Експериментально досліджено процес статичного і динамічного балансування кульовими автобалансирами крильчатки осьового вентилятора. Вивчено вплив автобалансирів на ділянці розбігу та вибігу вентилятора, а також ефективність роботи автобалансирів на ділянці крейсерського руху. Встановлено, що: наявність автобалансирів не погіршує вібраційний стан вентилятора на ділянці розбігу та покращує його на ділянці вибігу; на ділянці крейсерського руху доцільно застосовувати два автобалансири

Ключові слова: осьовий вентилятор, аеродинамічна незрівноваженість, вібропришвидчення, автобалансир, динамічне балансування, перехідні процеси

Экспериментально исследован процесс статической и динамической балансировки шаровыми автобалансирами крыльчатки осевого вентилятора. Изучено влияние автобалансиров на участках разгона и выбега вентилятора, а также эффективность работы автобалансиров на участке крейсерского движения. Установлено, что: наличие автобалансиров не ухудшает вибрационное состояние вентилятора на участке разгона и улучшает его на участке выбега; на участке крейсерского движения целесообразно использовать два автобалансира

Ключевые слова: осевой вентилятор, аэродинамическая неуравновешенность, виброускорение, автобалансир, динамическая балансировка, переходные процессы

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EXPERIMENTAL STUDY OF THE PROCESS OF THE STATIC AND DYNAMIC BALANCING OF THE AXIAL FAN IMPELLER BY BALL AUTO-BALANCERS

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1. Introduction

Axial fans are widely used both in industry and at home [1]. The usual and aerodynamic unbalances are the main sources of vibrations of such machines [2–6]. They are balanced both before the operation of the fan and during its operation [7–9]. There are known methods of the static and dynamic pre-balancing of fans, and their effectiveness is also determined [10, 11]. Simultaneous balancing of both usual and aerodynamic unbalances is possible [12, 13].

To investigate the transition processes at the dynamic balancing of both usual and aerodynamic unbalances is actual.

2. Literature review and problem statement

In the work [2], it was shown that the sources of vibrations of axial fans are: the usual (static and dynamic) unbalance of the rotating parts of the fan; technological unbalance, caused, also by aerodynamic forces, arising from the lack of symmetry of the impellers. In [3], vibrations caused by a non-axial installation of several rotating shafts, bearing supports and so on are added to these sources of vibrations. In the work [4], the basic approaches in the field of forecasting of aerodynamic unbalance are considered. In the work [5], the calculation of aerodynamic unbalance of a rotor of a turbocharger of an internal combustion engine is given. In [6], it was shown that the greatest contribution to the vibration of the considered fans is made by the usual and aerodynamic unbalances.

It is proposed in [7, 8] to reduce the aerodynamic and usual unbalances before the beginning of the operation of the fan. For this, first the aerodynamic unbalance is reduced by correction of the shape of the impeller blade [7]. Then the usual unbalance and remains of aerodynamic unbalance are reduced by balancing the rotating parts of the fan in assembly [8].

Such balancing can simultaneously reduce the load on a bearing, and the blade. It is considered in [8, 9] that mass correction acts at certain rotation velocity, that is, with a decrease or increase of velocity, the balancing is broken. Therefore, for pass-balancing of unbalanced variables, passive auto-balancers are used in the fan operation [9].

In [10], the statistical balancing process, its efficiency and the transition processes are studied experimentally. It is stated that the aerodynamic component of the unbalance does not prevent a balancing of usual unbalance. It is established that one auto-balancer, installed in the fairing of the fan, reduces the vibrations of its casing with a reserve of 1.5 times (in relation to the limiting value provided by GOST 11442-90).

In the work [11], the possibility of the dynamic pass-balancing (by two auto-balancers) of an impeller of an axial fan was shown. The conducted experimental studies confirm the effectiveness of dynamic balancing (loads on bearings are reduced by 40–70 %). Also, the possibility of further reducing of vibrations by improving aerodynamic characteristics is indicated. The transition processes were not investigated in the work.

In [12], the possibility of balancing of a swollen disk (the impeller) on a flexible shaft by means of a two-ball auto-balancer, located in the plane of the disk, was investigated. It was established that at supercritical velocities, the auto-balancers effectively balance both usual and aerodynamic unbalances.

In [13], the possibility of simultaneous balancing of both usual and aerodynamic unbalances of the impeller of an axial fan by rotating masses was proved. It was also proved that the usual and aerodynamic unbalances are similar to each other. Only aerodynamic unbalance depends on the density of air, weather conditions and the location of the fan above sea level.

Thus, for today the process of the auto-balancing of axial fans has been theoretically fairly well researched. However, these studies have not been confirmed experimentally: there are no experimental studies of the transition processes, arising during static and dynamic balancing. There is no evaluation of the effectiveness of these types of balancing.

3. The purpose and objectives of the study

The purpose of the work is an experimental study of the process of the static and dynamic balancing of the impellers of axial fans of the series VO 06-300 (Ukraine) by ball auto-balancers. This will improve the vibration characteristics of the fans at the manufacturing stage and increase the effectiveness of their balancing.

