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VIBRATION ISOLATION OF VARIABLE FAN SPEED IN HVAC SYSTEMS

WIBROIZOLACJA WENTYLATORÓW ZE ZMIENNĄ PRĘDKOŚCIĄ OBROTOWĄ W SYSTEMACH HVAC

A b s t r a c t

This article analyses two samples of existing vibration isolation systems. The first sample applies to rubber absorbers, the second, to spring absorbers. The article presents a method of setting the parameters of a second order differential equation describing oscillation movement. The equation parameters were selected on the basis of amplitude measurement and rotation velocity in two representative points – (i) in the resonance and (ii) at the maximum rotation velocity when the influence of the damping factor for the oscillation amplitude can be ignored. Equations of motion simulations were performed and the results were compared with the actual values of displacement and the velocity vibration platform.

Keywords: *vibration isolation, ventilation, air conditioning, HVAC*

S t r e s z c z e n i e

W artykule przeanalizowano pracę dwóch rzeczywistych układów wibroizolacji. Pierwszy przypadek dotyczył wibroizolatorów gumowych, drugi przypadek amortyzatorów sprężynowych. Przedstawiono metodę wyznaczania parametrów równania różniczkowego drugiego rzędu opisującego ruch drgający. Parametry równania zostały wyznaczone na podstawie pomiaru amplitudy i prędkości obrotowej w dwóch charakterystycznych punktach: (i) w rezonansie oraz (ii) przy maksymalnej prędkości obrotowej, kiedy wpływ współczynnika tłumienia na amplitudy drgań może zostać pominięty. Przeprowadzono symulacje równania ruchu, a wyniki porównano z rzeczywistymi wartościami przemieszczeń i prędkości platformy wibracyjnej.

Słowa kluczowe: *wibroizolacja, wentylacja, klimatyzacja, HVAC*

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1. Introduction

Current HVAC (Heating, Ventilation and Air Conditioning) equipment is usually designed for different operational parameters. This applies to variable airflow devices and also to devices with variable hydraulic profiles. In both cases, it can be get through changeable turnovers of the fun rotor. For units with variable airflows, rotation velocity may vary within the range 40% to 100%. For units with significantly changing hydraulic profiles, rotation velocity may vary from 20% to 100% of the maximum rotation velocity. Ventilation and air conditioning units with variable working parameters are used to serve living and technological facilities. Typical applications of this type of equipment are described below:

a) facilities with a high number of rooms where the room occupation factor can change and there is no need to serve non-occupied rooms (i.e. hotels),
b) technological facilities that require a constant airflow and the air need to be purified at few steps of filtration,
c) facilities that require definite pressure difference between particular rooms but are not constantly occupied (i.e. surgical complexes).

For units with a variable rotational speed, we are facing the problem of suitable device fixing and proper vibration damping of the fan and motor.

Properly designed, linear, supercritical vibration isolation can be a cost effective solution ensuring the correct functioning of driving systems. However, in some cases this can lead to the precipitated wear of machine parts. This arises from the fact that the damping system is not dedicated to work in frequencies similar to self-oscillation frequencies. In extraordinary situations, like ground bumps or hammering engine loads, even with properly designed vibration isolation, the unit can get into resonance. To avoid such situations, it's recommended to apply non-linear vibration isolation systems. Such solutions considerably eliminate sudden increases of oscillation amplitude (forces transferred to the ground) that are mostly related to the appearance of resonance. An overview of non-linear systems is presented in reference [1]. By applying passive vibration damping with nonlinear vibration isolation techniques Yang et al. decreased the frequency of normal mode from 11.7 Hz to 2.5 Hz. He achieved this by fitting compressing springs perpendicularly to the direction of oscillation. Compressing springs were reducing the stiffness of the system [2]. A similar solution is presented in reference [3]. The efficiency of a vibration isolation system, including additional, horizontal springs (with regulated parameters), was checked experimentally. As a result, it was affirmed that the usage of a negative stiffness structure (NSS) improves the features of vibration isolation at low force frequencies. This thesis was also confirmed in reference [4]. Xingtian et al. used two deformed Euler beams instead of springs.

Currently, except for passive vibration isolation systems based on springs or other flexible materials, active systems are intensively developed. Hanieh et al. [5] proposed the use of a piezoelectric device to correct the vibration level on the basis of vibrating mass velocity measurements. A slightly more complicated solution based on two spring-joined masses was show in reference [6]. Based on checks of oscillation velocity, a controlling system was generating a positive or negative force to compensate i.a. the displacement. It was proved that active compensation is favourable, but only in some specific cases.

