509. Dependence between stiffness of fixed foundation and blower supports

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Abstract. This paper analyzes mechanical system consisting of two blowers, each having a separate frame. The system is fixed on one foundation. High-level vibrations are excited during its operation. Results of experimental research of this mechanical system are presented here. The developed finite element model provides the possibility to analyze natural frequencies of the system. The reasons of observed high-level vibrations were identified by comparing experimental and numerical results. The conception of vibration reduction was proposed based on the results of this investigation.

Keywords: mechanical system, blower, finite elements, vibrations, diagnostics, vibration analysis.

Introduction

Research technology of vibrations of rotor machines enables precise identification of the evolving machine defects and protecting the machines from unexpected faults and failures. Practice indicates that stiffness of foundation influences machine vibrations. The influence of vibrating foundation was analyzed in [1].

The development of a test bench to characterize vibration sources which apply unspecified excitations on a receiving structure was described in [2]. The approach is based on characterizations of a component mounted on different plates for classification with respect to the mobility of receiving structures.

In this paper two methods are presented for improvement of force identification [3]. The first one is based on improvement of the condition of the system FRF matrix by a proper selection of the measurement positions. The second method relies on modification of the structure by attaching a dynamic damper in a suitable location to minimize the ill-conditional nature of the FRF matrix especially near a resonance frequency [3].

In practice, the vibratory measurements accomplished by sensors come from a mixture of vibratory sources corresponding to different machine components. Thus it becomes challenging to conduct state interpretation for each individual component. This article proposes linking of modal analysis and stability. This approach allows us to reduce the number of sensors and to avoid regularization methods [4].

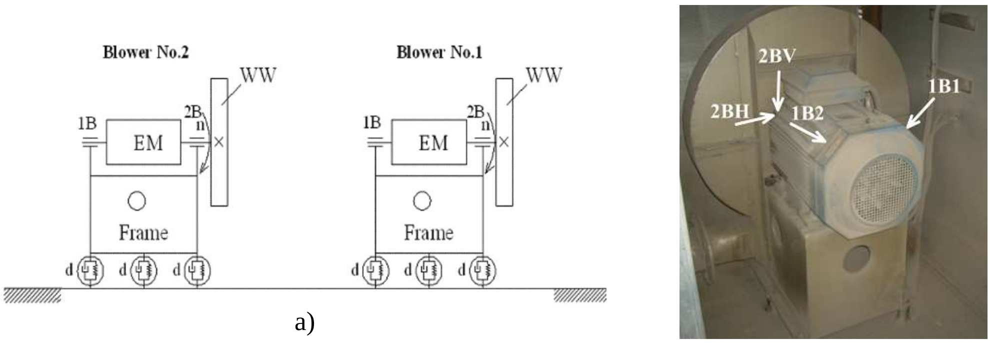
The diagnostics and monitoring of each component require the determination of the contribution of each source in the signal collection. The paper proposes a methodology, based on the restitution of the sources. Thus, the methodology ensures the detection and the localization of a defect of a component by the optimization of the position of a limited number of sensors [5].

The purpose of this paper is to analyze the reasons of generated high-level vibrations of two centrifugal blowers, which has negative influence to the health and work quality of employees. The conception of reduction of vibration intensity is developed as well.

Description of blowers

The two centrifugal blowers are used for production cooling in the food industry. The kinematics and technical parameters of the blowers are identical: power of electrical motor – 30 kW and nominal speed of rotor – 2958 rpm (the control of rotor rotation is implemented by means of frequency converters).

The blower consists of electrical motor with the rack fix working wheel and the frame (Figure 1). The dampers are mounted between the frame and the foundation (on the second floor) to reduce the intensity of vibrations induced by blowers. The blowers are assembled in one row – side by side.



a) b)

Fig. 1. a) Kinematics of two blower rotor systems, b) common view of one blower: 1B and 2B – the first and the second bearings; EM – electromotor; WW – the working wheel; n – rotational direction; d – the dampers; 1B1, 1B2, 2BH and 2BV – the vibration measuring points

There was a significant level of vibrations detected from the start of exploitation of the blowers in natural industrial conditions, which had a negative influence to the health of nearby employees working on the first and on the second floor of the building.

Dynamics of blower rotating system and fixed foundation before modernization

The research was performed using vibration signal analyzer A4300-VA3 (Adash, Czech Republic) with the vibrations accelerometer (Wilcoxon Research 797, USA). Experiments were carried out during operation of blowers at 100% load.