To achieve this purpose, it is necessary to solve the following research problems:

– to investigate the vibration characteristics of the axial fan stand: to find the quantity and the value of the resonant velocities (in the run-out section);

– to determine the frequency and source of the vibration (the section of the cruising motion);

– to perform a comparative analysis of the operation of the fan in various configurations of its stand (in the absence of auto-balancers and in the presence of an auto-balancer from the side of the impeller, the shank, both the impeller and the shank);

– to study the transition processes in the sections of racing and run-out in different stand configurations, as well as to determine the effectiveness of the static and dynamic balancing in the section of the cruising motion;

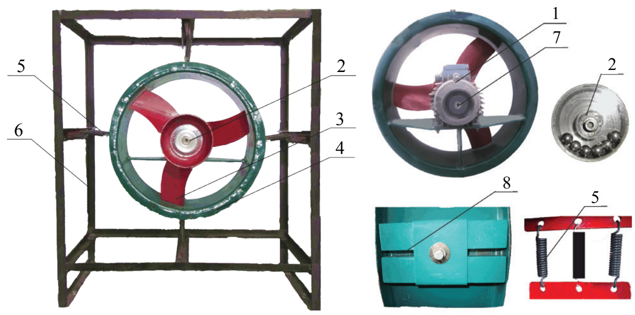
– to study the effect of additional masses, attached to the protective casing, on the vibration state of the fan.

4. Methods of researching the process of static and dynamic balancing of the axial fan impellers by ball auto-balancers

For the experiments, the known stand [10] (Fig. 1, *a*, *b*), consisting of: an asynchronous electric motor 1, an auto-balancer 2, mounted on the motor shaft from the impeller side 3, a fan protective casing 4, spring compliant supports 5, frame 6, was modernized. The modernization consists in:

– pushing the second auto-balancer 7 onto the shaft of the electric motor (in the shank);

– provision of attachment of additional masses 8 to the fan protective casing (they are fixed to the casing with the step of 90°, and an opportunity to change the weight and inertia of the fan characteristics).



a *b*

Fig. 1. Stand of the axial fan: *a* — stand in assembly (front view); *b* — components of the stand

For this stand, experimental models of ball auto-balancers are designed and manufactured. The balancing capacity of such auto-balancers with n identical balls was determined by the formula [14]:

$$S(n)=m(D-d)^2/(2d)\sin\{n \arcsin[d/(D-d)]\} \text{ (g mm)}, \quad (1)$$

where D (mm) is the diameter of the running track: m (r) is the mass of the ball, d (mm) is the diameter of the ball.

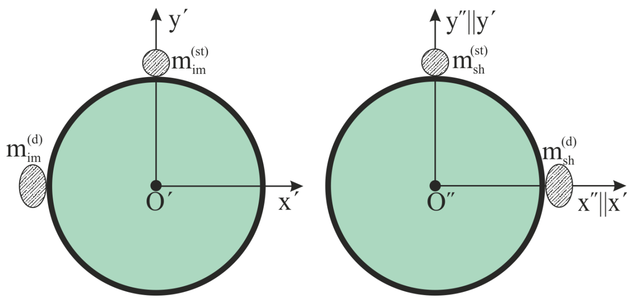
In the experiments, the auto-balancers with the following parameters: $n=6$, $m=6.93$ g, $d=11.9$ mm, $D_{im}=68$ mm, $D_{sh}=64$ mm were used (diameter and mass of the balls were selected in accordance with GOST 3722-81). With the selected parameters, the balancing capacities of the auto-balancers from the side of the impeller and the shank, respectively, are equal to $S_{imax\,im}=878.57$ g mm and $S_{imax\,sh}=776.42$ g mm.

Four configurations of the stand are considered: the 1st is without auto-balancers, the 2nd is with an auto-balancer from the impeller side, the 3-rd is with an auto-balancer from the shank, the 4-th is with two auto-balancers. The absence of auto-balancers is ensured by the absence of balls in it. The experiments were carried out in the absence and presence of additional masses, attached to the protective casing.

All experiments were carried out with fixed static and dynamic imbalances created by masses $m_{im}^{(st)}=m_{sh}^{(st)}=8$ g and $m_{im}^{(d)}=m_{sh}^{(d)}=17.4$ g.

The masses are located in the corresponding correction planes (in the impeller and the shank) as shown in Fig. 2. The mass distances to the longitudinal axis of the fan are equal to $R_{im}=42.5$ mm and $R_{sh}=40.75$ mm.

