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Modal Analysis of Exhaust Fan Impeller

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Abstract The paper deals with determination of modal parameters of exhaust fan impeller. The aim of performed analysis was to investigate whether resonance is the cause of excessive operational vibration of the fan. The paper describes results of experimental measurement by which the natural frequencies of non-rotating impeller were determined. The resonance frequencies and mode shapes during the rotation were estimated by using Finite Element simulation.

Keywords: *modal analysis, excessive vibration, impeller, fan*

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1. Introduction

During operation of every rotating machine dynamic loading invokes vibrations. In view of the ever-increasing power of machines and equipment, the intensity of possible dynamic effects increases and affects their lifetime and reliability. With respect to the fact that it is impossible to remove vibration completely, the allowed parameters of vibration, under which the long-time failure-free operation is ensured, are prescribed for every type of equipment. The causes as well as the development of vibrations can differ. In some cases, the cause is outer load acting during a short time period at the beginning of the operation, in other cases the loading can act during the whole operation of the machine. The causes and consequences of mechanical vibrations are dealt with in practice relatively often. At present, the problems of vibration are solved by various numerical and experimental methods of mechanics [1,2]. The authors have many years' experience with solution of operational problems related to vibration of machines and equipment [3,4,5,6]. For the solution of the given task the authors used their experience in conducting analysis of failure causes of similar equipment [7].

2. Analysis of the Problem

In the steel mill factory, there is installed the technological equipment serves for the exhaustion of combustion gases produced in the furnaces during steel making process. The equipment consists of two identical radial fans positioned opposite to each other. One of them in the long term shows the higher level of vibration. This excessive vibration finally led to the crack initiation in the concrete foundation (Figure 1a) and to damage of the bearing case on the free side of the shaft (Figure 1b). Because of the operator's apprehension that the equipment could be seriously damaged, the vibration and modal analysis of the fan were carried out.

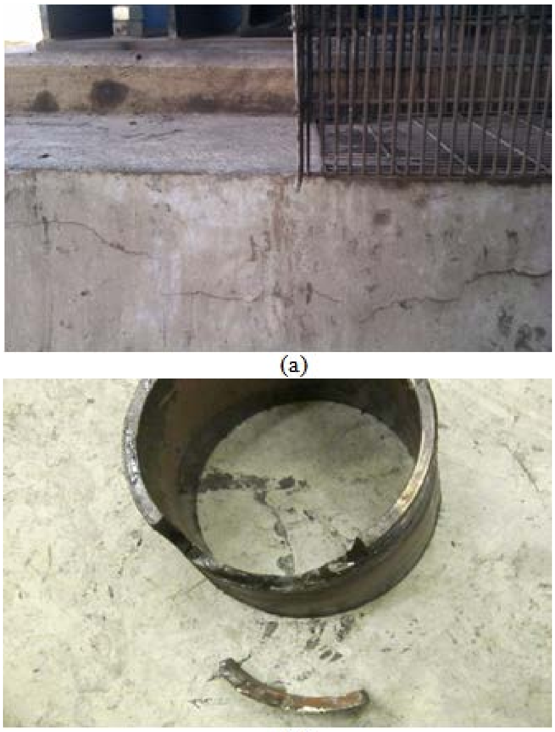
(a)
(b)

Figure 1. a) cracks in the foundation, b) damaged bearing case

The fan consists of the impeller and the shaft that is supported at its ends by two double-row barrel bearings. The impeller and the shaft are joined each other by 24 bolts on the center wheel disc. Figure 2 gives the basic dimensions of the shaft and the impeller. The fan is powered by motor with power of 1500 kW and maximum 1000 rpm. During steel melting process the fan operates in the interval from 400 to 1000 rpm that corresponded to the rotational frequency range 6.67 ÷ 16.67 Hz. Rotational velocity can be regulated by changing the frequency of the input voltage. The connection of the motor with the shaft is ensured by a sector spring clutch.

