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Technical note

EVALUATION OF THE PARAMETERS OF VIBRATIONS DURING DYNAMIC BALANCING OF THE FAN ROTOR ACCORDING TO POLISH AND INTERNATIONAL STANDARDS

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In the paper a procedure and methodology of acceptable calculations of powers, displacements (of amplitudes) and velocities of pulses according to ISO standards depending on the rotational speeds (frequencies) of rotor during attempts of dynamic breaking down taking into account effectiveness of vibration isolation were described. The proposed method is regarding one-sidedly hung rotors appearing in such appliances as rotors of fans, blowers, pumps and others. As permissible parameters of pulses values which are given in international standards PN-ISO-10816-1:1998P and PN-ISO-7919-1:2001 were assumed. Calculations were compared in the form of tables and they depicted with relevant graphs which have the character of useful nomograms. Also, recommendations for the design of the foundations and guidelines related to dimensions of the foundation, the weight and the way of fixing the device were given. In the end also closing remarks and conclusions were given.

Key words: parameters of vibrations, ISO standards, fan rotors, dynamic forces, dynamic balancing.

1. Introduction

The paper presents a method of calculations for the determination of suitable parameters of dynamic balancing of fan rotors, i.e., acceptable dynamic forces, displacements (amplitudes) of vibrations, acceptable effective values of displacements and vibration rates depending on rotational speed (frequency) of the rotor during dynamic balancing. It can be important during designing a stand for repairs of rotor disks (fans, pumps, blowers and so on), and next for their dynamic balancing. For this purpose it is necessary to determine values of mass of components and their static and dynamic centres of gravity, values of .dynamic loadings, coefficients of transfer, resonance speeds, effectiveness of anti-vibration insulation, values of efficient displacement (amplitudes) and speed of vibration. A procedure of preliminary calculations of the foundation and an acceptable pressure of the foundation should be also defined; they will be a base for requirements for design and manufacturing.

This paper presents a suitable procedure of calculations for a case of dynamic balancing of unilaterally suspended rotor wheels, for example those for draught fans in power steam boilers. The rotors are mounted on a special steel structure placed on suitable rubber pads, and it is fixed to the concrete foundation with special anchor bolts. Exemplary calculations were made for a rotor for fans of one-sidedly hung type, tested together with the shaft and the bearing housing. Critical values of the mentioned quantities were determined for the maximum unbalance caused by *1 kg* of mass at the radius *315 mm*. A machine support was assumed as a rigid one belonging to the 1st group of devices according to the international standards ISO 10816 and ISO 7919. Lift of the shaft axis, resulting from the considered structure is equal to *713 mm* and it is greater (acccording to the Standards ISO/TR 19201:2013, PN-ISO-10816-1:1998P and PN-ISO-7919-1:2001P [1-3], than *315 mm* (*H* > *315mm*). Calculations were made on the basis of papers (PN-EN

1997-1:2008P [4]; Team Work [5]; Gosiewski and Muszyńska [6]; Sklarow and Gulajew [7]; Goliński [8]; PN-ISO 14695:2008P [9]; ISO 1940-1:2003 [10]; ISO 11342:1998 [11]; Śloderbach and Witoś [12]) for the range of the shaft rotational speed (*0 ÷ 25 Hz*). According to the ISO standards, acceptable efficient values of displacements (amplitudes) are *90 μm*, and acceptable efficient rates of vibrations are up to *7.1 mm/s.*

In the paper, it has been shown that for the assumed maximum unbalance *0.513 kgm* in the range from zero to the maximum rotational speed of the rotor the acceptable (according to ISO standards) effective values of displacements (amplitudes) and speed of vibrations will not be exceeded. Some chosen results of the calculations have been presented in a table and as diagrams as well. The diagrams can be used as nomograms during tests of balancing and in service practice. In this paper, the author uses the available Polish and International literature, containing basic information on dynamics and dynamic balancing, as well as numerous, see (Polish and International Standards [1-4]; [9-11]) concerning vibrations, their measurements, applied nomenclature, and containing basic information about rotating machines such as fans, electric machines, steam turbines and others.

2. The procedure of calculations

The applied procedure of calculations includes the following tasks:

- determination of component masses and determination of position of static and dynamic centres of gravty ($\boldsymbol{S}_Q$ and $\boldsymbol{S}_{Fd}$),
- determination of dynamic forces and analysis of their influence on stability of a device,
- determination of the transfer coefficient (a ratio of amplitude of the transferred force to amplitde of the exciting force),
- calculation of effectiveness of anti-vibration insulation,
- calculation of effective and total amplitudes (displacement) of vibrations,
- calculation of the maximum effective speed of vibration,
- formulation of the table containing calculated values and suitable graphs,
- preliminary calculations of the foundation including the minimum weight,
- determination of the acceptable pressure of the foundation on the ground and its minimum surface,
- formulation of some important remarks, recommendations and conclusions for the user,
- determination of total mass of the test stand (device).

