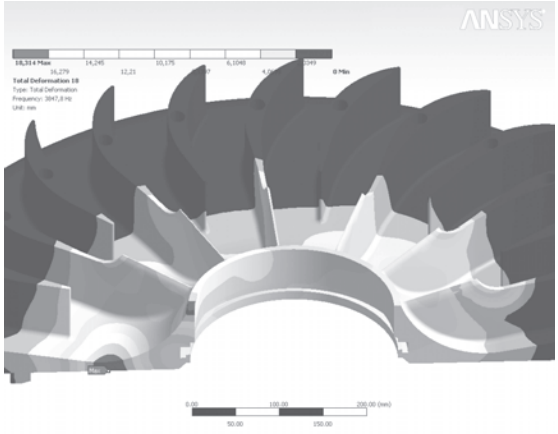


Fig. 12. Mode shapes of the diaphragm with the fixed cover from the bandwidth between *3.5÷4kHz*

The obtained model identification results were considered satisfactory, so the model could be used in a further stress analysis. The purpose of the stress analysis was to explain the origin of vane cracking during operation.

5. STRESS ANALYSIS OF VIBRATING COVERED DIAPHGRAGM

To approximate real load conditions in the model, the following assumptions were taken:

- Pressure pulsation caused by the transient fluid flow did not exceed *100kPa*.
- Pressure pulsation influence area was limited to the elongated part of the trailing edge of the diaphragm vane.

Figure 13 presents load conditions applied to the diaphragm partial model.

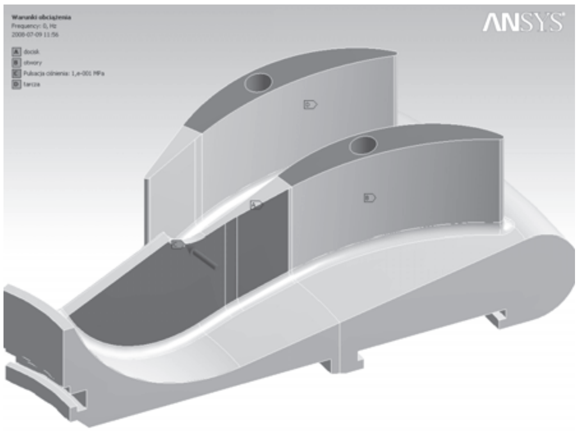


Fig. 13. Load conditions applied to the partial diaphragm model

Assuming that load conditions were only a qualitative approximation of the real fluid pressure pulsation, the obtained results could be also qualitative only.

The analysis performed during this research stage was also qualitative and consisted of the following steps:

1. Calculation of mode shapes and a stress distribution in the present diaphragm partial model.
2. Calculation of mode shapes and a stress distribution in the modified diaphragm partial model.
3. Comparison of the numerical results obtained in steps 1 and 2.

Figure 14 shows a von Mises stress distribution in the vane and a nodal line of the mode shape associated to the load conditions applied to the present structure.

