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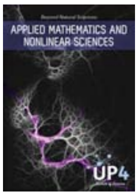

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Vibrations attenuation of a system excited by unbalance and the ground movement by an impact element

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Abstract

This paper concentrates on the vibrations attenuation of a rotor driven by a DC motor and its frame flexibly coupled with a baseplate by linear cylindrical helical springs and damped by an element that can work either in inertia or impact regime. The system oscillation is governed by three mutually coupled second-order ordinary differential equations. The nonlinear behaviour occurs if the impact regime is adjusted. The damping element operating in inertia mode reduces efficiently the oscillations amplitude only in a narrow frequency interval. In contrast, the damping device working in impact regime attenuates vibrations of the rotor frame in a wider range of the excitation frequencies and it can be easily extended if the clearances between the rotor casing and the damping element are controlled. The development of a computational procedure for investigation of vibration of a flexibly supported rotor and for its attenuation by the inertia and impact dampers; learning more on efficiency of the individual damping regimes; finding possibilities of extension of the frequency intervals of applicability of the damping device; and obtaining more information on the character of the vibration induced by impacts are the main contributions of this research work.

Keywords: Rotating system; Damping effect; Inertia damper; Impacts; Vibration attenuation
AMS 2010 codes: 34H20; 34H10; 37N30.

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1 Introduction

Collisions of solids are physical phenomena of great importance. They occur during a large number of natural and technological processes. Their significant characteristic is short duration of the body collisions, very large impact forces and almost sudden changes of the system's kinematic parameters. Both the observations and theoretical analyses show that behavior of the systems with impacts is highly nonlinear, considerable sensitive to initial conditions and loading effects, arriving often at irregular oscillations and hardly predictable motions. Behavior of each system where the body collisions take place is different, and therefore, each of them must be investigated in an individual way.

Because of the practical importance, a good deal of attention is focused on analysis of vibro-impact systems, where the vibrations are governed by the momentum transfer and mechanical energy dissipation through the body collisions. This is utilized for impact dampers applied to attenuate high-amplitude oscillations, such as those appearing in subharmonic, self-excited and chaotic vibrations.

Problem of the body collisions is old, but the development of advanced computational facilities and software tools at the end of the 20th century brought new possibilities for its investigation. The topic is massively studied by many authors focusing on the impact systems (see e.g. [1–7]), the pendulums [8], the gear transmissions systems [9], the systems with clearances and non smooth stiffness ([10, 11]) and electromechanical systems ([12]).

In this paper, a system formed by a rotor and its casing flexibly coupled with a baseplate and of an impact body, which is separated from the casing by two, lower and upper, gaps is analyzed.

The work presented in this article was motivated by the research started in [13] and later developed in [14] and [15]. Vibration reduction of an electromechanical system by an impact damper having rigid stops was investigated by [13] and with soft stops by [14]. As a natural continuation of the research the system with nonlinear plate springs and the soft stops was a subject of the study in [15]. The intention of the further research was to analyze a larger extent of design solutions utilizing application of impact damping devices on the vibration attenuation of flexibly supported machine sets. On contrary to [13–15], in this paper a rotating system with linear cylindrical helical springs is investigated. Moreover, the damping in the connection between the impact element and the damper housing is newly taken into account.

A new contribution of the presented work consists of investigating the system oscillations as a result of a combined time variable loading caused by two sources, the rotor unbalance and the baseplate vibrations, and in investigating the interaction between the motor. Emphasis is put on observing the influence of the inner impacts on the character and reduction of the system vibration dependent on the width of the upper and lower clearances between the rotor frame and the impact body. The investigated system is of great practical importance as it represents a simplified model of a rotating machine, which is excited by a ground vibration and unbalance of rotating parts and damped by an impact damper. Results of the performed simulations contribute to better understanding of the dynamic behavior of such technological devices and of impact systems with complicated loading, in general.

