46th International Scientific Conference EXPERIMENTAL STRESS ANALYSIS 2008
Department of Mechanics of Materials, VŠB - Technical University of Ostrava
June 2 - 5 2008, Horní Bečva, Czech Republic

František TREBUŇA *, František ŠIMČÁK **, Jozef BOCKO ***, Vladislav BERINŠTET ****

ASSESMENT OF VENTILATOR FAILURE ON EXHAUSTING OF CONVERTER

POSÚDENIE PRÍČINY HAVÁRIE VENTILÁTORA ODSÁVANIA KONVERTORA

Abstract

Operation of fan with rotor of diameter approximately 3m resulted to crash. The aim of analysis by numerical and experimental methods of mechanics was to determinate the steps that have led to the failure of individual parts of ventilator. For the problem solution was exploited the analysis of residual stresses in rotor as well as the measurement of vibrodiagnostic parameters on the second ventilator of the same type.

Abstrakt

Pri prevádzke ventilátora s priemerom obežného kolesa cca 3 m došlo k havárii. Cieľom analýzy s využitím numerických a experimentálnych metód mechaniky bolo určenie postupnosti porušenia jednotlivých prvkov ventilátora, ako aj ich príčinných súvislostí. Pri riešení úlohy bola využitá analýza zvyškových napätí v obežnom kolese ako aj meranie vibrodiagnostických parametrov na prevádzkovanom ventilátore rovnakého typu.

1 INTRODUCTION

Crashed ventilator was used for exhausting of converter gas during operation of converter. Radial ventilator in question has rotor diameter approximately 3 m. It lies in two bearings and it is powered through the clutch by electromotor (Fig.1a).

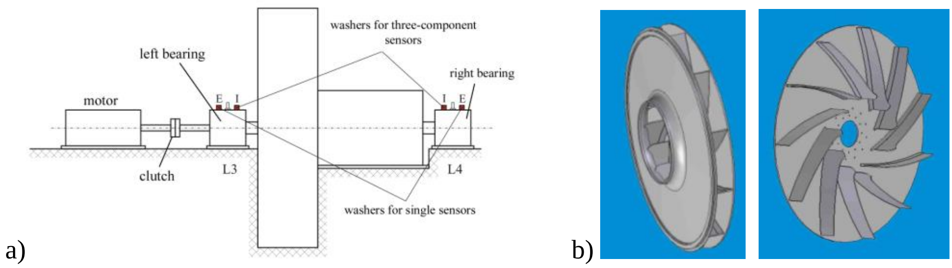

Fig. 1 Radial ventilator a) whole view, b) rotor with details of blades.

In Fig.1b is given model of rotor that was used for computation by the finite element method. Rotor with outer diameter 2923 mm is welded and consists of covering sheet (thickness 12 mm), supporting sheet (thickness 20 mm) which is fastened to the flange of ventilator's shaft by 24 bolts and eleven blades made of steel sheets of thickness 12 mm and 20 mm (Fig.1b). After several months of operation with 600 to 1400 rpm was the rotor damaged and consequently was damaged also whole

* Dr.h.c. mult. Prof. Ing. CSc., tel. (+420)556022461, e-mail frantisek.trebuna @tuke.sk
** Prof. Ing. CSc., tel. (+420)556022458, e-mail frantisek.simcak @tuke.sk
*** doc. Ing. CSc., tel. (+420)556022920, e-mail jozef.bocko @tuke.sk
**** Ing., tel. (+420)556022719 e-mail vladislav.berinstet @tuke.sk
Department of Applied Mechanics and Mechatronics, Faculty of Mechanical Engineering, TU Košice, Letná 9, Košice

fan. In Fig.2 is view to crashed rotor and deformed covering sheet of ventilator case. In Fig.3 is shown right bearing house of ventilator before and after crash. On the base of visual analysis of broken surfaces was found out that the first damaged element of ventilator was its rotor. This fact was confirmed by additional revision of rotor in the second parallel ventilator of the same type that is used for the same purpose. Penetration test of non-damaged rotor has confirmed presence of crack in supporting sheet of rotor. That is the reason why were the causes of crash searched in the properties of rotor. In the paper are given the results of numerical and experimental analysis with the aim to determine the reasons of failure of individual parts of ventilator.