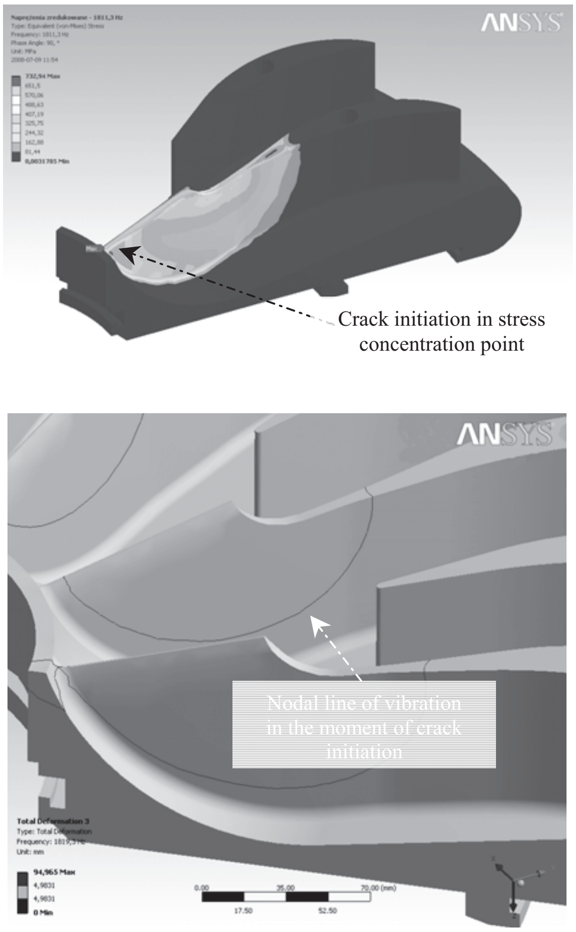


Fig. 14. Von Mises stress distribution in the diaphragm partial model and a nodal line of vane vibration at the moment of the crack initiation

The obtained results show clearly that a stress concentration occurs on the trailing edge of the drawn-out diaphragm vane and the nodal line overlaps the crack propagation lines on the damaged vanes.

Figure 15 shows a theoretical crack propagation caused by the presented phenomena. A crack initiation appears in the same place as in the real object – on the trailing edge of the elongated vane.

During the crack propagation, releasing of the trailing edge causes a change of the initial nodal line to the nodal line of the vibration mode called flutter (black dot line in fig. 15).

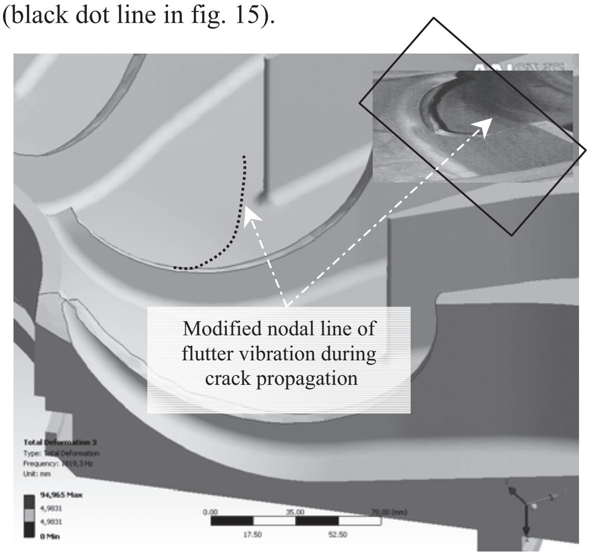


Fig. 15. Nodal line modification during vane crack propagation in comparison with real shape of the crumbled part of the damaged diaphragm

This explains why crumbled fragments are smaller than the parts confined with the initial nodal line of the undamaged vane.

6. RESPONSE ANALYSIS OF THE MODIFIED STRUCTURE

A temporary repair was proposed. In the repair, a part of the vane trailing edge was welded to the diaphragm cover.

In consequence, the diaphragm and the cover became unable to disjoin. Fortunately, the machine assembly procedure allows for this kind of modification.

Figure 16 presents a von Mises stress distribution in the modified diaphragm partial model. This stress distribution is more favorable for the analyzed structure. For the same model load conditions, von Mises stresses in the critical point of the trailing edge are reduced *2.5* times.

It should be pointed out that in the welding point (about *20mm* long – assumed in the numerical analysis), a half as high von Mises stress as in the crack initiation point of previous design has appeared. It would be favorable to join the vanes with the cover using a fillet weld with weld penetration to reduce the stresses occurring in the weld.

From a vibration point of view, stiffening the elongated vanes by welding them to the cover improves the dynamic properties of the diaphragm.