2 The investigated system

The studied system is formed by a rotor (body 1, Fig. 1), its casing (body 2, Fig. 1) and a baseplate (body 3, Fig. 1). The coupling of the rotor casing and the baseplate is accomplished by a spring and damping element (body 4, Fig. 1). The casing and the baseplate can move in vertical direction and the rotor can rotate and move together with its casing. The system is excited by the ground vibration and unbalance of the rotor. For attenuation of the casing oscillations an impact damper is proposed. It consists of a housing fixed to the rotor casing (body 2, Fig. 1) and of an impact element (body 4, Fig. 1) coupled with the housing by a linear spring. The impact body can move only in a vertical direction and is separated from the housing by the lower and upper clearances that limit its vibration amplitude. The rotor is loaded by an external moment produced by a DC motor given by

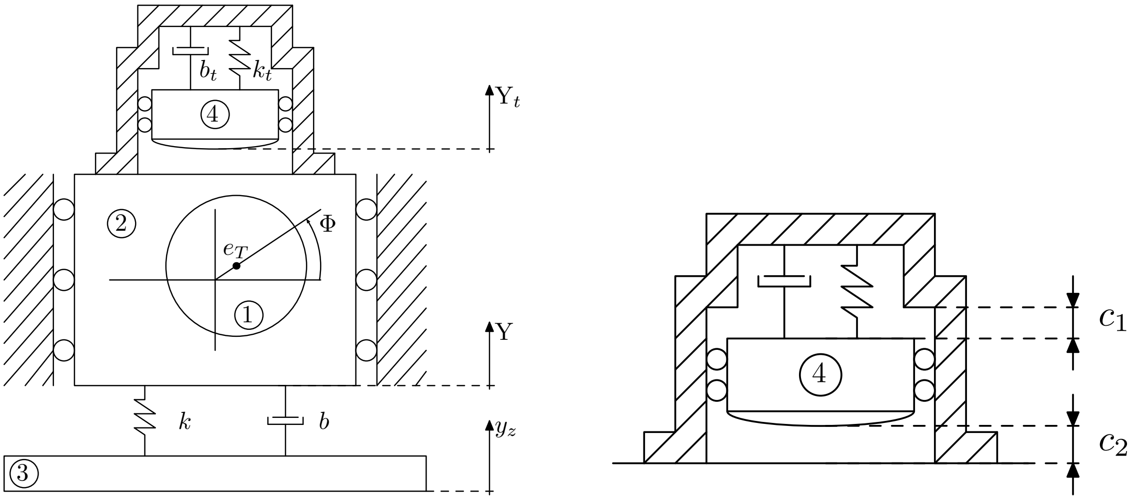

Fig. 1 Model of vibrating system

the moment characteristic.

The task was to analyze the influence of the upper and lower clearances and the mass of the impact body respectively on attenuation of the rotor frame oscillation and character of its motion.

In the computational model all bodies are considered as absolutely rigid except the contact areas between the impact element and the rotor frame. The cylindrical helical springs coupling the rotor casing and the baseplate have linear characteristic.

The damper between the rotor frame and the baseplate and the spring coupling the impact body with the damper housing are linear. The Hertz theory has been accepted to describe the impacts. The nonlinear contact stiffness and damping were linearized in the expected range of the contact deformation.

The investigated system has three mechanical degrees of freedom. Its instantaneous position is defined by three generalized coordinates:

- Y - vertical displacement of the rotor casing,
- Y_t - vertical displacement of the impact body and
- Φ - angular rotation of the rotor.