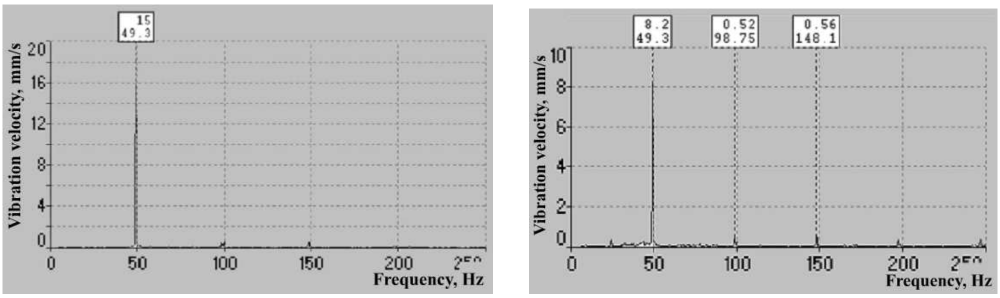
The main objectives of the experimental research were to measure the velocity values v_{RMS} of the excited vibrations (10÷1000 Hz) of the blowers, to identify the sources of these vibrations and to determine the causes of these vibrations. Vibrations velocity of fixed foundation was measured as well.

The experimentally measured vibration velocity values v_{RMS} of the rolling bearings before modernization of blowers are presented in Table 1. All measured bearing support vibration velocity values v_{RMS} of the blowers, except for the fourth machine, were in D vibration intensity range (by ISO 10816). The maximum vibration velocity value v_{RMS} measured was 16,29 mm/s (for the first machine of the first bearing in a horizontal direction). According to ISO 10816 standard these machines can not be exploited.

Spectral analysis of bearing vibrations velocity was performed to identify the possible sources of the high-level bearing support vibrations of the examined blowers. The spectrum of the horizontal vibrations speed of the first bearing of the first and second blower is presented in the Figure 2. All spectrums of velocity of bearing support vibration are analogous, because the dominant (under 1000 Hz) vibrations in these supports are only rotor synchronous (1X=49,3 Hz) and super synchronous (2X=98,75 Hz, 3X=148,1 Hz). These vibrations are generated due to low stiffness of the foundation and blower frame.

Table 1. Experimental research results for centrifugal blowers

Measuring direction	Vibrations velocity values v_{RMS} of bearing support, mm/s			
	Blower No.1		lower No.2	
	Load 100 %			
	1st bearing	2nd bearing	1st bearing	2nd bearing
Vertical	8,47	5,24	6,42	7,65
Horizontal	16,29	3,25	8,65	4,70



a b)

Fig. 2. Horizontal vibration velocity spectrum of the first and second blower (a) and (b) respectively of the first bearing (before modernization)

Low stiffness of the foundation and frame induce changes in axis position of blower rotor therefore resulting in excitation of additional dynamic forces (dynamic unbalance) at bearing supports and wear of these supports.

More detailed investigation of reasons of generated high-level vibrations of the first three blowers was performed by measuring and analyzing vibrations of blower frames. Measurements of frame vibration velocity v_{RMS} were taken in 9 points of the frame and are presented in Figure 3. The measurement was performed with the same equipment as mentioned above. The results of these measurements are listed in Table 2. The experimental data revealed that the highest values of vibrations velocity v_{RMS} of the measured frames occur at 5A (the 1st blower frame) and 8A (the 2nd blower frame) measuring points.

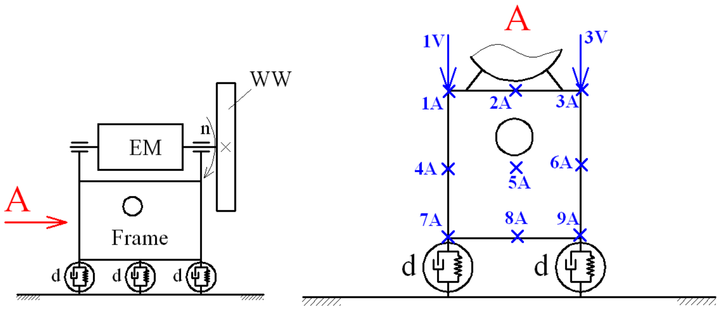


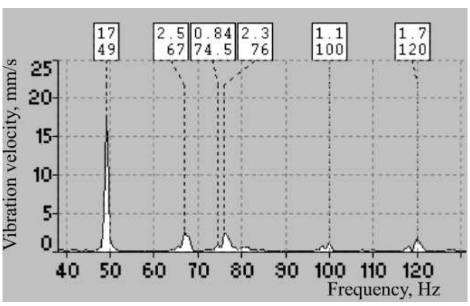
Fig. 3. The scheme of measurement points of vibrations of the blower frame: EM – electromotor; WW – working wheel; 1A, 2A, 3A, 4A, 5A, 6A, 7A, 8A and 9A – measuring points in the axial blower direction; 1V and 3V – measuring points in vertical direction