The values of the unbalances in the correction planes are equal to $S_{im}=813.92$ g mm, $S_{sh}=780.42$ g mm. They are chosen so that the vibrations of the casing (in the absence of auto-balancers and additional masses on the casing) exceed the maximum permissible level of vibrations not less than 2.5 times. This corresponds to an increase of the maximum permissible unbalance of the impeller by one class of the balancing accuracy.



a *b*

Fig. 2. Location of static and dynamic unbalances on auto-balances in: *a* – the impeller, *b* – the shank

Measurement of vibration accelerations at the stand (Fig. 3, *a*) is carried out with the help of accelerometer sensors MMA6231Q 2AX (Motorola, Fig. 3, *b*) with a sensitivity of 1.5 g. The sensors are connected to a personal computer (Fig. 3, *e*) through a USB oscilloscope-IRIS (Fig. 3, *c*, the Ukrainian assembly, based on the ADXL202EB-232A analog-digital board) (Fig. 3, *c*) with a USB interface (Fig. 3, *d*).

a *b* *c*

d *e*

Fig. 3. Scheme of connection of sensors for measuring of vibration accelerations to the stand: *a* – stand; *b* – sensor accelerometer; *c* – analog-digital board; *d* – USB interface; *e* – personal computer

Sensors and USB oscilloscope allow:

– to measure the absolute vibration acceleration (in mV);

– to record the oscillogram of vibration accelerations;

– to determine the number of components of vibration accelerations, their range (in mV) and frequency (in Hz).

The vibration sensor is isolated from the surface on which it is installed for eliminating the ground contour.

Vibration acceleration sensors were installed in three positions (Fig. 4):

– on the axis of rotation of the fan (on special platforms) from the side of the impeller (Fig. 4, position 1), the shank (Fig. 4, position 2);

– on the fan protective casing (Fig. 4, position 3) (for generalizing characteristics).

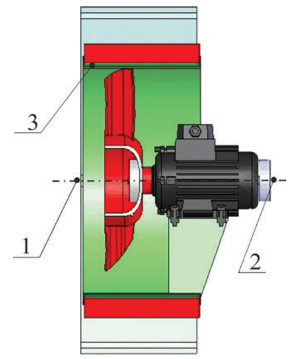


Fig. 4. Schematic of sensors for measuring of vibration accelerations: 1 – from the side of the impeller; 2 – from the shank side; 3 – on the protective casing

In the process of measurement, the instantaneous values of vibration accelerations, frequencies and intensities of the component oscillations, the amplitude of the vibration accelerations (V) at the frequency ω are determined. The USB oscilloscope measures the components of the vibration accelerations U (in mV). The standard formula [9] is used for the transition to the dimension m/s^2,

$$A = k \cdot U/2, \tag{2}$$

where k (m/(s^2 mV)) is the transition coefficient.

The amplitude of the vibration acceleration is found from the formula

$$V = A/\omega = A/(2\pi n), \tag{3}$$

where n (rpm) is the frequency of the rotor.

Sensors are set in such a way that they react to transverse vertical oscillations in the planes, close to the planes of balancing.

There are four types of experiments on the stand.

Experiment 1. Determination of the vibration characteristics of the stand

The electric motor is started in the first configuration without additional masses and the indicators from the sensor 3 are taken. The test conditions correspond to GOST 11442-90, GOST 10921-90, GOST 1940-1-90, GOST 31350-2007, GOST 31351-2007.

The instantaneous values of vibration accelerations on the fan corps in the frequency range 0–50 Hz (oscilloscope mode), as well as frequencies and intensities of the components of the oscillations in the range up to 50 Hz (the mode of the spectral analyzer) are determined.

The resonant fan velocities are determined on its run-out, since it is much longer than the racing. The vibrations of the motor corps significantly increase when passing through these velocities. The USB oscilloscope is started in the recorder mode and is monitored on the vibration acceleration diagrams (they display the value of the vibration accelerations in real time) to determine the resonant velocities.

Experiment 2. Investigation of the transition processes on the sections of racing of the cruising motion and run-out, and also the influence of auto-balancers and additional masses on the value of vibration accelerations.

The USB oscilloscope is started in the recorder mode and the vibration acceleration diagrams are studied in real time. The duration of each experiment is 45 seconds.

In the study of transition processes, the following sections are identified (Fig. 5):

– the section of rest section 0, from 0 to 3 seconds), in this the fan is cut off, and the balls (if any) are at the bottom of the auto-balancer;

– the section of racing (section Ia, from 3 to 4 seconds), in this the fan is turned on and goes to the cruising velocity, the balls initially (at low velocities of the rotor) are at the bottom of the rotor and then (when the rotor reaches a certain velocity) they tend to overtake the rotor; after the rotor has reached the cruising velocity, the motion of the rotor itself and the balls in the auto-balancers have not yet been established;

– the section of the cruising unsteady motion (section Ib, from 4 to 8 seconds), in this the fan operates at cruising velocity and, together with the balls (if any), tends to the steady motion;

– the section of the cruising steady motion (section II, from 8 to 20 seconds), in this the fan operates at cruising velocity and, together with the balls (if any), accomplishes the steady motion;

– the section of the run-out (section III, from 20 to 45 seconds), in this the electric motor is cut off; the fan velocity gradually decreases to 0; the balls (if any, and when the rotor velocity is less than the second resonant velocity) come out of auto-balancing positions.