The system vibration is governed by the equations of motion that were derived by application of the Lagrange equations of the second kind (see [13] or [14]) or by the decoupling method (see [15]):

$$\begin{aligned}(m+m_R)\ddot{Y}+m_R e_T\cos(\Phi)\ddot{\Phi} &= m_R e_T\dot{\Phi}^2\sin(\Phi)-\\ &-b(\dot{Y}-\dot{y}_z)+b_t(\dot{Y}_t-\dot{Y})-\\ &-k_1(Y-y_z)-k_t(Y-Y_t)-\\ &-(m+m_R)g-F_{I1}-F_{I2},\\ m_t\ddot{Y}_t &= F_{I1}+F_{I2}-k_t(Y_t-Y)-\\ &-b_t(\dot{Y}_t-\dot{Y})-m_t g,\\ (J_{RT}+m_R e_T^2)\ddot{\Phi}+m_R e_T\cos(\Phi)\ddot{Y} &= -m_R g e_T\cos(\Phi)+M_Z-k_M\dot{\Phi}\end{aligned} \tag{1}$$

where ($\dot{}$) and ($\ddot{}$) denote the first and second derivative with respect to time, respectively.

The impact forces are defined as follows:

$$F_{I1}=\begin{cases}-k_c(Y_t-Y-c_1)-b_c(\dot{Y}_t-\dot{Y}) & \text{if condition (3) is satisfied,}\\ 0 & \text{if condition (3) is not satisfied}\end{cases} \tag{2}$$

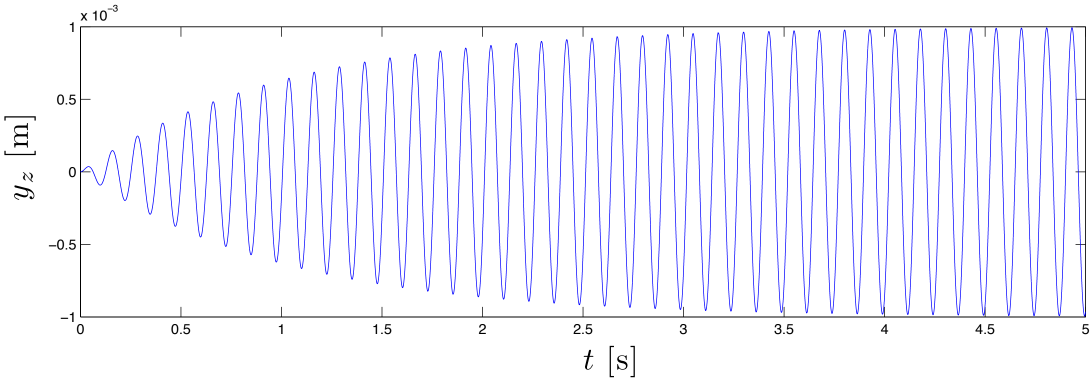

Fig. 2 Vibration of the baseplate given by function (6).

where condition (3) is specified by

$$\mathrm{Y}_t - \mathrm{Y} - c_1 > 0 \text{ and } -k_c(\mathrm{Y}_t - \mathrm{Y} - c_1) - b_c(\dot{\mathrm{Y}}_t - \dot{\mathrm{Y}}) < 0 \tag{3}$$

and

$$F_{I2} = \begin{cases} -k_c(\mathrm{Y}_t - \mathrm{Y} + c_2) - b_c(\dot{\mathrm{Y}}_t - \dot{\mathrm{Y}}) & \text{if condition (5) is satisfied,} \\ 0 & \text{if condition (5) is not satisfied} \end{cases} \tag{4}$$

where condition (5) is specified by

$$\mathrm{Y}_t - \mathrm{Y} + c_2 < 0 \text{ and } -k_c(\mathrm{Y}_t - \mathrm{Y} + c_2) - b_c(\dot{\mathrm{Y}}_t - \dot{\mathrm{Y}}) > 0 \tag{5}$$

Here k_c, b_c denote the contact stiffness and damping and c_1, c_2 stand for the upper and lower clearances. Conditions (3) and (5) express that the contact forces act only if the impact body and the rotor frame are in contact and that they can be only compressive.

The movement of the baseplate is described by

$$y_z(t) = A\,(1 - \mathrm{e}^{-\alpha t})\sin(\omega t) \tag{6}$$

as a rheonomic constraint, where A is the ground vibration amplitude, α is the constant determining how fast the vibration of the baseplate becomes a steady state and ω stands for the excitation frequency, see Fig. 2.