Table 2. Experimental research results of centrifugal blowers frame vibrations

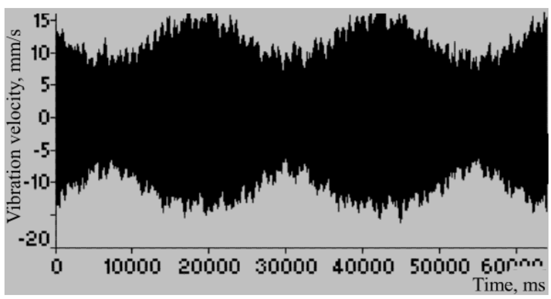
Vibrations velocity values v_{RMS}, mm/s Measuring points										
1st blower frame										
1A	2A	3A	4A	5A	6A	7A	8A	9A	1V	3V
10,90	7,06	6,74	18,40	**54,60**	14,40	23,90	38,90	20,60	9,62	10,10
2nd blower frame										
10,30	3,70	4,73	15,60	25,80	7,34	20,40	**30,70**	13,70	5,05	7,97

Figure 4(a) presents vibration velocity spectrum of the first blower at 8A measuring point and dominating vibration frequencies of rotor which are nearby synchronous 49 Hz/17 mm/s and nearby super synchronous frequencies vibrations (67 Hz/2,5 mm/s; 74,5 Hz/0,84 mm/s; 76 Hz/2,3 mm/s etc.).

Loose damper bolts were detected during the experimental research of blower vibrations. Vibration velocities of the foundation between the blowers were measured. Time response of vibration velocity of the foundation between 1st and 2nd blower is illustrated in Fig. 4(b). The dominant vibrations of foundation have a beating form which has a negative influence on the health of nearby working employees and work quality.



a)



b)

Fig. 4. Vibration velocity spectrum of the first blower frame at 8A measuring point (a) and vibration velocity plot measured between the 1st and 2nd blowers (b)

High-level of foundation vibrations were generated due to low stiffness of the foundation with blowers mounted atop. Therefore stiffening of foundation is necessary in order to diminish vibrations and improve working conditions for employees.

Modeling and simulation of the frame of blower

Theoretical calculations were performed for the blower frame resonant frequencies and vibration modes by using finite element method when finite element consists of eight nodes (Figure 5a) and six degrees of freedom (DOF). The first three are translations in the nodal x, y, and z directions and last three - rotations about the nodal x, y, and z-axes.

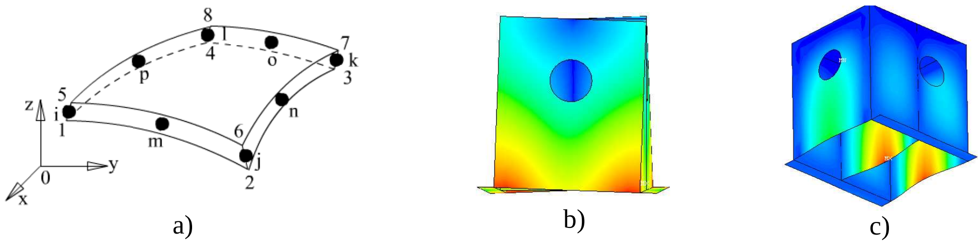


Fig. 5. a) finite element of eight nodes, b) – the first vibration mode of the blower frame (36 Hz) in case of loosened rubber dampers bolts and limp foundation, c) The second vibration mode of the blower frame (73 Hz), in the case of loosened rubber dampers bolts and limp foundation; the first vibration mode of the blower frame (73 Hz) in the case of dampers reducing the blower vibrations and stiff foundation

The equation of motion for an undamped system of the total blower frame, expressed in matrix notations is:

$$[M]\{\ddot{U}\}+[K]\{U\}=\{0\} \tag{1}$$

where: [M] - the mass matrix of frame finite element; $\{\ddot{U}\}$ - the second order nodal element displacement vector; {U} - the nodal element displacement vector; [K] - the stiffness matrix of frame finite element.
Nodal element displacement vector consists of:

$$\{U\}=\begin{bmatrix}u_x & u_y & u_z & \varphi_x & \varphi_y & \varphi_z\end{bmatrix}^T \tag{2}$$

The equation of motion for an undamped system was solved using these assumptions: the element has constant stiffness and mass effects, there is no damping and the element has no time varying forces, displacements, pressures.

For a linear system, free vibrations will be harmonic:

$$\{U\}=\{\phi\}_i \cos\omega_i t \tag{3}$$

Where: $\{\varphi\}_i$ - eigenvector representing the mode shape of the i-th natural frequency; ω_i – i-th natural circular frequency, *t* – time.