3. Determination of total and component masses and dynamic forces

A scheme of the test stand for dynamic balancing of unilaterally suspended rotors with the given suitable physical quantities and a method of fastening to the foundation as well as a scheme of the foundation have been shown in Fig.1. The basic data used for the determination of the test method, description and realization of calculations were taken from Polish and International Standards [1-3].

In Fig.1 (a, b, c, d, H) – characteristic dimensions of the structure are given in [*mm*],

$a = 300$, $b = 500$, $c = 350$, $d = 1550$, $H \cong 713$.

- rotor with no blades $m_{wr} \cong 1450$ *kg*, shaft of the rotor $m_{wl} \cong 210$ *kg*.

The total swirling mass $m_w \cong 1660$ *kg*. For calculations it was assumed that $m_w = 1700$ *kg*,

- bearing housing $m_{ol} \cong 600$ *kg*,

- carrying frame $m_r \cong 700$ *kg*, because $m_r = (m_{rl} + m_{rp}) \cong 600 + 100 \cong 700$ *kg*,

where m_{rl} – mass of the left side of the frame, and m_{rp} – mass of the right side of the frame related to the point C. Total mass of the device (without the foundation) $m_c \cong 3000$ *kg*.

Total weight of the device $Q_0 \cong 30$ *kN*.

Note: During dynamic balancing the rotor has no working blades.

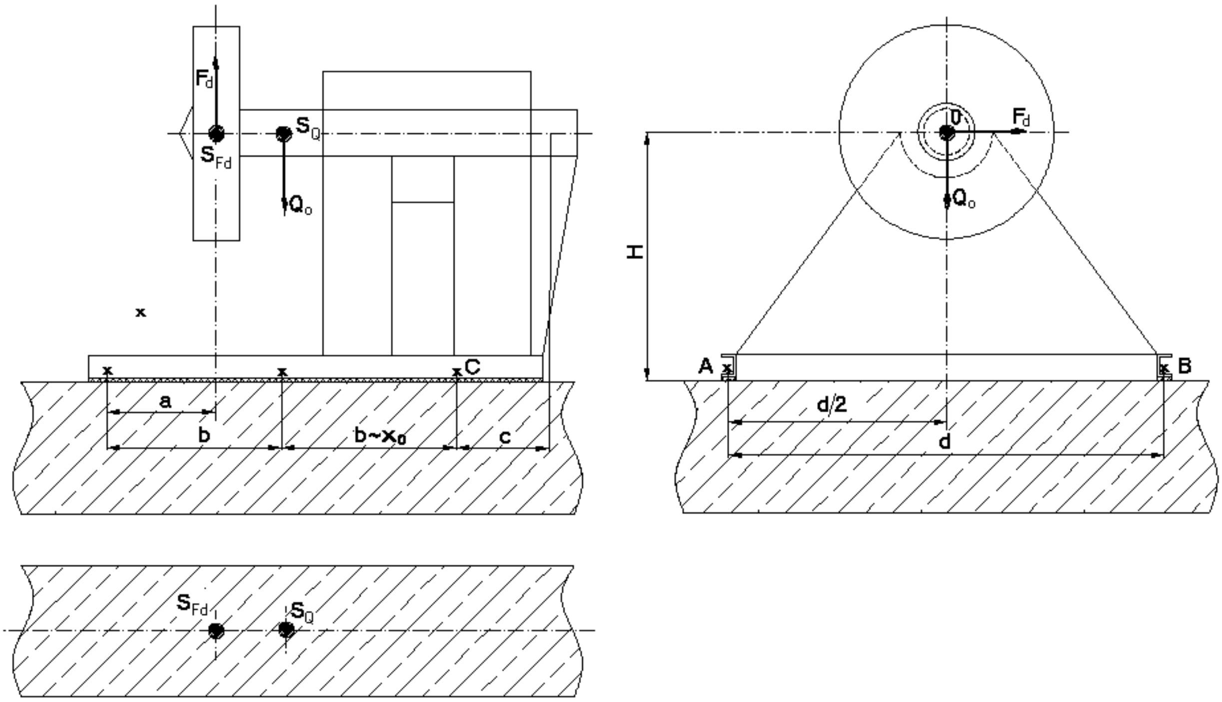

Fig.1. Scheme of the test stand for dynamic balancing of unilaterally suspended rotors.

3.1. Determination of location of the device centre of mass - S_Q

The following positions of the force moments related to the point C result from Fig.1

$\Sigma M_C: (2b\text{-}a)m_{wr} + bm_{wl} + b/2 \cdot m_{ol} + b/2 \cdot m_{rl} - cm_{rp} = Q_0 \cdot x_0$ [*kgm*],

$\Sigma M_C: 0.7m_{wr} + 0.5m_{wl} + 0.25m_{ol} + 0.25m_{rl} - 0.35m_{rp} = Q_0 \cdot x_0$ [*kgm*],

Thus, $x_0 \cong 0.51\ m$ or $x_0 \cong 511\ mm$.