In the beginning no moment is applied on the rotor of the electric motor, no contacts between the impact body and the frame occur, the system is in rest, including the baseplate, and takes the equilibrium position.

The Runge-Kutta method of the forth order with a variable time step was used to solve the equations of motion (1). It is a step by step method. After accomplishing each integration step it is checked if the contact occurs or not (by the difference of displacements of bodies 4 and 2 given by conditions (3) and (5)). If it occurs the contact force used in the next integration step is determined by relations (2) and (4), if it does not magnitude of the contact force in the next integration step remains zero. The Matlab program was utilized for accomplishing the computations.

3 Behavior of the impact element

It was shown by [13] that there is a resonance peak for the baseplate excitation frequency of $\omega = 102$ rad s^{-1} for the system parameters summarized in Tab. 1. Next, parameters c_1 and c_2 are assumed to be equal for simplicity and together with the mass of the impact element m_t and angular frequency ω were taken as variables.

Table 1 Parameters of the system (1).

value	quantity	format	description
m	100	kg	mass of the damping body
m_R	40	kg	mass of the rotor
m_t		kg	mass of the impact element
k_1	1.5×10^5	N m^{-1}	linear stiffness coefficient
J_{RT}	5	kg m^2	moment of inertia of the rotor
b	1.5×10^3	N s m^{-1}	damping coefficient of the suspension
k_t	8×10^4	N m^{-1}	coupling stiffness of the impact element
b_t	500	N s m^{-1}	damping coefficient of the impact element
e_T	2	mm	eccentricity of the rotor center of gravity
Φ		rad	rotation angle of the rotor
M_Z	100	N m	starting moment
k_M	8	N m s rad^{-1}	negative of the motor characteristic slope
α	1	s^{-1}	parameter of the baseplate excitation
ω		rad s^{-1}	baseplate excitation frequency
A	1	mm	ground vibration amplitude
k_c	4×10^7	N m^{-1}	contact stiffness
b_c	3×10^3	N s m^{-1}	coefficient of contact damping

In the following, various situations are discussed in detail, dependent on clearances between the rotor casing and the impact body, the mass of the impact element and the baseplate excitation frequency.

For simplicity in the beginning of the study it is assumed that there is no viscous damping ($b_t = 0$ N s m^{-1}) in the coupling between the impact element and the rotor frame (see Fig. 3).

The dependence of the peak-to-peak amplitude of the rotor body vibration (the difference between the maximum and minimum value of Y) on the clearance $c_1 = c_2$ width and the mass of the impact element m_t is shown in Fig. 4 and Fig. 5 for two values of the excitation frequency ω. In the next Fig. 6 occurrence of impacts are drawn for these situations. Combining results of these simulations together it is observed that in the case where the excitation frequency equals $\omega = 100$ rad s^{-1} the damping element shows the maximum vibration attenuation for the mass $m_t = 8$ kg and the clearance at least $c_1 = c_2 = 18$ mm (see Fig. 4(left), Fig. 5(left) and Fig. 6(left)), here $\max_Y - \min_Y = 0.01$ mm. In the second case (Fig. 4(right), Fig. 5(right) and Fig. 6(right); the excitation frequency equals $\omega = 115$ rad s^{-1}) the similar behaviour is observed for the mass $m_t = 6$ kg and the clearance at least $c_1 = c_2 = 19$ mm, here $\max_Y - \min_Y = 0.29$ mm. In both these cases there are no impacts and therefore, the impact element works as an inertia damper. On the other hand, significant attenuation of the rotor body vibrations can be also reached for parameters when impacts occurred. In this regime where the excitation frequency equals $\omega = 100$ rad s $^{-1}$ the maximum vibrations attenuation is reached for parameters $c_1 = c_2 = 6$ mm and $m_t = 25$ kg, here $\max_Y - \min_Y = 0.89$ mm. The attenuation is significant since the peak-to-peak amplitude of the baseplate oscillations is 2 mm. In the second simulation performed for the excitation frequency $\omega = 115$ rad s^{-1} the maximum attenuation of the vibrations is attained for parameters $c_1 = c_2 = 4$ mm and $m_t = 25$ kg, here $\max_Y - \min_Y = 0.81$ mm.