The equation (4) was obtained as a result of two step differentiation of equation (3) and using equation (1):

$$\left(-\omega_i^2[M]+[K]\right)\{\varphi\}_i = 0 \quad (4)$$

Modeling of frame resonant frequencies and vibration modes were done in two cases:

a) when damper bolts used for fixing the frame to the foundation are loosened (dampers do not reduce vibrations of blowers) and the foundation is limp;

b) the frame is stiffly fixed to the foundation.

The theoretical calculations revealed that in first case the 1st resonant frequency is 36 Hz, the 2nd – 73 Hz and in second case the 1st resonant frequency – 73 Hz. Resonant vibration modes of the frame are presented in Figure 5.

Dynamics of blower rotating system and fixed foundation after modernization

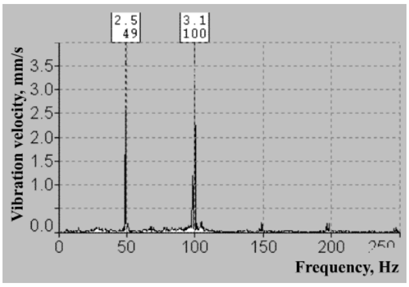
The decision was adopted to stiffen the foundation of blowers in order to confirm experimental and theoretical results of investigations. The stiffening of the foundation was accomplished by mounting additional stiff frame (the rest of the blowers rotary systems were left the same).

Fig. 6. The common view of the 1st blower with stiffened foundation

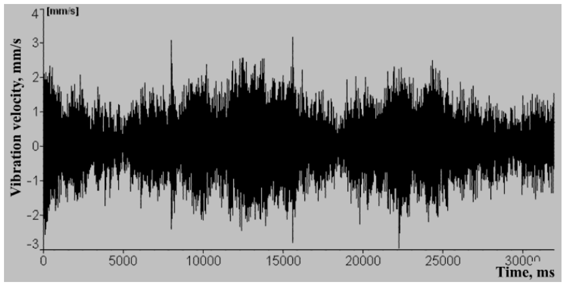
Table 3. Experimental research results of centrifugal blowers (load 100 %)

Measuring direction	Vibrations velocity values v_{RMS}, mm/s										
	Blower No.1		1st blower frame								
	1 bearing	2 bearing	1	2	3	4	5	6	7	8	9
Vertical	5,45	6,45	3,09	-	4,72	-	-	-	-	-	-
Horizontal	4,67	5,83	-	-	-	-	-	-	-	-	-
Axial	-	-	3,90	1,84	2,99	8,50	16,50	6,38	12,2	21,5	11,4

After the modernization vibration measurements were repeated for the blower bearing supports, blower frames and foundation between the blowers. The common view of the blower with the additional stiff frame is illustrated in Fig. 6. The vibration velocity RMS values v_{RMS} of the bearing supports of the 1st and 2nd blowers and of the frame of the 1st blower are presented in Table 3. The horizontal vibration velocity spectrum of the bearing of the 1st blower is given in Fig. 7(a) and the time response of vibration velocity of the foundation between the 1st and 2nd blowers is provided in Fig. 7(b).



a)



b)

Fig. 7. Horizontal vibration velocity spectrum of the bearing of the 1st blower (a) and time response of the vibration velocity of the foundation between the 1st and 2nd blowers (b)

After the modernization of the 1st and 2nd blowers the vibration intensity (v_{RMS}) in the bearing supports of these blowers decreased more than 3.4 times and in the frame of the 1st blower – more than 3.8 times, however frame vibration intensity still was high (the highest v_{RMS} value after modernization was 21,5 mm/s) due to the low stiffness of the frame. Meanwhile the comparison of the Fig. 4b and Fig. 7b indicates the decrease of the peak value of the vibration velocity of the foundation between 1st and 2nd blowers more than 5 times.

Conclusions

1. Numerical analysis demonstrated that in the case of loosened damper bolts and limp foundation the 1st resonant frequency of blower frame equals 36 Hz, while the 2nd one – 73 Hz. Operation of blowers when rotation frequency of electric motor may vary till 50 Hz is dangerous to the machine because it is working near the 1st resonant frequency.
2. Measurement results of vibration velocity values v_{RMS} of blower frame and spectral analysis of the measuring points as well as the theoretical research enable to identify reasons of generation of high-level vibrations in blowers.
3. After modernization of the blowers the vibration intensity (v_{RMS}) in the bearing supports was reduced by more than 3.4 times and in the frame of the 1st blower – more than 3.8 times. Moreover, peak value of the vibration velocity of the foundation between blowers decreased more than 5 times and the beating phenomenon in the working places was not observed.

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