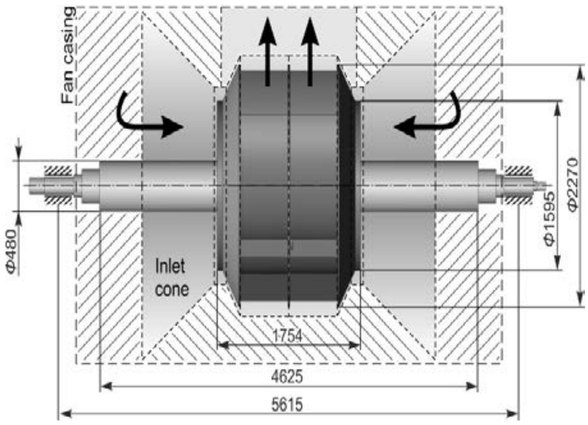

Figure 2. Basic dimensions of the impeller and shaft

3. Operational Analysis of the Fan Vibration

As the first, it was needed to get the estimation of dynamic behavior of the fan during its operation. For that purpose we performed the measurement which main aim was to determine vibration level at different values of the rotational frequency. The responses were measured by three uniaxial accelerometers Bruel&Kjaer 4507B and one triaxial accelerometer Bruel&Kjaer 4506B. The locations of the individual sensors as well as measurement directions are obvious from Figure 3.

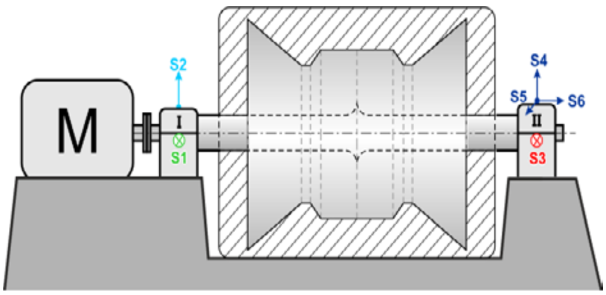

Figure 3 . Locations and measurement directions of accelerometers

During the measurement angular velocities of the fans were increased in the interval from 400 to 1000 rpm. The rotation velocities were regulated by using frequency changer. Its frequency was increased gradually by 1 Hz. The vibration level was assessed on the base of effective vibration velocity. Its value was determined from measurement data in accordance with international standard STN ISO 10816-1(Mechanical vibration - Evaluation of machine vibration by measurements on non-rotating parts) by the equation:

$$v_{eff} = \frac{10^3}{2\pi}\sqrt{\left(\frac{a_1}{f_1}\right)^2 + \left(\frac{a_2}{f_2}\right)^2 + \ldots + \left(\frac{a_n}{f_n}\right)^2} \qquad (1)$$

where a_1 to a_n are the acceleration amplitudes that correspond to frequencies f_1 to f_n of the frequency spectrum with range 2.5 ÷ 1000 Hz. Graphical charts of the effective vibration velocities for the whole range of the impeller‘s working rotations are shown in Figure 4.

Figure 4 shows that for certain rotational frequencies the vibration intensity of the fan is several-fold higher than in others, especially in horizontal and axial directions. According to STN ISO 10816-1 [8], for big power units with the power exceeding 300kW and other big machines with rotating masses that are fixed to solid and heavy foundations which are relatively solid in the direction of the vibration measurement, it is necessary to consider class III for which the band ranges are defined according to Table 1.

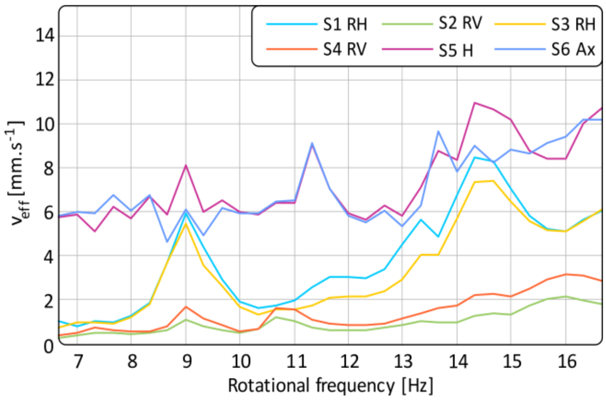

Figure 4. Effective vibration velocity as function of rotation frequency

Table 1. Typical band boundaries of effective vibration velocities of machine in class III [8]