Fig. 2 Damaged rotor and covering steel sheet

Fig. 3 Right bearing house of ventilator before and after damage

2 ANALYSIS OF ROTOR IN VENTILATOR BY THE FINITE ELEMENT METHOD

The finite element method was used for determination of stresses in the rotor of the ventilator as well as for computation of its eigenshapes and eigenfrequencies. Model of rotor for the stress analysis is shown in Fig. 1b. In Fig. 4 are given deformations and equivalent stresses in rotor for frequency 1400 rpm. The shaft was modelled for computation of eigenshapes and eigenfrequencies of rotor. In Fig. 5 is the first eigenshape of vibration. Corresponding eigenfrequency is f = 18,759 Hz for the rotor without rotation and f = 31,817 Hz for rotation frequency 1400 rpm. Eigenfrequency that corresponds to the third eigenshape was f = 31,376 Hz without rotation of rotor and f = 52,697 Hz for rotation frequency 1400 rpm. The second eigenshape (and also the second eigenfrequency) were in essence the same as the first eigenshape (ortogonality is preserved).

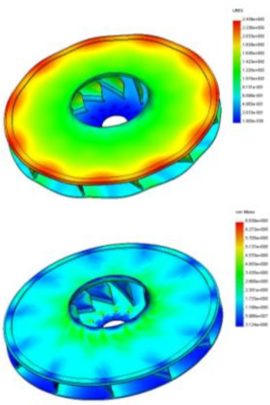

Fig. 4 Strains and equivalent stresses in rotor

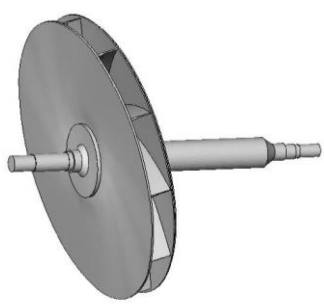

Fig. 5 The first eigenshape of rotor with shaft

3 DETERMINATION OF RESIDUAL STRESES IN ROTOR OF VENTILATOR

Because the crack in ventilator was located in welds or in intimate distance from the welds on the rotor (Fig. 6) there was accomplished measurement of residual stresses around welded joints. These stresses have very harmful effect because they contribute to material fatigue and they decrease lifespan of structure.

Fig. 6 Damaged rotor prepared for measurement of residual stresses

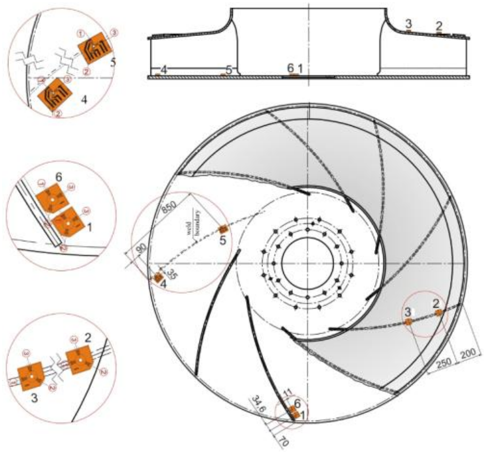

Fig. 7 Schema of strain gages locations and orientations

The hole drilling method was used for determination of residual stresses. The drilling was accomplished by hole drilling equipment RS – 200. In locations 1,2,3 and 6 was for drilling near to strain gages RY –21 used cutter with diameter 3,2 mm and in locations 4,5 for the drilling near to strain gages CEA 062 UL the cutter with diameter 1,6 mm. Hole depth was 3,5 mm for locations 1,2,3 and 6 mm and 2,4 mm for locations 4 and 5. For the measurement of strains was used strain gage apparatus P3. The magnitudes of residual stresses were determined from measured strain by code ASTM E 837 – 01 [4] as well as by Integral method and Method of Power – Series [3]. In Table 2 are given values of residual stresses in individual locations according to code ASTM E 837-01 (it presumes regular stress distribution along the hole depth).