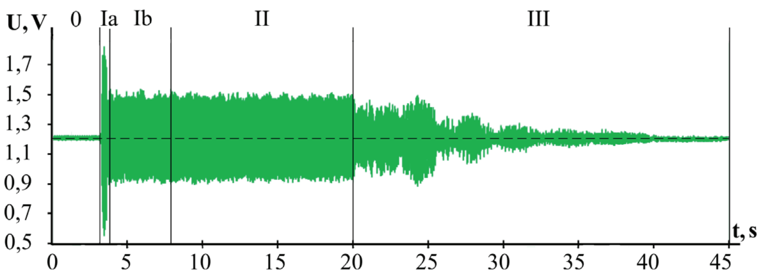
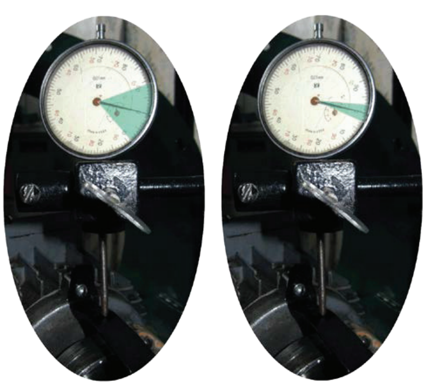


Fig. 5. Division of the fan vibration acceleration diagram into sections (readings taken from the sensor in position 2 for the stand in the 1st configuration): Ia — a racing section; Ib — a section of the cruising unsteady motion; II — a section of the cruising steady motion; III — a run-out section

In the cruising steady motion (section III), the range of residual vibrations is measured with the help of mechanical sensors of the watch type ICH-05 (Ukraine) GOST 577-68 (TU2-034-611-84) (Fig. 6).



a *b*

Fig. 6. The range of residual vibrations on the mechanical sensor: *a* — in the absence of auto-balancers; *b* — if there are 2 auto-balancers

The general behavior of the fan is investigated in the experiment, so the numerical measurements of vibration accelerations are not carried out.

Experiment 3. Investigation of the influence of auto-balancers on the sections of the racing and the run-out

The readings are taken from the sensor 3. The USB oscilloscope is started in the recorder mode. The racing and the run-out are studied according to the vibration acceleration diagrams.

Experiment 4. Estimation of the effectiveness of balancing

The USB oscilloscope is started in the oscilloscope mode. The range of vibration accelerations is measured on the vibration acceleration diagram on the section of the cruising steady motion.

For each stand configuration:

– n_e starting are conducted and the average (arithmetic) value of the range of vibration accelerations U_{avj}, $/ j=\overline{1,4}\,/$ is determined; the averaging results are recorded in Table 2;

– the efficiency (in %) of the work of auto-balancers is determined (Table 3):

$$\Delta j=(1-U_{avj}/a_{v1})\ 100\ \%,\ j=2.3.4). \qquad (4)$$

According to the results of experiments in different configurations of the stand, the conclusions are made, about the sections of the racing and the run-out of the fan, about the effectiveness of the auto-balancers operation.

5. The results of studies of the process of the static and dynamic balancing and their effectiveness

5. 1. General estimation of the stand

Since,

$$\frac{S_{im}}{S_{maxim}}\cdot 100\,\%=\frac{824.5}{878.57}\cdot 100\,\%=94\,\%,$$

$$\frac{S_{sh}}{S_{maxsh}}\cdot 100\,\%=\frac{733.5}{776.42}\cdot 100\,\%=95\,\%, \qquad (5)$$

the balancing capacities of auto-balancers from the side of the impeller and the shank correctly matched with a margin of more than 5 % for the respective imbalances.

5. 2. Vibration characteristics of the stand and source of vibration

Analysis of the results of the first experiment.

On the spectral analyzer (Fig. 7), there is a single harmonic of oscillations (a clear peak is observed) with a rotor frequency of the electric motor of 25 Hz (Fig. 7, *a*), and, practically, there are no other harmonics. This is explained by a balancing of the electric motor rotor, according to the 2nd class of accuracy and the noiseless operation of the latter.

Thus, the rotation of unbalances with the fan frequency is the only source of vibrations. This can be seen from:

– the graph of the spectral analyzer (Fig. 7, *a*);

– the oscillogram of vibration accelerations (Fig. 7, *b*), on which the correct sinusoid is created; the latter oscillogram, implies the presence of ~~a~~ one source of vibration and it is the unbalance of the impeller.