The designed lift of the fan shaft axis is $H = 713\ mm$, so $H > 315\ mm$ (according to ISO Standards [1]). According to the International and Polish Standards (ISO/TR 19201:2013 [1]; PN-ISO-10816-1:1998P [2]; and PN-ISO-7919-1:2001P [3]) we assume that the machine suport is rigid, and the device is classified into the first group.

3.2. Determination of the maximum centrifugal forces

- maximum rotations of the rotor $n_{max} = 1500\ rev/min$ or $n_{max} = 25\ Hz$,
- maximum angular velocity $\omega_{max} = 2\pi n_{max} \cong 9420\ rad/min$ or $\omega_{max} \cong 157\ rad/s$.

$F_{d1} = m_w \cdot \rho \cdot \omega^2_{\max}$ - dynamic force from eccentricity of the rotor mass,

ρ - radius of eccentricity of the fan rotor mass, $\rho \approx 0.3\ mm$, according to [2],

Thus $F_{d1} \cong 1700\ kg \cdot 0.3 \cdot 10^{-3}\ m \cdot (157\ rad/s)^2 \cong 12.571\ kgm/s^2$, so $F_{d1} \cong 13\ kN$,

$F_{d2} = m_0 \cdot \rho_0 \cdot \omega^2_{\max}$ - dynamic force from the assumed unbalance of the rotor where

m_0 – maximum mass of unblance.

According to the results of experiments, it was assumed that $m_c = 1\ kg$,

ρ_0 – radius of dynamic unbalance.

According to DTR for AP1-22/12, according to [13], $\rho_0 = 530\ mm$,

thus $F_{d2} \cong 1\ kg \cdot 0.53\ m \cdot (157\ rad/s)^2 \cong 13.064\ kgm/s^2$, so $F_{d2} \approx 13\ kN$.

A value of the maximum centrifugal dynamic force of the rotor is calculated from the following equation
$F_{dmax} = F_{d1} + F_{d2}$ [N]. Thus $F_{d\,max} \cong 26\ kN$
At the rotating rotor, the force $\mathbf{F}_d$ can act in three main directions:
- vertical (up and down)
- horizontal (in direction $\perp$ to the rotor axis, horizontally),
- along the rotor axis.

For fundamental bolts and nuts M 24 class {3.6} of mechanical properties was assumed. In the case of such bolts acceptable loading for tension equals *75 kN* according to (Team Work, [5], see page 249, Tab.6, for MSt7 steel), and for shearing *37.5 kN.*

3.2.1. When the force $\mathbf{F}_d$ acts vertically upward, then

$F_d < Q_0$ – condition of stability is satisfied because bo $26<30$ [kN].
ΣM_C: $(2b-a)\cdot F_d < b\cdot Q_0 + 3\cdot 75$ [kNm]. Thus, $\Sigma M_C\ 0.7\cdot F_d < 0.5\cdot Q_0 + 225$ [kNm].
Thus, the maximum reactions R_s extending the fundamental bolts are $R_s \approx 1\ kN$, so $1 << 75$ [kN].

3.2.2. When the force $\mathbf{F}_d$ acts vertically downward, then

Additional pressure to the foundation occurs, so stability of the tested device becomes higher.

3.2.3. When the force $\mathbf{F}_d$ acts horizontally, then

ΣM_B: $H\cdot F_{dmax} < d/2\cdot Q_0$. Thus, ΣM_B: $0.713F_{dmax} < 0.775Q_0$ – the condition of stability is satisfied because $18.5 < 23.5$ [kNm].
ΣM_C: $(2b-a)F_{dmax} < 2b\cdot 75 + b/2\cdot 75 + d/2\cdot 35 + 0.5Q_0\mu_s$ or
ΣM_C: $0.7\cdot F_{dmax} < 75\cdot 1\ m + 75\cdot 0.5\ m + 37.5\cdot 0.775\ m + 0.5Q_0\cdot\mu_s$. The condition of stability is satisfied because $18.5 < 1070$ [kNm], and the condition $F_{dmax} < Q_0\cdot\mu_s + 3\cdot 75$ is satisfied because $26<243$ [kN].
Here μ_s is the coefficient of static friction of steel against rubber. It was assuned that $\mu_s \approx 0.6$ (or more), according to (Team Work, [5], see page 200, Tab.5).

3.2.4. When the force $\mathbf{F}_d$ acts along the rotor axis, then

The condition $F_{dmax} < Q_0\cdot\mu_s + 3\cdot 75$ is satisfied, as in the case presented above because $26 < 243$ [kN].