Band	v_{eff} [mm.s^{-1}]	Characteristics
A	0,28 ÷ 1,80	Vibration typical for new machines
B	2,28 ÷ 4,50	Vibration of machines that are determined for unlimited long-time operation
C	7,10 ÷ 11,2	Vibration of machines that are unsuitable for long-time non-stop operation
D	18,0 ÷ 45,0	Not allowed vibration of machines

4. Modal Analysis of the Impeller

The knowledge of impeller’s natural frequencies is necessary for confirmation or exclusion of the assumption that the excessive vibration is caused by resonance. In the first phase, we performed experimental modal analysis of the non-rotating fan. Ten different input DOFs were chosen on the analyzed structure. Nine of which were on the shaft, point 10 was on the central disc of the rotating wheel. Selection of exciting points was limited because of the restricted access to the space of the rotating wheel. Individual locations and impact directions are given in Figure 5.

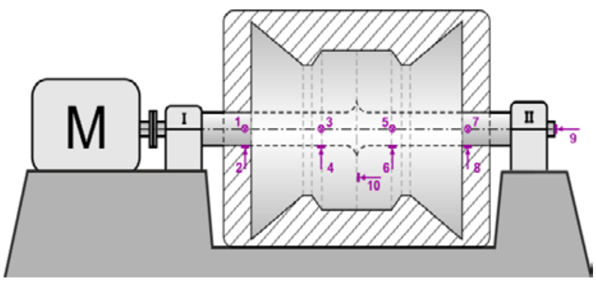

Figure 5. Locations and directions of excitations

The structure was excited by impact hammer Bruel&Kjaer 8210. The responses of vibration were measured by using five accelerometers Bruel&Kjaer 4507B. Their locations and measurement directions are given on Figure 6.

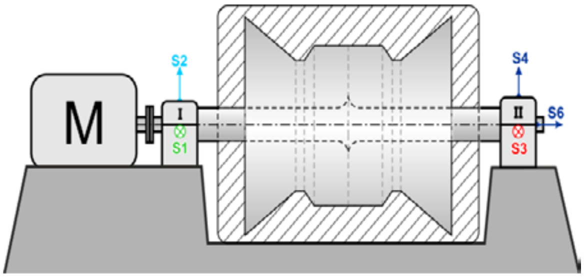

Figure 6. Locations and directions of measured responses

The result of measurement is the set of frequency response functions between the individual input and output DOFs. This set was used to obtain Complex Mode Indicator Function which allows simply estimate the natural frequencies of the impeller. The CMIF plot is shown in Figure 7.

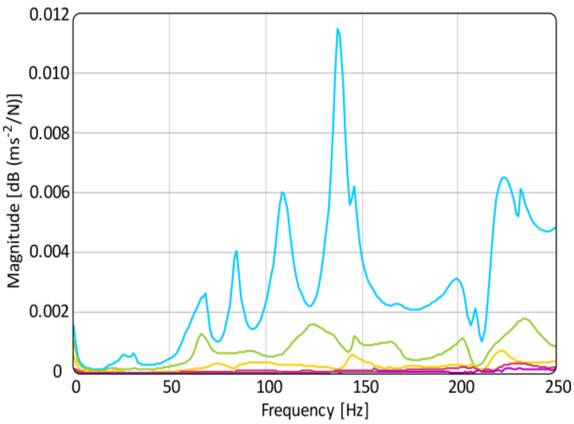

Figure 7. Complex Mode Indicator Function of the analyzed structure

As obvious from Figure 7, the first natural frequency of the impeller is approximately 25.6 Hz. During operation, its maximal rotational frequency is 16.67 Hz. Since in the course of the experiment it was not possible to choose more locations of excitation, numerical modal analysis of the system shaft - rotating wheel was performed by using finite element method. Finite element mesh consisting of ten node tetrahedrons is shown in Figure 8. A much finer mesh was used for the bolts and contact regions.