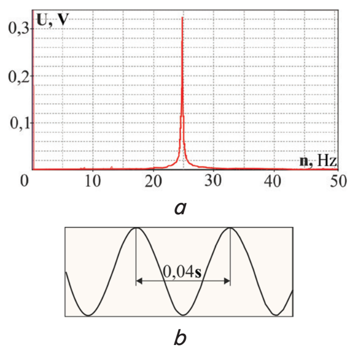


Fig. 7. Vibration analysis, that is characteristic of the stand in all configurations: *a* — spectrogram; *b* — oscillogram of vibration accelerations

On the vibration acceleration diagram (a part of the run-out section of the rotor, Fig. 8), two clear peaks are observed, that is, the fan has 2 resonant velocities. At each peak, the period of oscillations is determined: $T_1=0.119$ s is at the first peak, $T_2=0.079$ s is at the second peak. The frequencies of oscillations on the peaks, respectively, are equal to:

$$n_1=1/0.119=8.4 \text{ Hz}, \; n_2=1/0.079=12.7 \text{ Hz}.$$

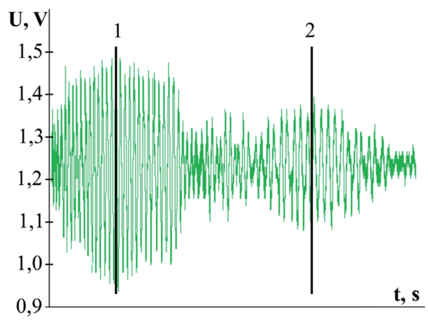


Fig. 8. The vibration acceleration diagram on the run-out

Determination of resonant velocities with vibration sensors is more visual, than with multi-turn indicators [10], and it leaves behind the vibration acceleration diagram.

5. 3. Estimation of transition processes in different stand configurations

5. 3. 1. Analysis of the results of the 2nd experiment

1. Sections of starting and racing of the rotor.

At startup, the balls are almost immobile relative to the impeller casing and they are at the bottom of the auto-balancer (the roughness of the running track is minimal and the forces of gravity are greater than the viscous drag forces). In this, the balls (under the action of tangential forces) move along the running track in the direction opposite to the rotation of the rotor relative to the corps of the auto-balancer.

Then (when the rotor achieves a velocity at which frictional and viscous drag forces counterbalance the gravity forces of the balls), the balls begin to rotate around the axis of rotation of the rotor and try to catch up with it.

2. A section of the cruising unsteady motion. After the rotor reaches the cruising velocity, the balls first quickly catch up with the rotor, and then slowly come to the positions, in which they balance the unbalance of the rotor.

3. A section of the cruising steady motion. The balls together with the rotor rotate as one unit.

4. A section of the run-out. The rotation velocity of the rotor gradually decreases to 0 at run-out. The balls first move with the rotor as one unit, and then (after the rotor passes through the second resonant velocity) they leave the auto-balancing positions and introduce additional unbalance into the system. When the rotor reaches a certain velocity, the balls assemble at the bottom of the auto-balancer under the action of gravity forces. After the rotor stops, the balls slow down (making oscillatory motions) under the action of tangential forces and also stop after a while.

In Fig. 9, the vibration acceleration diagram is shown (similar to that in Fig. 5) for the stand in configuration 4. When we take the readings from the sensor 1, similar (qualitatively) diagrams are obtained, but with larger amplitudes. From the obtained diagrams it follows that:

– in the absence of auto-balancers, the worst characteristics are obtained at all sections of the vibration acceleration diagram;

– with one auto-balancer (from the shank or impeller side), the results are better than in the previous configuration, but each auto-balancer improves the characteristics only from its side;

– with two auto-balancers, the best characteristics are obtained at all sections of the vibration acceleration diagram.

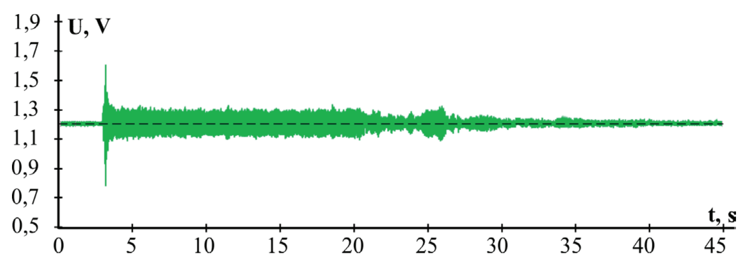


Fig. 9. The vibration acceleration diagram of the balancing process (the recorder mode) from the sensor 2 for the stand in the configuration 4

Similar trends also occur in the presence of additional masses.

The readings from the sensor in position 3 give the integral characteristics for the sensors in positions 1 and 2.

In the study of the transition processes, subsequent experiments are carried out without additional masses and readings are taken only from the sensor 3.

5. 3. 2. Analysis of the results of the 3-rd experiment

Below the voltage diagrams are shown, which characterize the vibration state of the fan during its operation, on sections of the racing (Fig. 10) and the run-out (Fig. 11, 12) at the corresponding time interval.

The vibration accelerations are estimated from the value of the voltages at peaks 1 and 2. In Table 1, it is shown, how much (in %) the value of the voltages and also the transit time of the corresponding peak decrease (the configuration results 2–4 are compared to the configuration results 1).

Analysis of the results of the 4th experiment.