4. Determination of the transfer coefficient ζ_0 – according to Golinski [8]

$$\varsigma_0 = \frac{F_p}{F_d} = \sqrt{\frac{1+\left(2\gamma_g\cdot\mu\right)^2}{\left(1-\mu^2\right)^2+\left(2\gamma_g\cdot\mu\right)^2}} \tag{4.1}$$

where F_p, F_d –amplitudes of the transferred force and the exciting force, respectively,
γ_g – damping coefficient for rubber, assumed as 50^o Sh, $\gamma_g \approx 0.1$ according to [4],
$\mu = \dfrac{n}{f}$ – coefficient of the ratio of frequency of forced vibrations to free vibrations,
n – frequency of rotor rotations in Hz,
f – frequency of free vibrations of the rotor or a device on vibroinsulators in Hz.

4.1. Determination of frequency of flexural vibrations *f* for a unilaterally suspended rotor

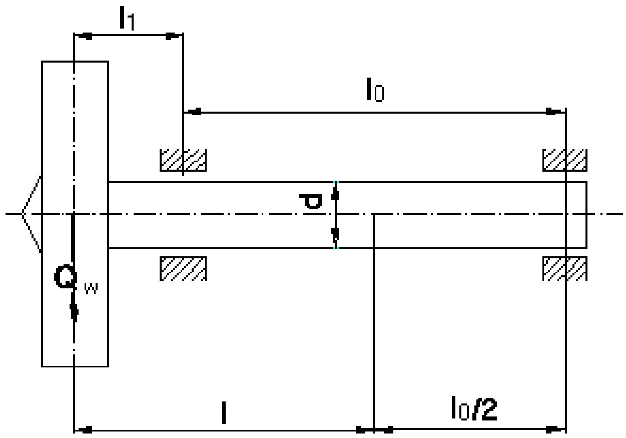

Fig.2. Scheme of the unilaterally suspended rotor.

According to papers [6, 8, 11], f can be expressed as

$$f \cong 300\sqrt{\frac{3E \cdot J}{Q_w \cdot l^3}} \text{ [rot/min]} \quad \text{or} \quad f = 5\sqrt{\frac{3E \cdot J}{Q_w \cdot l^3}} \text{ [Hz]}, \tag{4.2}$$

where $E \approx 2.1 \cdot 10^5$ MPa – Young's modulus for steel

$$J = \frac{\pi \cdot d_w^4}{64} \; [m^4] \tag{4.3}$$

where J - moment of inertia of the rotor shaft section, $d_w = 20\ mm$ – diameter of the shaft,
Q_{wr} – weight of the rotor, $Q_{wr} \approx 1.450\ kG$ or $Q_{wr} \approx 14.5\ kN$,
l – the characteristic length, see Fig.2, here calculated according to (Team Work [13]), is $l \approx 0.316 + 0.188 \approx 0.505\ m$ or $l \approx 505\ mm$. Thus we obtain $f \approx 84$ Hz from Eq.(4.2).

4.2. Determination of vibration frequency f_w – for a device on rubber vibroinsulators

There is the following formula for f for this case according to paper [8]

$$f_w \approx \frac{5}{\sqrt{\delta_{st}}} \text{ [Hz]} \tag{4.4}$$

where δ_{st} is the static deflection of rubber in [m]. It should satisfy the following conditions $\delta_{st} \le (0.1 \div 0.2)h$, see [8], and

$$\delta_{st} = \frac{\sigma \cdot h}{E_{st} \cdot C_0 + \sigma} \; [m] \tag{4.5}$$

where $\sigma = \frac{Q_c}{2a_s \cdot b_s}$ - total unit pressure to rubber in [MPa],
Q_c – total force (static and dynamic) acting on the foundation.

$Q_c = Q_0 + F_{dmax}$, so $Q_c = 30\ kN + 26\ kN = 56\ kN$,
here a_s, b_s and h – dimensions of the rubber pad:
a_s- length, b_s- width and h – height in [m]. Thus, $\sigma \approx 0.255$ MPa,
E_{st} – Young's modulus of elasticity for rubber, $E_{st} \approx 4.5$ MPa according to [8],
C_0 – equivalent shape factor

$$C_0 = 1 + m\frac{a_s \cdot b_s}{2h \cdot (a_s + b_s)} \tag{4.6}$$

where m – coefficient of contact of the metal plate and the rubber plate $m \approx 4.67$.

From Eq.(4.6), $C_0 \approx 22.4$ was obtained, and from Eq.(4.5) we obtain $\delta_{st} \approx 25.2 \cdot 10^{-6}\ m$ or $\delta_{st} \approx 25.2\ \mu m$. Thus, after calculations we obtain $f \approx 99.6\ Hz$ from Eq.(4).
When $Q_c = Q_o$ - then only static loading occurs coming from the device weight. Then, $\sigma \approx 0.136$ MPa and $\delta_{st} \approx 14 \cdot 10^{-6}\ m = 14\ \mu m$.

Thus, from Eq.(4.4) we have $f \approx 134.2$ Hz. Let us assume that $\sigma = 0.35$ MPa – as in the case of soft rubbers according to [8]. Then, $\delta_{st} \approx 35 \cdot 10^{-6} m = 35\ \mu m$.