Despite advanced research in the science of active vibration isolation systems, passive linear systems are the most popular in everyday use, as they are simpler and give sufficient oscillation reduction. The assessment methods of such systems are contemporarily carried out. Michalczyk proposed a new judgement method of maximum oscillation amplitude with few degrees of freedom during temporary resonance [7]. The execution of adequate calculations shouldn't be the final step of the working system assessment. Existing ventilation systems are relatively complicated so the experimental examination of each unit is highly recommended if possible at the site of manufacture (factory acceptance testing FAT), or before start-up of the system at final user facilities [8].

This study describes basic problems related to the carrying dynamic forces of working fans at variable rotational speeds. Displacement and velocity measurements of two existing systems with flexible motor fixing will be presented as well. The first system is equipped with rubber insulators, the second, with spring insulators. This study attempts to create a model of vibration isolation system, based on a commonly used mathematical description. Equation parameters describing oscillating movement were calculated on the basis of displacement and frequency measurements taken at two key working points - in resonance and at four times higher frequency than the resonance frequency. Research confirmed that a two-point check is enough to correctly determine all parameters describing the vibration isolation system.

2. Mathematical description

The flexible motor fitting system is commonly described with a second order differential equation:

$$m\frac{d^2y}{dt^2} + c\frac{dy}{dt} + k_d y = F_0 \sin(\omega t) \tag{1}$$

where:

- F_0 – amplitude of the driving force [N],
- ω – angular frequency [rad/s],
- t – time [s],
- m – vibration isolation loading mass [kg],
- y – displacement [m],
- c – damping coefficient [Ns/m],
- k_d – stiffness coefficient [N/m].

The left side of the equation $F_0\sin(\omega t)$ describes the sinusoidal driving force. This equation can also be presented as follows:

$$\frac{d^2y}{dt^2} + 2\xi\omega_r\frac{dy}{dt} + \omega_r{}^2 y = \frac{F_0}{m}\sin(\omega t) \tag{2}$$

where:

- ξ – dimensionless damping factor,
- ω_r – resonance angular frequency (normal mode) [rad/s].

Joining equation (1) and (2) we get the dependence below:

$$k_d = \omega_r^{\,2} m \tag{3}$$

$$c = 2\xi\omega_r m \tag{4}$$

The above leads to the conclusion that knowing the mass of the system, the frequency of the resonance oscillation and the dimensionless damping factor, we can determine the parameters of the differential equation describing oscillation movement. To make the comparison of the frequency easier, it's recommended to introduce the coefficient μ defined as follows:

$$\mu = \frac{n}{n_r} = \frac{\omega}{\omega_r} \tag{5}$$

where:

n – driving frequency [Hz],
n_r – normal mode frequency (resonant) [Hz].

If $\mu = 1$ it means that vibration protection system operates in resonant area (critical area). If $\mu < 1$ it means that vibration protection system runs in under-critical area and if $\mu > 1$ it runs in over-critical area.

3. Enumeration of differential equation parameters

To calculate the stiffness coefficient, use equation (3), substituting the mass and frequency and the frequency of the normal mode (resonant). To get the value of the damping coefficient (equation 4), it is necessary to know the value of the dimensionless damping coefficient ξ. To know ξ value, it is preferable to use equation (6) and (7) describing dimensionless relative amplitude [9]:

$$\upsilon_0 = \frac{\mu^2}{\sqrt{(1-\mu^2)^2 + (2\xi\mu)^2}} \tag{6}$$

$$\upsilon_0 = \frac{m y_0}{M_w R_m} \tag{7}$$

where:

υ_0 – dimensionless relative amplitude,
y_0 – oscillation amplitude [m],
M_w – rotation mass [kg],
R_m – eccentric radius [m].

In resonant conditions, when $\mu = 1$, equation (6) looks as follows:

$$\upsilon_0 = \frac{1}{2\xi} \tag{8}$$

And after transformation of equation (8) coefficient value ξ can be solve by (9):

$$\xi = \frac{1}{2\upsilon_0} \tag{9}$$

The above equation (9) is correct only for resonant conditions. The dimensionless relative amplitude υ_0 should be calculated from equation (7) after substituting the multiplied rotation mass, eccentric radius and oscillation amplitude (resonant). The oscillation amplitude can be calculated experimentally, and the multiplication of the rotating mass and the eccentric radius can be calculated from joining equations (6) and (7), with the initial assumption that the damping coefficient $\xi = 0$ (equation 10).

$$M_w R_m = m y_1 \frac{\left|1 - \mu^2\right|}{\mu^2} \tag{10}$$

where:

y_1 – oscillation amplitude in the area where the damping factor influence to the conduct of the system can be ignored [m].