Values of residual stresses according to code ASTM E 837-01 **Table 2**

Location of measurement	σ_{max} [MPa]	σ_{min} [MPa]	φ [°] declination angle from weld axis
1	-77,49	3,37	-86,71
2	-20,65	-2,19	-74,30
3	-25,76	-3,70	-84,24
4	-0,06	25,69	-7,73
5	19,83	68,39	62,35
6	-63,02	18,66	88,75

4 VIBRODIAGNOSTICS OF VENTILATOR

Every, also well-designed and well-working rotation machine generates mechanical vibration. This mechanical vibration is considerable big in ventilator in case if the eigenfrequencies of vibrations (or their multiples) are near to the frequencies of mechanical vibrations generated by rotation of rotor, to number of blades (blade frequency), to number of rolling parts of bearings (bearing frequency) and so on [1,2,5]. From the point of view of diagnostics is very important design and positioning of rotor (stiff or elastic). Vibrodiagnostic measurements were realized during operation on non-damaged ventilator of the same type and used for the same purposes, as was the damaged one. For the frequency analysis of ventilator was used system PULSE 6 made by

Brüel&Kjaer [6]. There were applied three-component sensors of acceleration of the type 4506 B. Location of washers for the fastening of sensors is shown in Fig. 1. 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. Measurements and subsequently frequency analyses were realized in frequency range 2 – 800 Hz for 600 – 1300 rpm (ventilator without loading) and for 1100 rpm during restricted operation of ventilator (because of damage of one ventilator was the rotation frequency limited to 1100 rpm). Maximal values of acceleration amplitudes were detected on the right bearing of ventilator (L4) during normal operation (1100 rpm). In Fig. 8 are given the charts of acceleration in *z* direction with respect to time and frequency for one measurement.

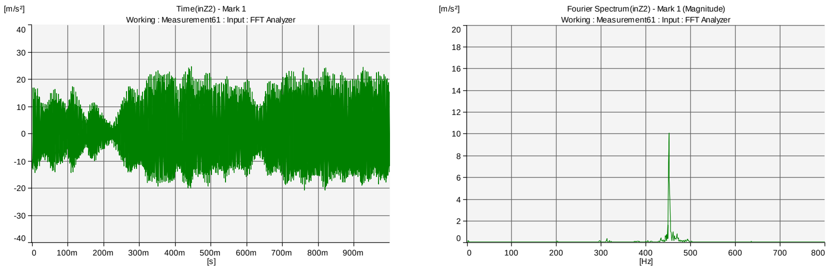

Fig. 8 Accelerations dependent on time and frequencies – Bearing L4, 1100 rpm

Experiences show 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. Code STN ISO 10816-1 categorizes ventilator in question to class III with the following band limits: A - $v_{ef} \leq 1{,}8$ mms^{-1} (new machines); B - $v_{ef} = 2{,}81 – 4{,}5$ mms^{-1} (common operation without time limit); C - $v_{ef} = 7{,}1 – 11{,}2$ mms^{-1} (not suitable for long-time operation – limited operation); D - $v_{ef} \geq 18$ mms^{-1} (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 mms^{-2} and this shift operation to boundary of band C that defines limited operation of ventilator.

5 CONCLUSION

In order to find reason of ventilator failure there was accomplished analysis of residual stresses, eigenshapes, corresponding eigenfrequencies 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.

This work was supported by project VEGA No. 1/0004/08 and No. 1/4163/07.

REFERENCES

[1] ČERMÁK, J.: *Ventilátory*. SNTL Praha, 1974.

[2] ECK.B.: *Ventilátoren*. Heidelberg-New York, 1972, 576 s.

[3] TREBUŇA, F., ŠIMČÁK, F.: *Kvantifikácia zvyškových napätí tenzometrickými metódami*. Grafotlač, Prešov, 2005.

[4] ASTM E 837-01. *Determining Residual Stresses by the Hole-Drilling Stran-Gage Method*. American Society for Testing and Materials, Philadelphia, 2001.

[5] ŽIARAN, S.: *Znižovanie kmitania a hluku v priemysle*. STU Bratislava, 2006.

[6] TREBUŇA, F., ŠIMČÁK, F.: *Príručka experimentálnej mechaniky*. TypoPress, Košice, 2007.

Reviewer: prof. MSc. Jan FUXA, CSc., VŠB - Technical University of Ostrava