Thus, $f \approx 84\ Hz$. It is the value like that obtained in item 4a of this paper.

Let us assume the resonant frequency $f \approx 84\ Hz$ in further considerations and calculations. Thus, it appears that $n < f$ [Hz] because $25 < 84$ [Hz], so $\mu < 1$ because $\mu \approx 0.3$.

It appears that the device for dynamic balancing will work at much smaller rates (frequencies) than subcritical ones (before the resonances) because the area of resonance according to (Goliński [8]) is: $0.6<\mu<1.5$.

After substituting the values γ_g and μ to expression (4.1) we obtain $\zeta_0 \approx 1.1$.
Thus $F_p = F_{pmax} \approx 1.1 \cdot F_{dmax} \approx 28.6\ kN$, so it appears that $F_{pmax} < Q_0$ because $28.6 < 30\ kN$.

Thus, the maximum force transferred to the foundation from the dynamic exciting force is lower than the force of static pressure of the all device. Such a case also stabilizes the foundation system of the test device.

5. Determination of effectiveness *Si* of vibroinsulation according to PN-ISO-10816:1998P-1, [2]

$Si = 1 - \zeta_0$ so $Si \approx -0{,}1$. A value of vibroinsulation effectiveness is negative which takes place when $\mu < 1$ and then $F_p > F_d$.

6. Calculation of the total effective amplitude A_{cs} and effective displacement U_s of vibrations

A value of the instantaneous amplitude the (displacement) A is calculated according to the following formula, see (Goliński [8])

$$A = U \cong \frac{(m_w \cdot \rho + m_0 \cdot \rho_0)}{m_c} \cdot \frac{\mu^2}{\sqrt{(1-\mu^2)^2 + (2\gamma_g \cdot \mu)^2}}. \tag{6.1}$$

After substitution of suitable data for the case of the maximum rotational speed n_{max} to the above formula, from the calculation we obtain $U_{max} = A_{max} \approx 0.03314\ mm \approx 33.14\ \mu m$. Thus, $A_{cmax} \approx 0.0663\ mm \approx 66.3\ \mu m$, where A_{cmax}- the maximum value of the total amplitude of vibrations and $A_{cmax} = 2 \cdot A_{max}$.The effective maximum and total amplitude of vibrations is calculated according to the following equqtion

$$A_{s\max} = \frac{A_{\max}}{\sqrt{2}} \approx 23.33\,\mu m.$$

Thus, $A_{csmax} = 2A_{smax} \approx 46.6\ \mu m$, respectively. Thus, $A_{csmax} \approx 47\ \mu m$.

The acceptable effective value of displacement U_{sall} resulting from the effective amplitude of vibrations ($U_s = A_s$) for the 1st group of machines and devices of rigid supports according to international stanards ISO 10816 and ISO 7919 is $U_{sall} = A_{sall} = 90\ \mu m$.

Thus, the conditions $U_{smax} < U_{sall}$ are satisfied; the condition $A_{csmax} < A_{sall}$ is also satisfied.

7. Calculation of the total maximum effective speed of vibrations

According to papers (ISO/TR 19201:2013, [1]; PN-ISO-10816-1:1998P, [2]; PN-ISO 7919-12001P, [3]), the expression for total maximum effective speed of vibrations takes the following form

$$V_{smax} = 2\pi n_{max} \cdot A_{smax} \cdot 10^{-3}\ [mm/s], \tag{7.1}$$

$$V_{csmax} = 2\pi n_{max} \cdot 2A_{smax} \cdot 10^{-3}\ [mm/s]. \tag{7.2}$$

Thus, from Eq.(7.2) we obtain $V_{csmax} \approx 7.2\ mm/s$ and $V_{smax} \approx 3.6\ mm/s$.
The value $V_{sall} = 7.1\ mm/s$ - is given in the standards (PN-ISO 10816-1 and ISO 7919, [2-3]).

The condition of acceptable speeds of vibrations $V_{smax} \le V_{sall}$ is satisfied because $3.6<7.1$ [mm/s], moreover the additional stronger assumed condition $V_{csmax} \le V_{sall}$ is almost satisfied because $7.2 \approx 7.1$ [mm/s]. After recalculation, (Eqs (7.1) and (7.2)) a value of V_{sall} can be used for the formulation of suitable characteristics (diagrams) depending on the rotor rotations n [*rot/min*] or n [Hz]. The recalculation is connected with a certain estimation resulting from the assumption that the calculated forced vibrations are of a harmonic form and they are presented in this form on the diagrams. In the case of real vibration courses which are usually not harmonic, the determined parameters of vibrations are results of averaging (integration) a wide spectrum of real (non-harmonic) mechanical vibrations automatically realized by means of modern electronic measuring devices, see (PN-ISO 7919-1:2001P, [3]; Sklaro and Gulajew, [7]; ISO 1940-12003, [10]). It means that the obtained results cannot be fully comparable with the real ones, but they can be approximate.