The ranges of vibration accelerations at cruising velocity on the steady motion were put down in Table 2.

The estimation of vibration velocities (Table 3) is carried out in accordance with GOST 11442-90.

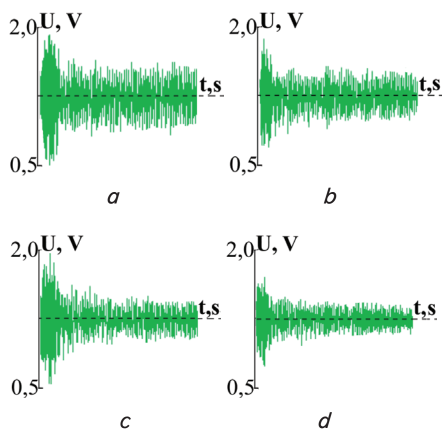


Fig. 10. The voltage diagram on sections of the racing and the unsteady cruising motion in the absence of additional masses: *a* — without auto-balancers; *b* — with an auto-balancer in the impeller; *c* — with an auto-balancer in the shank; *d* — with two auto-balancers

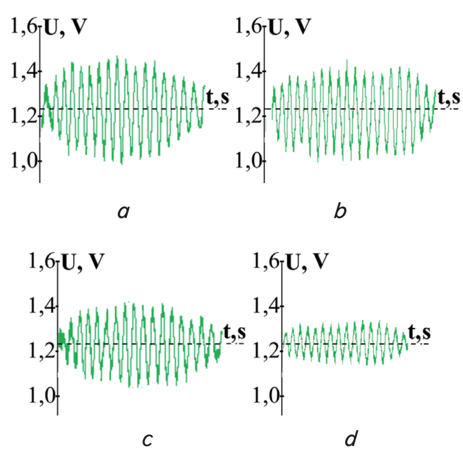


Fig. 11. The voltage diagram in the run-out section (peak 1) in the absence of additional masses: a — without auto-balancers; b — with an auto-balancer in the impeller; c — with an auto-balancer in the shank; d — with two auto-balancers

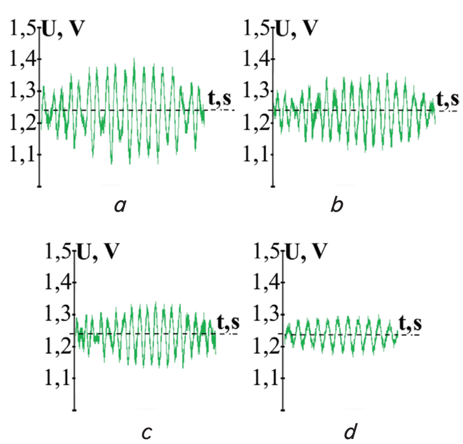


Fig. 12. The voltage diagram in the run-out section (peak 2) in the absence of additional masses: *a* — without auto-balancers; *b* — with an auto-balancer in the impeller; *c* — with an auto-balancer in the shank; *d* — with two auto-balancers

Table 1

Characteristics of the diagram of vibration accelerations in sections of the racing and the run-out

The configuration of the stand		The racing section		The run-out section			
				The peak I		The peak II	
		U, V	ΔU, %	U, V	ΔU, %	U, V	ΔU, %
1	without auto-auto-balancers	1.26	0	0.62	0	0.33	0
2	an auto-balancer on the impeller side	1.10	–13	0.56	–10	0.30	–9
3	an auto-balancer on the shank side	1.43	13	0.48	–23	0.25	–24
4	two auto-ba-lancers	1.17	–7	0.38	–39	0.23	–30

Table 2

The results of the experiments

The configuration of the stand		Without additional masses			With additional masses		
		Positions and indications of the sensor (U_{mid}, V)					
		1	2	3	1	2	3
1	without auto-auto-balancers	0.36	0.24	0.747	0.22	0.127	0.267
2	an auto-balancer from the impeller side	0.157	0.24	0.347	0.14	0.133	0.13
3	an auto-balancer from the shank side	0.28	0.17	0.733	0.17	0.053	0.25
4	two auto-balancers	0.067	0.053	0.137	0.047	0.05	0.117

Table 3

Estimation of vibration velocities and efficiency of their reduction

The configuration of the stand		Without additional masses					
		The sensor position					
		1		2		3	
		V, mm/s	ΔV, %	V, mm/s	ΔV, %	V, mm/s	ΔV, %
1	without auto-balancers	10.22	0	6.81	0	21.2	0
2	an auto-balancer from the impeller side	4.45	56	6.81	0	9.84	54
3	an auto-balancer from the shank side	7.95	22	4.83	29	20.82	2
4	Two auto-balancers	1.89	81	1.51	78	3.88	82
The configuration of the stand		With additional masses					
		The sensor position					
		1		2		3	
		V, mm/s	ΔV, %	V, mm/s	ΔV, %	V, mm/s	ΔV, %
1	without auto-balancers	6.3	0	3.6	0	7.57	0
2	an auto-balancer from the impeller side	4.0	36	3.79	–5	3.7	51
3	an auto-balancer from the shank side	4.83	23	1.51	58	7.1	7
4	Two auto-balancers	1.33	79	1.42	61	3.31	57

The vibration velocities are estimated from the value of the voltages for each sensor position in the absence (Fig. 13, *a*) and the presence of additional masses (Fig. 13, *b*). On the graphs (Fig. 13), the variation (in %) of the value of the vibration velocity is shown (the configuration results 2–4 are compared to the configuration results 1).