Assumption $\xi = 0$ is legitimate, when angular driving frequency significantly differs from angular frequency in normal mode then $\mu \neq 1$ and damping coefficient ξ hasn't got meaningful influence for the dimensionless relative amplitude. This is clearly shown in Figure 1.

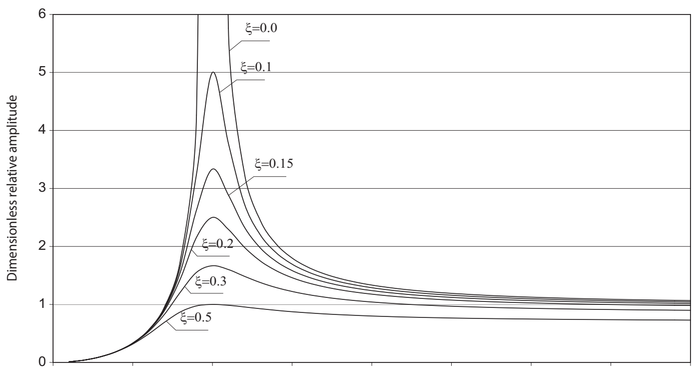

Fig. 1. Relationship between dimensionless amplitude υ_0 and rate ratio μ for different damping coefficients ξ

Example: difference between dimensionless relative amplitudes calculated when μ = 4 with the assumption that ξ = 0,0 and ξ = 0,1 equals 2%. The most common values of damping coefficients are below 0.1, so usage of equation (10) to calculate the multiplication of the rotating mass and the eccentric radius at possibly high μ values is justified.

4. Forces in the vibration isolation system

Using equation (11) it is possible to calculate the amplitude of the oscillation driving force.

$$F_0 = M_w R_m \omega^2 \tag{11}$$

Equation (12) can be useful to calculate the coefficient of force transfer (ratio of the force amplitude transferred to the foundation to the driving force amplitude).

$$TR = \frac{P_0}{F_0} = \sqrt{\frac{1+(2\xi\mu)^2}{\sqrt{(1-\mu^2)^2+(2\xi\mu)^2}}} \tag{12}$$

where:

TR – force transfer coefficient,

P_0 – amplitude of the force transferred to the foundation [N].

Knowing the dimensionless damping coefficient ξ and μ, it's easy to calculate the coefficient of force transfer for the particular driving frequency. That can be used to further calculate the amplitude of the force transferred to the foundation. It's important to notice that as the dimensionless damping coefficient gets lower, the force transferred to the foundation gets higher (in resonant area). This phenomenon is presented in diagrams 2 and 3. These diagrams present 4 variants of dynamic force amplitude that are transferred to the foundation at different dimensionless damping coefficients. ξ, different driving frequencies (n) and different frequencies of normal mode (n_r). The amplitude of the driving force was calculated on the basis of a rotating mass equal to 15 kg and the eccentric radius of 0.2 mm. Additionally, the dashed line presents the amplitude of the driving force for the doubled eccentric radius. That shows the rotor unbalance influence for the appearance of the amplitude of dynamic forces. Figures 2 and 3 were developed for driving frequencies of 0-50 Hz and for different maximum μ values. Figure 2 was developed for μ = 3, and figure 3 was developed for μ = 5. The consequences of these assumptions are different resonant frequencies of the units settled on vibration isolators. In the first case, n_r = 16.67 Hz, and in the second, n_r = 10 Hz.