7.1. The obtained calculation results

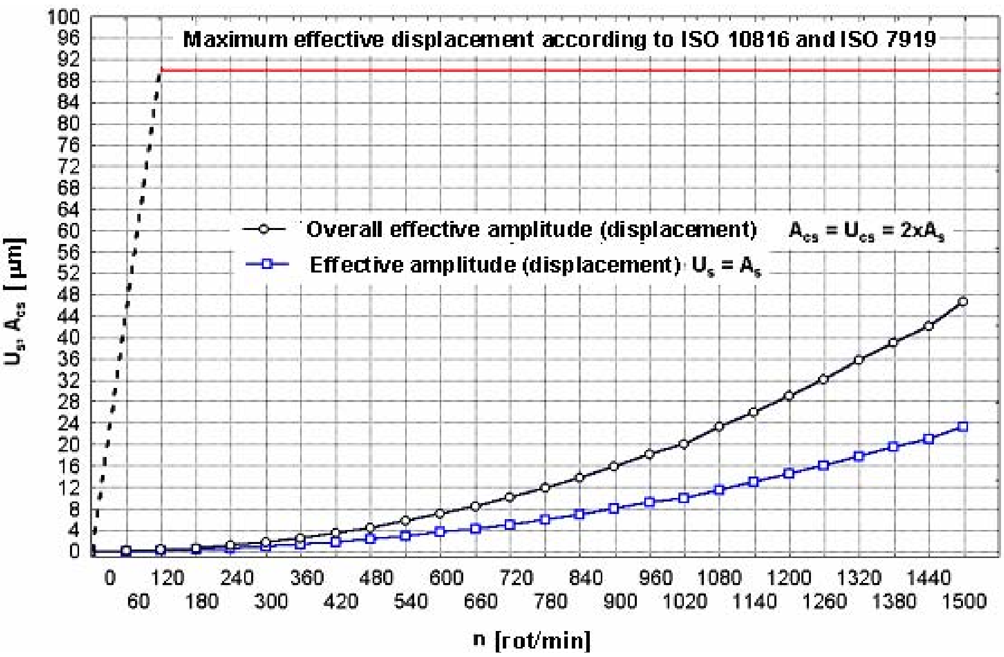

Fig.3. Graphs of the calculated total and effective values of displacement (amplitude) depending on rotational speed of the rotor.

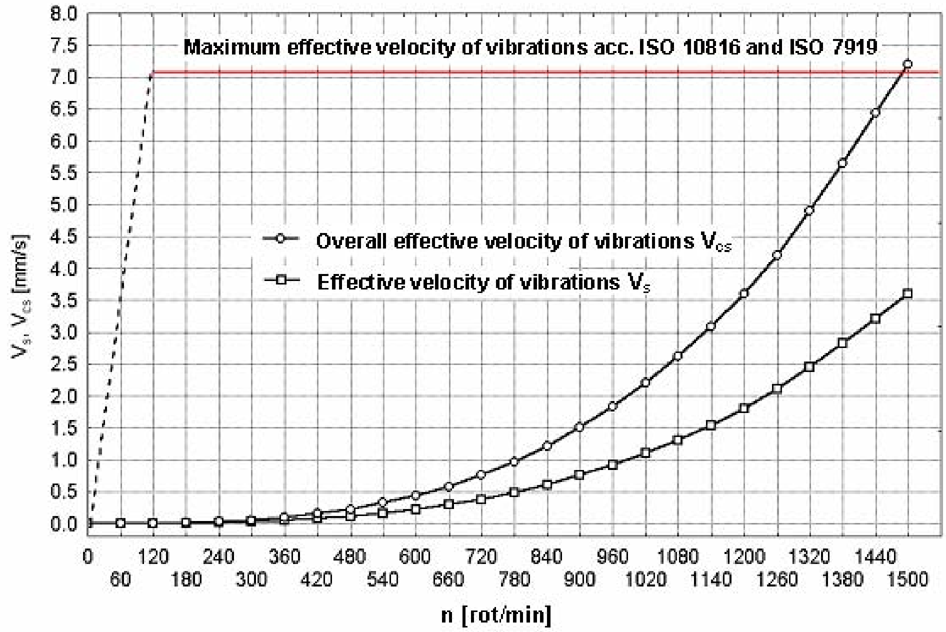

Fig.4. Graphs of calculated total and effective values of velocity of vibrations depending on rotational speed of the rotor.

Note: In the range of rotational speed $0 \le n < 360$ [*rot/min*] or $0 \le n < 6$ [Hz], according to (ISO/TR 19201:2013 [1]) we can assume averaged values for $U_s \approx 1$ μm, $A_{cs} \approx 2$ μ*m* , and for $V_s \approx 0.05$ *mm/s* and $V_{cs} \approx 0.1$ *mm/s*, i.e. the same as for the rotational speed ($n \approx 330$ *rot/min*).