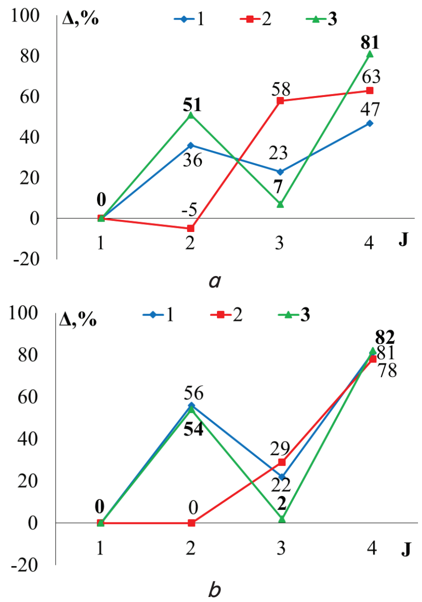


Fig. 13. Graph of the effectiveness of the static and dynamic balancing of the fan stand with its various configurations: *a* — in the absence of additional masses; *b* — in the presence of additional masses; 1 — sensor from the impeller side; 2 — sensor on the shank side; 3 — sensor on the protective casing

From Table 3 it follows that:

– arising vibration velocities (configuration 1, sensor 3) exceed the limit permissible values of 6.3 mm/s (GOST 11442-90) 3.4 times;

– the efficiency of using two auto-balancers (configuration 4, sensor 3) is confirmed: the vibration velocities decrease in relation to the case of the absence of auto-balancers 5.5 times; and, in relation to the limit permissible values, they decrease 1.6 times.

6. Discussion of the research results about the balancing process of the axial fan impeller by ball auto-balancers

Spectral analysis of vibration accelerations at the rated rotor frequency shows that vibrations occur with the same frequency of 25 Hz (the impeller frequency of rotation). This is explained by the following: usual and aerodynamic unbalances are the main sources of vibrations and they are similar to each other. Therefore, both of these unbalances give rise to vibrations with the rotor frequency.

The study of vibration accelerations on the run-out of the rotor shows that the impeller has two resonant frequencies, although the impeller (separately taken) is a short rotor and has, respectively, one resonant frequency. This is explained by the following: the impeller is mounted into a massive corps and dynamically it behaves like a long rotor. The larger resonant frequency is equal to 12.7 Hz and this is almost 2 times less than the operating frequency of the impeller rotation. Therefore, the working frequency of the impeller rotation falls in the area of the beginning of balancing with a reserve.

The analysis of the transition processes in various stand configurations during the racing showed that the presence of one auto-balancer, both from the impeller side and from the shank side, does not worsen the process of the fan racing. This is explained by the following: the rotor racing is short (less than 1 second); when the rotor is started, the balls are located at the bottom of the auto-balancer and they are captured to the motion only at high rotor velocities (above the second resonant velocity); the balls very quickly come in auto-balancing positions. Thus, the balls in the racing section do not introduce additional unbalance into the system.

Analysis of the transition processes in various stand configurations during the run-out showed that on the peaks:

a) the presence of one auto-balancer reduces:

– the value of vibration accelerations from the impeller side by 20 % and 73 %, and also the value of vibration accelerations from the shank side by 20 % and 43 %, respectively;

– the duration of the passage of the peaks from the impeller side by 43 % and 19 %, and also the duration of the passage of the peaks from the shank side by 43 % and 16 %, respectively;

b) the presence of two auto-balancers shows the best characteristics:

– the value of vibration accelerations decreases by 80 % and 60 %;

– the duration of the passage of the peaks decreases by 48 % and 49 %, respectively.

These are explained by the following: in auto-balancers at the run-out there is retention of the balls and the balls are retained in the balancing positions almost until the rotor stops. And this reduces the vibration accelerations at the run-out.

Since auto-balancing occurs after passing through resonant velocities, it can be concluded that dynamic balancing occurs faster than static balancing.

Checking of the effectiveness of the static and dynamic balancing (both in the absence and in the presence of additional masses, attached to the fan protective casing) on the cruising steady motion section showed that:

a) one auto-balancer, mounted from the impeller side:

– significantly reduces vibration accelerations in its plane (by 36–56 %);

– almost does not reduce vibration accelerations from the shank side (by 0–5 %);

– significantly reduces vibration accelerations on the protective casing (by 51–56 %);

b) one auto-balancer, mounted from the shank side:

– almost does not change the vibration accelerations on the protective casing (by 2–7 %);

– slightly reduces the vibration accelerations from the impeller side (by 22–23 %);

– significantly reduces vibration accelerations in its own plane (by 29–58 %);

c) two auto-balancers significantly reduce vibration accelerations:

– from the impeller side by 47–81 %;

– from the shank side by 63–78 %;

– on the protective casing by 81–82 %.