Presented diagrams show that:

(1) amplitudes of the dynamic forces caused by rotary machines are not dependent on amortization,

(2) applied amortization significantly influence the amplitudes of the dynamic forces transferred by isolators to the ground,

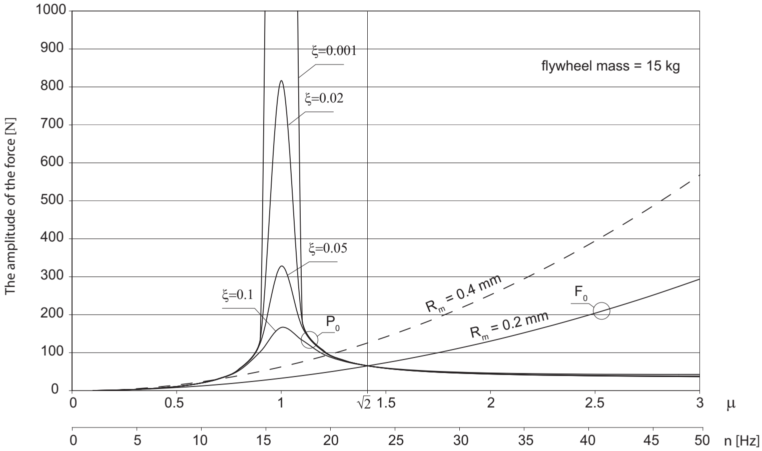

Fig. 2. Amplitudes of dynamic forces in the vibration isolation system at maximum μ = 3 for different damping coefficients ξ and resonant velocity n_r = 17 Hz

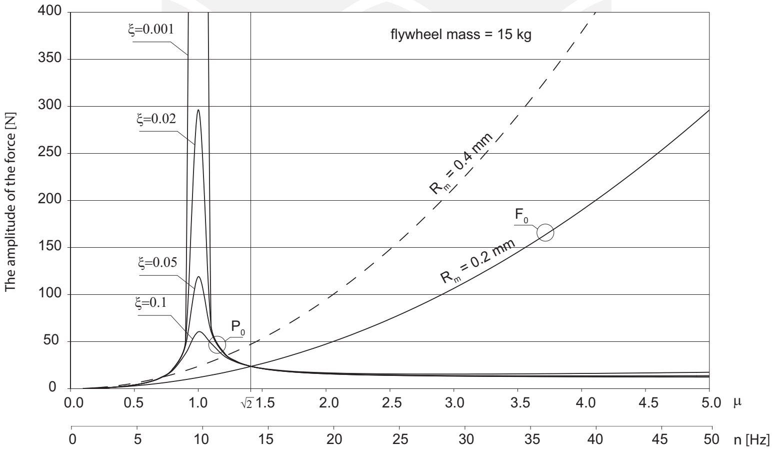

Fig. 3. Amplitude of dynamic forces in vibration isolation system at maximum μ = 5 and different damping coefficients ξ and resonant velocity n_r = 10 Hz

(3) during the machine motion in the resonant area, the amplitudes of dynamic forces can be up to several dozen higher than amplitudes of dynamic forces caused by other machine motion,

(4) along with the increase of the dimensionless damping coefficient ξ, amplitudes of dynamic forces transferred to the ground via isolators are reduced,

(5) dynamic features of the isolators impact the amplitudes of dynamic forces transferred to the ground. Usage of isolators that ensure lower frequencies of normal mode (higher values of maximum μ) effects in decrease of amplitude of dynamic forces. (6) settling the fan on elastic elements and ensuring the value of μ ≥ 5 will result in amplitudes of dynamic forces transferred to the ground lower than the amplitude of the force appearing during periods of maximum rotation speed of the moving fan.

5. Measurements of existing object

Parameter measurements were taken for the vibration platform with a mass of 242.4 kg. The platform was located on 4 equally loaded isolators. The examination was carried out in two parts – firstly, with the rubber isolators, secondly, with the spring isolators. Measurements of the vertical displacement of the platform were taken during the experiment. Displacement values were taken in the function of motor rotation speed.

Displacement values were taken in 2 independent series. The first series concerned the frequency range below 10 Hz, and second series, above 10 Hz. Such a division of the frequency range was due to the specification of measuring devices. The employed vibration meter (Lutron VB-8200) operates for frequencies up 10 Hz. Displacements for lower frequencies were measured with a mechanical tactile vibration recorder. The frequency was measured with a stroboscope.

Vibration meter VB-8200 can measure the acceleration (peak) and the velocity (peak), therefore, displacements were obtained indirectly. In order to maintain the reliability of the measurements, frequencies were also obtained in an indirect manner. In this work we used only the results when frequencies measured by the stroboscope were equal to frequencies measured by vibration meter.

Examinated unit had an extra weight added to the shaft to emphasize the unbalance of the unit. Displacement values of the platform settled on the rubber and spring isolators are shown in picture 4.