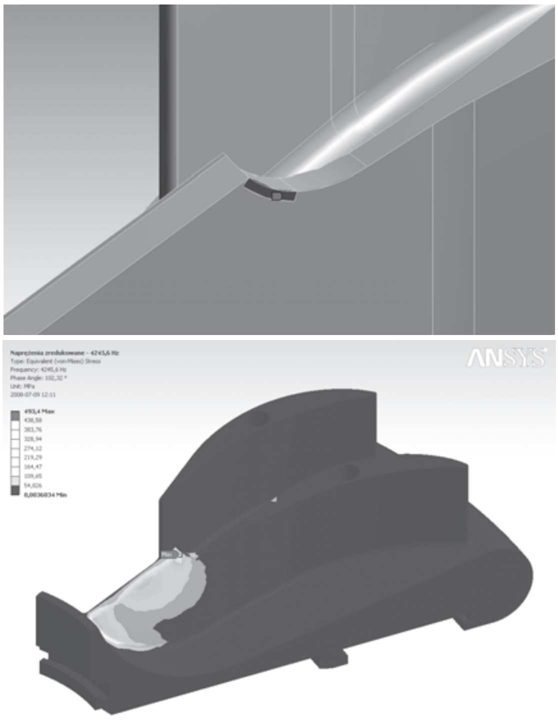


Fig. 16. Altering weld and a von Mises stress distribution in the modified diaphragm partial model

Figure 17 presents a comparison of frequency responses of the free and welded vane under the same load conditions.

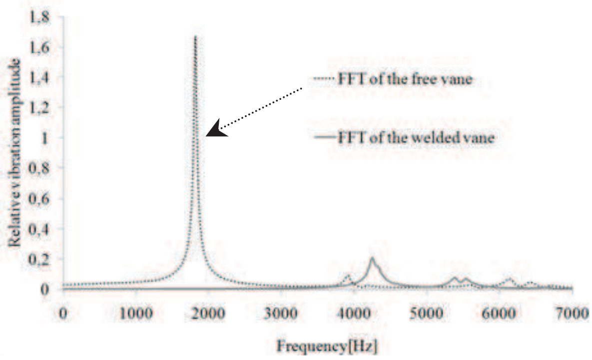


Fig. 17. *FFT* harmonic spectrum response of the free and welded vane for the applied load conditions

Stiffening the vanes by welding them to the cover eliminates dangerous low–frequency modes being the origin of all operational problems related to the compressor diaphragm.

7. CONCLUSIONS

The elaborated method allowed for explaining the causes of fracture and damage of two-stage gas compressor diaphragm vanes that appeared during the machine operation.

The theoretical model was built and verified by comparing it to the real diaphragm vibration measurement results.

The compatibility between the model and the real object behavior allowed for performing a number of numerical analyses.

Positive results allowed for introducing a modification and expressing the following conclusions:

1. Fracture and damage had a fatigue nature and were caused by a transient flow through diaphragm vane ducts.
2. Tests and numerical approach evaluated positively the proposed modification (repair technology) of welding vane trailing edges to the cover.
3. The results of the presented work applied in industrial practice caused elimination of the mentioned problem of diaphragm vane defects in large centrifugal compressors working in the natural gas transport line.

REFERENCES

[1] Morel J.: *Drgania maszyn i diagnostyka ich stanu technicznego*, Polskie Towarzystwo Diagnostyki Technicznej, Warszawa 1992.

[2] *Theory Reference for ANSYS and ANSYS Workbench 11.0*, ANSYS Inc. 2007.

[3] *The LMS Theory and Background Book*: Signal Processing, *LMS* International 2004.

[4] Harris C. M., Piersol A. G.: *Harris' Shock and Vibration Handbook*, 5th edition, MacGraw-Hill, 2002.

[5] Kozanecki Z.: *Systemy wirujące maszyn przepływowych małej i średniej mocy*, Biblioteka Problemów Eksploatacji, Wydawnictwo Naukowe Instytutu Technologii Eksploatacji, Radom, 2008.

[6] Osiński Z. i in.: *Tłumienie Drgań*, praca zbiorowa pod redakcją Z. Osińskiego, PWN, Warszawa, 1997.

[7] Eisenmann R. C, Eisenmann R. et al.: *Machinery Malfunction Diagnostic and Correction*, Hewlett-Packard Company 1998.

[8] Cempel C., *Podstawy wibroakustycznej diagnostyki maszyn*, WNT Warszawa ,1982.

[9] Wachel J. C.: *Compressor Case Histories, Presented at Rotating Machinery and Controls (ROMAC) Short Course*, University of Virginia, June 8-10, 1983.

[10] API 617 – *Centrifugal Compressors for Petroleum, Chemical and Gas Service Industries*, American Petroleum Institute – seventh edition, 2002.