Attachment of additional masses to the fan protective casing in the absence of auto-balancers:

– reduces the value of vibration accelerations (they are almost inversely proportional to the total mass of the moving parts of the fan);

– does not reduce the unbalances of the impeller and, as a consequence, does not reduce the load on the bearings.

The obtained results can be applied for:

– an improvement of the vibration characteristics of the fans of the VO 06-300 series at the manufacturing stage;

– an increasing of the efficiency of the fan static and dynamic balancing, during operation or repair works.

The main drawback of the conducted studies is the possibility of using the obtained results only for a certain series of fans. However, it is compensated by the fact that this series has a wide range of sizes and the results can be applied to any of them.

In the future, similar studies are planned for other types of rotor machines.

7. Conclusions

1. At the nominal frequency of the impeller, the only source of vibration is unbalance (usual and aerodynamic), which causes harmonic oscillations with a rotor velocity of 25 Hz. Two resonant frequencies are detected at the run-out of the rotor, the resonant frequencies are equal to n_1=8.4 Hz and n_2=12.7 Hz, this fact corresponds to a long rotor on isotropic supports.

2. Analysis of the transition processes in various configurations of the fan stand showed that:

– the presence of auto-balancers does not worsen the vibration state of the fan in the racing section;

– the presence of auto-balancers improves the vibration state of the fan in the run-out section (the greatest accelerations decrease by 60–80 %, and the time of passage of peaks decreases by 48 %);

– to improve the vibration characteristics of the fan in the sections of the cruising steady motion, it is advisable to balance its impeller dynamically with two auto-balancers.

3. The use of two auto-balancers is the most effective. In the absence of auto-balancers, vibrations occur, exceeding 3.4 times the limit permissible values in accordance with GOST 11442-90. The presence of two auto-balancers reduces vibrations 1.6 times in relation to the limit permissible values and 5.5 times in relation to the case of the absence of auto-balancers.

4. Attachment of additional masses to the fan protective casing reduces its vibrations, but does not reduce the load on the bearings.

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MODELING OF FRACTURE SURFACE OF THE QUASI SOLID-BODY ZONE OF MOTION OF THE GRANULAR FILL IN A ROTATING CHAMBER

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Розглянуто задачу визначення межі переходу руху з колових на квазіпараболічні траєкторії частинок зернистого заповнення циліндричної камери, що обертається навколо горизонтальної осі. Поведінку заповнення описано за допомогою осереднених величин. Застосовано пластичну реологічну модель зернистого середовища. На основі розрахунку напружено-деформованого стану визначено положення поверхні переходу, як лінії ковзання картини руху. Виявлено залежності межі від швидкісного, геометричних і реологічних параметрів системи

Ключові слова: зернисте заповнення, обертова камера, квазітвердотільна зона, поверхня руйнування, лінія ковзання

Рассмотрена задача определения границы перехода движения с круговых на квазипараболические траектории частиц зернистого заполнения цилиндрической камеры, вращающейся вокруг горизонтальной оси. Поведение заполнения описано с помощью осредненных величин. Применена пластическая реологическая модель зернистой среды. На основе расчета напряженно-деформированного состояния определено положение поверхности перехода, как линии скольжения картины движения. Выявлены зависимости границы от скоростного, геометрических и реологических параметров системы

Ключевые слова: зернистое заполнение, вращающаяся камера, квазитвердотельная зона, поверхность разрушения, линия скольжения

1. Introduction

The main equipment in the large-tonnage processing of granular materials in chemical, mining, metallurgical, building materials industry, power engineering and many other sectors of industrial production is still traditional machines of the drum type [1]. The main varieties of such a wide class of machines are rotary kilns [2] and coolers [3], drying drums [4], ball [5] and tube [6] mills.

A total number of different materials, which annually undergo processing in such machines in the world, are several tens of billions of tons. A widespread use of machines of this class is due to the possibility of their large performance capacity, reliability in operation, ease of use, versatility and high efficiency.

However, a paradoxical feature of this technological equipment is the combination of the utmost simplicity of design and behavior of the treated medium, which is extremely difficult to describe. Significant difficulties in predicting the implementation of processes in drum machines considerably limit a possible technological scope of their application.

Given that the implementation of working processes in the drum-type machines is predetermined by the character of motion mode of the treated granular material in a chamber, the task on establishing the patterns of change in the parameters of this mode appears to be rather relevant.

2. Literature review and problem statement

Motion modes of granular fill in the rotating chambers significantly affect the realization of technological processes and energy efficiency of the drum-type machines drive [7].