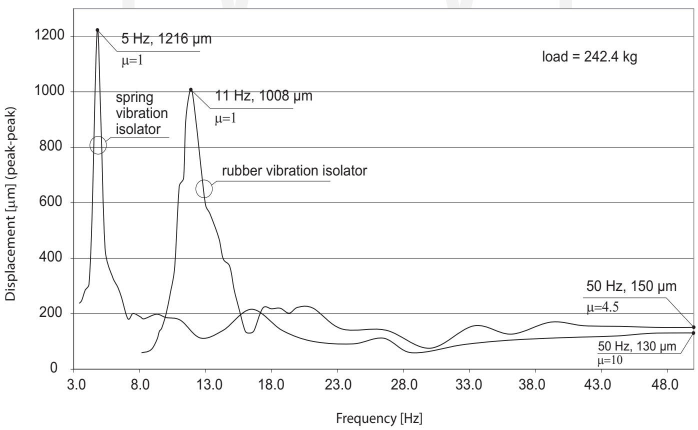

Fig. 4. Displacement (peak-peak) of vibration platform based on a rubber and spring in a function of driving frequency

6. Calculation of the differential equation parameters

The elasticity coefficient in dynamic conditions was calculated with equation (3), after the substitution of normal mode frequency and mass.

The procedure of damping factor c calculation was as follows:

a) the calculation of the multiplied rotating mass and eccentric radius from equation (10), after the substitution of system mass, oscillation amplitude (for highest possible μ) and μ itself.
b) calculation of dimensionless relative amplitude for resonant conditions using equation (7) after substituting system mass, multiplied $M_w R_M$ and oscillation amplitude Turing resonance,
c) calculation of the damping coefficient ξ on a basis of equation (9) after substitution of dimensionless relative amplitude in resonance,
d) finally, the calculation of the damping coefficient from equation (4). All acquired data is presented in Table 1.

Table 1

Parameters calculated for vibration isolation system

	n_r	$2y_0$	n_1	$2y_1$	k_d	$M_w R_M$	ξ	c
	Hz	μm	Hz	μm	$10^6 \frac{\text{N}}{\text{m}}$	kg · m		$10^3 \frac{\text{Ns}}{\text{m}}$
Rubber	11	1008	50	150	1.3	0.017	0.07	2.5
Spring	5	1216	50	131	0.2	0.016	0.05	0.8

7. Computer model of the vibration isolation

The calculated parameters of *the* second order differential equation were employed for the development of a computer model of vibration isolation. The transmittance of the vibration platform was calculated using equation (13) and the transmittance of the driving force coming from the motor using equation (14).

$$G(s) = \frac{\frac{1}{m}}{s^2 + \frac{c}{m}s + \frac{k_d}{m}} \tag{13}$$

$$U(s) = \frac{F_0\omega}{s^2 + \omega^2} \tag{14}$$

Equivalent transfer functions of the motor-vibration platform were calculated afterwards (15) for driving frequencies from 3 to 50 Hz (one transfer function for each one frequency).

$$G_{zn}(s) = G(s)U(s) \qquad n = 3...50 \text{ Hz} \tag{15}$$

Each transfer function from $G_{z3}(s)$ to $G_{z50}(s)$ was explored in simulation in MATLAB. The transfer functions were subjected to Dirac delta $\delta(t)$ using MATLAB command *impulse*. In effect, impulse responses of the objects $G_{zn}(s)$ were explored. A direct result of the simulation was the value of the momentary displacement for the particular driving frequency. Figure 5 presents the peak values of the displacements coming from the simulation with real measured values taken from the existing object.

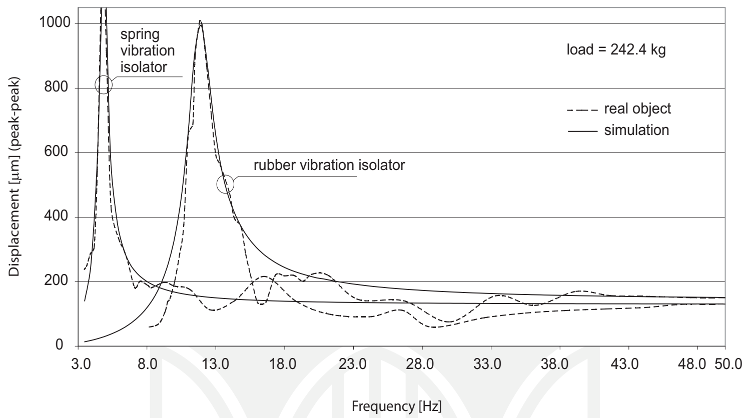

Fig. 5. Real and simulated displacement (peak-peak) of vibration platform depending on driving frequency and the type of applied isolators

By calculating the first derivative from the momentary displacement, momentary oscillation velocity was obtained. The maximum velocity values coming from the simulation and real measures are presented in Figure 6. This diagram additionally shows the maximum allowed velocity level, according to the ISO 2372 (10816) [10].