8. Preliminary calculations of the foundation

The preliminary calculations were made according to the papers (Standards ISO/TR 19201:2013; PN-ISO-10816-1:1998P; PN-ISO-7919-1:2001P, [1-3] and Golinski [8]).

The ratio of weight of the rotor together with the shaft Q_w to the total weight Q_{0F} of the device (machine with the foundation) is

$$\frac{Q_w}{Q_{0F}} = \frac{A_{0all}}{1.2\rho} \tag{8.1}$$

where Q_w – weight of the rotor together with the shaft, $Q_w \approx 17$ *kN*.
$Q_{0F} = Q_0 + Q_F$ [*kN*], and Q_F – weight of the foundation, $A_{sall} \approx 90$ μ*m* , $\rho \approx 0.3$ *mm*.

Thus $Q_{0F} \approx 68$ *kN*. From Eq.(8.1) we obtain: $Q_F \approx 38$ *kN* is the required minimum weight of the foundation. We must assume that $Q_F \ge 38$ *kN*.

9. Calculation of the foundation pressure on the ground

According to paper [8], the acceptable pressure on the ground is $p_{all} \approx 4$ N/cm^2.
The general condition of ground loading below the stand is

$$\frac{Q}{P} \le p_{all} \tag{9.1}$$

where Q – general total force of pressure [*kN*]. We have $Q = Q_{0F}$ and $Q = Q_{0Fd}$ alternately.
P – minimum required area of the foundation in [cm^2] lub [m^2].

9.1. Calculation of the minimum area of the foundation for static pressure Q_{0F}

From Eq.(9.1) we obtain $P_{stmin} \geq \dfrac{Q_{0F}}{p_{all}}$, where $Q_{0F} \approx 68\ kN$. Thus, $P_{stmin} \geq 1.7\ m^2$.

9.2. Calculation of the minimum area of the foundation for dynamic loading

In such a case $Q_{0Fd} = Q_{0F} + F_{dmax}$. Thus, $Q_{0Fd} \approx 94\ kN$.

From Eq.(9.1) it appears that $P_{dmin} \approx 2.35\ m^2$ or $P_{dmin} \approx 23.500\ cm^2$ when the force of transfer P_{pmax}, is assumed for calculations because $P_{pmax} > P_{dmax}$. Thus $Q_{0Fp} \approx 97\ kN$. From Eq.(9.1) $P_{dmin} \geq 2.45\ m^2$ or $P_{dmin} \geq 24.500\ cm^2$ were calculated. Thus, we should assume the minimum area of the foundation $P_{dmin} > 2.45\ m^2$ or ($P_{dmin} > 24.500\ cm^2$).

10. Dynamic inverse problem

Calculations carried out according to the presented methodology generate an approximate result, which results from the simplifying assumption that the forcing vibrations are only single harmonic(monotone) and the location of applied forces is known.

The real vibrations of the machine are generated by different sources (generalized force / mass and aerodynamic moments of the unknown exact location) and have a multi-tone nature (they are composed of multiple harmonics). To diagnose the quality of balancing of the rotor and evaluate the technical condition of isolation, measurements and analysis of a selected portion of the spectrum of vibrations are made in papers (PN-ISO-10816-1:1998P [2]; PN-ISO 7919-1:2001P [3]; Śloderbach and Witoś [12]; Team Work [13]). Only the rotational component of vibration f_n or vibrations in narrow band $\langle f_{min}; f_{max}\rangle$ are most often taken into account. (f_{min} is generally greater than zero, f_{max} is most often less than or equal to $2.0\, f_n$). The separation of the desired part of the spectrum is implemented with hardware (using analog filters) or/and with software (analysis of the vibration spectrum, e.g. using algorithm of Fast Fourier transformation FFT).

The cause of the vibration of the machine (e.g. the symptoms of imbalance, the misalignment of the rotor) is determined on the basis of an analysis of amplitude of vibration components: $0.5f_n$, $1.0f_n$, $1.5\, f_n$, $2.0\, f_n$. An analysis of changes of vibration phase angle together with a change of rate of rotation makes it possible to specify the actual distance of working speed of the machine from the resonance range resulting from the state of vibro-isolation.

11. Remarks and conclusions

1. In the paper it has been shown that the maximum dynamic forces from eccentricity of the fan rotor and the assumed maximum unbalance of the order *~513 kgmm* or (*~0.513 kgm*) do not cause exceeding acceptable loadings in relation to the assumed method of fixing the device to the foundation with six foundation bolts.
2. In the whole range of variations of rotational speed of the tested rotor $0 \leq n < 1500$ [*rot/min*] or ($0 \leq n < 25$ [Hz]) there are lower calculation effective values of displacements U_s – from acceptable values $U_{sall} = 90\ \mu m$ (according to international standards (ISO/TR 19201:2013 [1]; PN-ISO-10816-1:1998P [2]; PN-ISO-7919-1:2001P [3]), see item 6. The above results are shown in the graphs in Fig.3.
3. In all the range of variations of rotational speed of the fan rotor there are also lower calculation values of effective speeds of vibrations V_s – from acceptable values V_{sall}. ($V_s < V_{sall}$). Only in the case of the boundary maximum rotational speed $n \approx 1500\ rot/min$ or ($n \approx 25$ Hz) (in practice inaccessible because of a slip occurring in electric motors) there is almost equality of values of total and acceptable effective speeds of vibrations $V_{csmax} \approx V_{sall}$, where $V_{sall} \approx 7.1\ mm/s$ according to standards (ISO/TR 19201:2013;