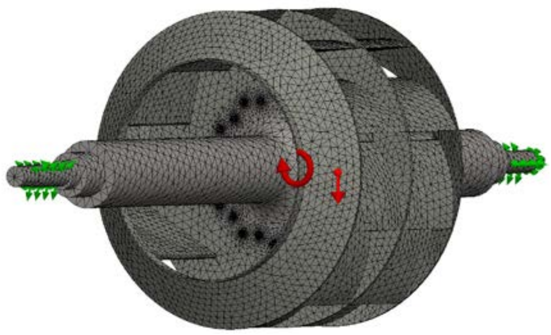

Figure 8. Finite element mesh of the numerical model

Resonance frequencies were determined for five different values of rotational frequencies of the shaft. The results of the numerical simulations are given in Table 2. Mode shapes of the analyzed finite element mode are shown in Figure 9. The results of the numerical solution also confirmed the finding that the first natural frequency of the structure lay outside the working rotational frequencies of the fan.

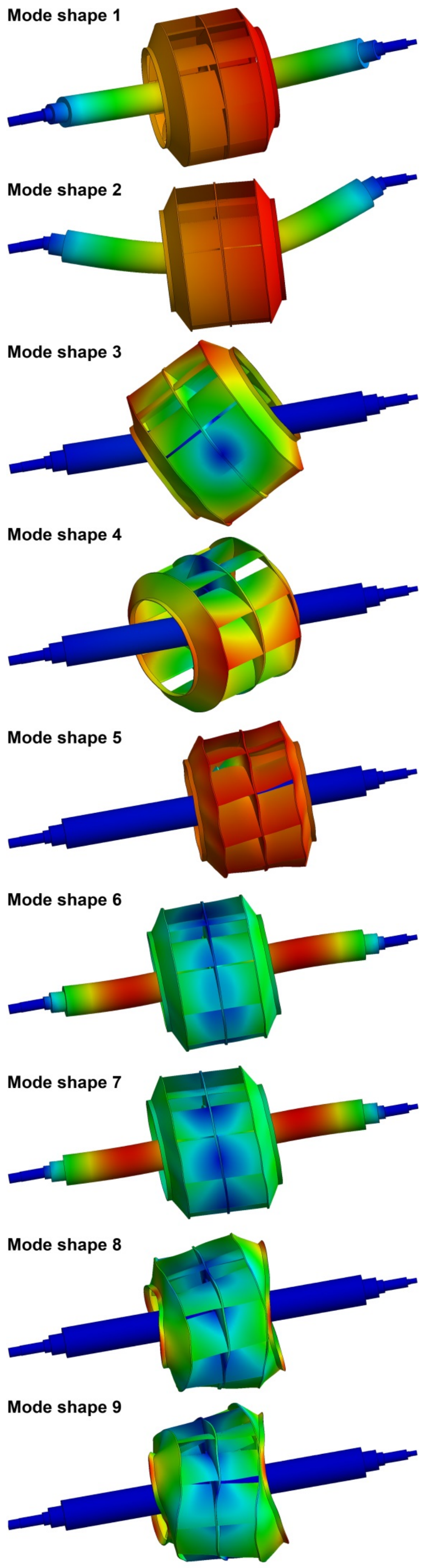

Figure 9. Mode shapes of the analyzed model at 1000 rpm

Table 2. Natural frequencies of the impeller obtained by FEM analysis

RPM	0	400	600	800	1000
Mode	Natural frequencies [Hz]				
1.	31.224	31.830	31.836	31.845	31.855
2.	31.253	31.859	31.866	31.874	31.884
3.	36.974	37.234	37.788	38.548	39.499
4.	37.165	37.430	37.978	38.731	39.675
5.	64.988	66,076	66.341	66.709	67.178
6.	137.76	138.56	138.61	138.66	138.73
7.	137.91	138.64	138.68	138.74	138.81
8.	144.21	145.12	145.91	146.69	147.68
9.	144.70	145.35	146.01	147.23	148.78

5. Evaluation of the Results and Conclusion

On the basis of the results of experimental and numerical modal analysis, it can be concluded that natural frequencies of the impeller are higher than the highest rotational frequency under which the equipment operates. From the above-mentioned it is apparent that during operation of the fan, the resonance areas of the impeller are not excited and accordingly the increase in the vibration intensity on the fan above the level of band C is caused by different factors. Therefore the following recommendations were given to the operator:

- perform experimental modal analysis and operational vibration analysis of the second (identical) fan,
- determine differences in modal and dynamic characteristics of both fans,
- perform vibrodiagnostic measurements on the first fan to determine possible damages.

Acknowledgement

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