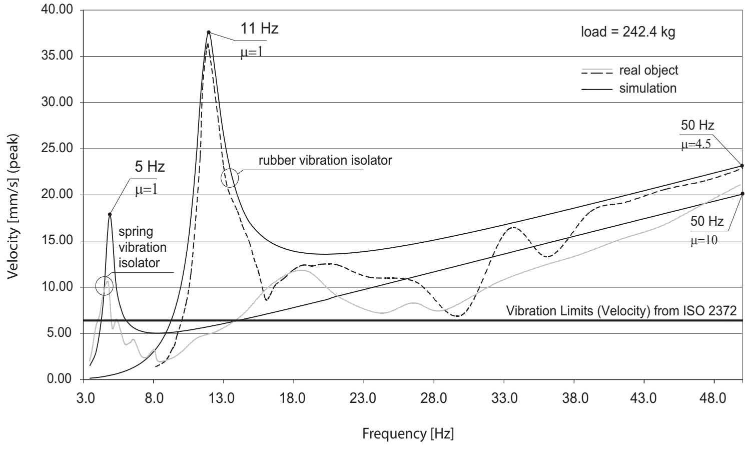

Fig. 6. Real and simulated velocity (peak) of the vibration platform as a function of driving frequency and the type of applied isolators

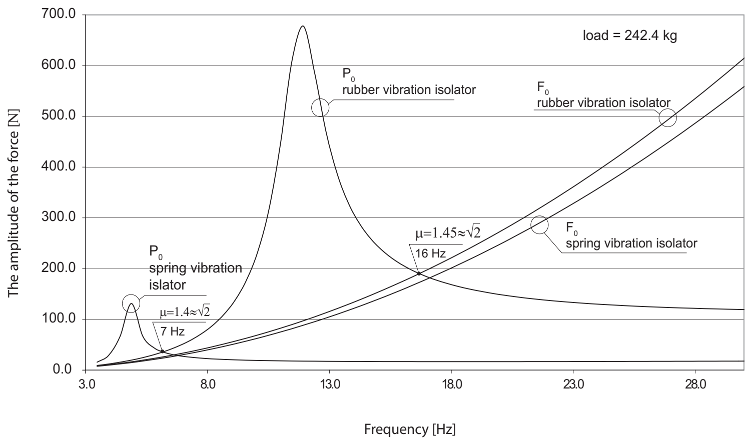

Fig. 7. Amplitudes of dynamic forces appearing in considered systems as a function of driving frequencies

Based on the simulation results, we developed the graphs showing the amplitudes of the dynamic forces in the examined system. The results are shown in Figure 7.

8. Model results analysis

Displacement graphs confirm that the isolators with lower dimensionless damping coefficient (for spring isolator) are more efficient in displacement reduction but this applies only to the situations when the vibration isolation works in overcritical area

In the resonant area, oscillation amplitude exceeds oscillation amplitude of the vibrating platforms placed on the rubber elements. Additionally, increase of the oscillation amplitude for the spring isolators takes place at significantly lower driving force. According to this, reduction of the damping coefficient may lead to excessive vibrations in the resonant area, for instance, in situations with a long accelerating drive when the unit gets through resonance area to achieve nominal rotation speed.

At $\mu > \sqrt{2}$, the force transferred to the foundation is lower than the driving force, this confirms the accuracy of the simulation results.

In Figure 7, two lines appear illustrating the amplitude of driving force $F_{0.}$ This is a result that the product of $M_w R_M$ calculated according to equation (10) for the spring isolators configuration is different to the rubber isolators configuration and the results are as follows:

$(M_w R_M)^{sprongs} = 0.016$ kg/m,

$(M_w R_M)^{rubbers} = 0.017$ kg/m.