PN-ISO-10816-1:1998P and PN-ISO-7919-1:2001P [1-3]), and the calculated values are $V_{csmax} \approx 7.2$ *mm/s* and $V_{smax} \approx 3.6$ *mm/s*. The above obtained analytic results are shown in Fig.4.

4. We assume that acceptable test parameters (acceptable effective displacement and permissible effective speed of vibrations) are those given in the mentioned international standards ISO. As it was said, their values (for the 1st group of devices of a rigid support) are up to 90 μm for an effective value of displacements, and up to *7.1 mm/s* for an effective value of speed of vibrations.
5. In the range of rotational speed $0 < n < 120$ [*rot/min*] or ($0 < n < 25$ Hz) for Group I of devices and machines of the class of rigid support according to the PN and ISO standards [1-3], linear approximation (broken line) was made for acceptable limit values of vibrations, i.e., for effective displacement and effective speed of vibrations.
6. In the range of rotational speed $0 < n < 360$ [*rot/min*] or $0 < n < 6$ [Hz], according to [1] we can assume the following averaged values of $U_s \approx 1$ μm, thus $A_{cs} \approx 2$ μm and for $V_s \approx 0.05$ *mm/s* o $V_{cs} \approx 0.1$ *mm/s*, respectively, i.e. the same values as for rotational speed $n \approx 330$ *rot/min*.
7. Items 8 and 9 also contain preliminary calculations of foundation for the calculated maximum static and dynamic forces, i.e. calculations for the minimum mass of the foundation and minimum dimensions of the foundation surface being in contact with the ground. Some preliminary recommendations concerning designing were given.
8. As it was previously said, suitable graphs were prepared in order to illustrate some calculations (see Figs 3 and 4). They seem to be useful nomograms.

Nomenclature

A – instantaneous amplitude of the vibrations (displacement)
A_{max} – maximum value of the amplitude of vibrations
A_{smax} – effective maximum of the amplitude of vibrations
A_{cmax} – maximum value of the total amplitude of vibrations
A_{csmax} – effective maximum value of the total amplitude of vibrations
A_{cs} – total effective amplitude
A_{sall} – allowable effective value of displacement
(a, b, c, d, H) – characteristic dimensions of the stand structure
a_s, b_s and h – dimensions of the rubber pad
C_0 – equivalent shape factor
d_w – diameter of the shaft
E – Young's modulus for steel
E_{st} – Young's modulus of elasticity for rubber
F_{d1} – dynamic force from eccentricity of the rotor mass
F_{d2} – dynamic force from the assumed unbalance of the rotor
F_{dmax} – value of the maximum centrifugal dynamic force of the rotor
f – frequency of free vibrations of the rotor or a device on vibroinsulators in Hz
J – moment of inertia of the rotor shaft section
l – characteristic length
m_c – total mass of the device (without the foundation)
m_0 – maximum mass of unbalance
m_r – mass of carrying frame
m_{0l} – mass of bearing housing
m_{rl} – mass of the left side of the frame related to the point C, see Fig.1
m_{rp} – mass of the right side of the frame related to the point C, see Fig.1
m_{wl} – mass of shaft of the rotor
m_{wr} – mass of rotor with no blades
n – frequency of rotor rotations in Hz
n_{max} – maximum rotations of the rotor
P – minimum required area of the foundation
p_{all} – acceptable pressure to the ground

Q – general total force of pressure
Q_c – total force (static and dynamic) acting for foundation
Q_0 – total weight of the device
Q_F – weight of the foundation
Q_w – weight of the rotor together with the shaft
Q_{wr} – weight of the rotor
Si – effectiveness of vibroinsulation
U_s – effective displacement of vibrations
U_{sall} – the allowable effective value of displacement
V_s – the effective values of speeds of vibrations
V_{sall} – the allowable effective values of speed of vibrations
V_{smax} – values of maximum effective speed of vibrations
V_{smax} – total values of maximum effective speed of vibrations
δ_{st} – static deflection of rubber
γ_g – damping coefficient for rubber
ζ_0 – transfer coefficient ζ_0
μ – coefficient of the ratio of frequency of forced vibrations to free vibrations
μ_s – coefficient of static friction of steel against rubber
ρ – radius of eccentricity of the fan rotor mass
ρ_0 – radius of dynamic unbalance
σ – total unit pressure for rubber in [MPa]
ω_{max} – maximum of the rotor angular velocity

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