The motor shaft located at the platform had an extraordinary imbalance weight to increase the dynamic forces during the experiments. This had an negative impact for the working conditions. The examined unit in wide scope of work was exceeding allowed vibration amplitudes according to the ISO 2372 According to this norm, the maximum allowed momentary velocity for machines below 15 kW equals 6.36 mm/s.

Experiments related to vibration isolation become an effective tool in the rotary machine assessment process. Comprehensive mathematical tooling and it's availability results in simplified creation of the mathematical models for real vibrating units. The obtained results can be used for the current evaluation of operating machinery, this can lead to potential failure detection (Fault Diagnostic and Detection FDD). Simulation can be also useful in estimating the effectiveness of the applied vibration isolation system in different conditions, i.e. at different rotation velocity or different weight mass.

The developed tests of the isolators prove the necessity of checks for newly implemented isolators as their real parameters may be different to their design and manufacturing features. The reasons for the above may be different – i.e. additional friction or rubber features that weren't taken into account during the design phase.

9. General tips for amortization systems selection

In HVAC systems, we always consider two-part rotating machinery. The work of the pump, the fan or the compressor is always related to the work of the powering motor. We can feature three main types of couplings between the motor and the rotary machine:

a) direct system, where the rotation velocity of the fan and motor are the same,

b) clutching systems, where the rotation velocities on the motor and fan are usually similar,

c) belt driven systems, where the rotation velocities of the motor and fan can be different as a function of driving wheels.

For the systems where the rotation velocity is the same on both elements, driving velocity is obvious. The value of rotation velocity is extremely important for the selection of amortization systems with belt driven units. Generally, in HVAC systems, isolators should be selected to let the unit work in conditions distant from the resonance at the frequency ratio $\mu \geq 3$.

For the calculation of the frequency ratio μ, it is recommended to apply the rule of selecting the minimum rotation speed $n^{\min}(n_w, n_s)$. When for lower rotation velocity, the vibration isolation condition is met, it is also met for higher rotation velocities. Following this, we can say that for higher rotation velocities, the value of the ground transferred reduced force will be much smaller than for lower rotation velocities. At the same time, let's not forget that correctly selected isolators should have small enough ξ so that the limitation of the force should be as high as possible. The value of the ξ should also be big enough to let the machine go through the resonance phase without damaging the foundation as a result of too high amplitude of the force transferred to the foundation. Considering the design of reinforced supporting frames, the basis for the calculations should be maximum driving velocities.

10. Vibration isolation of the fans with variable airflows

Considering for calculations the frequencies ratio $\mu \geq 3$ referred to minimum driving velocity leads to a major reduction of dynamic forces transferred to the ground by isolators.

The achievement of low normal mode frequencies (i.e. 2 Hz) requires special isolator constructions and specific fitting systems to the ground and mounting frame. Considering variable rotation speed fans working in the range of 50–100% of maximum speed, the assumption of frequency ratio $\mu = 5$ allows us to ensure the satisfactory working parameters of the motor/fan at minimum speeds. When fans are designed to work in the range 20–100% of maximum speed, the assumption of frequency ratio of $\mu = 5$ for maximum speed will result in resonance of the fan/motor. In those cases, systems located on the isolators should be calculated with $\mu = 8$ for maximum speeds. At a minimum speed around of 20% of the maximum speed, values of the dynamic forces transferred to the ground would be lower than the amplitudes of created forces.

11. Conclusions

During the design phase of vibration isolation of variable speed fans, it is essential to consider the minimum and maximum rotation speeds. It is recommended to keep the vibration isolation in the overcritical area along with the full range of the designed rotation speeds and keep the minimum value of μ at least at a level of 1.6. Dimensionless damping coefficients should be as low as possible. In cases when the maximum rotation speeds of the motor/fan are not exceeding 20 Hz and the minimum are not exceeding 4 Hz, achievement of overcritical isolation can be difficult. To remedy this, it is worth thinking about the possibility of resonant work at minimum rotation speeds and applying isolators with dimensionless damping coefficients $\xi > 0.05 \div 0.1$. Amplitudes of dynamic forces transferred to the ground via isolators during resonant work will be much lower than the dynamic forces created at maximum speeds. To reduce the amplitudes of fan/motor oscillation, it’s advised to use **lean limiters**, similar to the ones used in marine vibration isolation. In this case, implementation of under critical vibration isolation will cause that amplitudes of dynamic forces transferred to the ground will be higher than the amplitudes created as a result of working. It is true for the full range of fun work.

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