As the system with no impacts is linear, it is possible to find such a mass of the impact element working as an inertia damper which would enable to attenuate vibrations of the rotor frame in the maximum way. Unfortunately, such a damper can be effective only in a small range of excitation frequencies. This is confirmed by Fig. 7 which shows the peak-to-peak vibrations amplitude of the rotor frame dependent on the excitation frequency for the mass of the impact element m_t from 7 kg to 9 kg. There are no impacts and the oscillations attenuation is significant.

To extend the interval of efficient damping, the mass of the damping element would have to be changed but

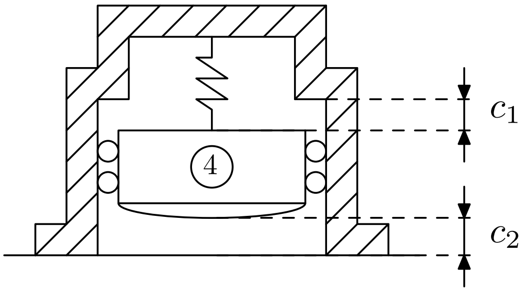

Fig. 3 Model of vibrating system, damping device without damper.

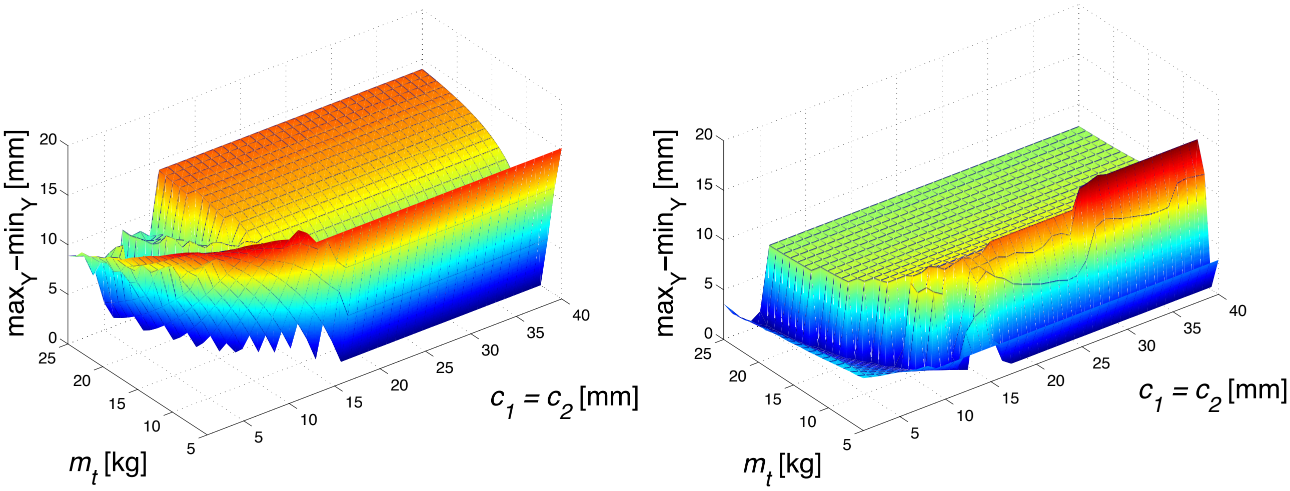

Fig. 4 Peak-to-peak vibration of the rotor frame dependent on the clearance $c_1 = c_2$ and the mass of the impact element m_t for the excitation frequency (left) $\omega = 100$ rad s^{-1} and (right) $\omega = 115$ rad s^{-1}.

such manipulation is not easy to accomplish from the technological point of view.

Remark 1[on computational demands] Simulations shown in Fig. 4 (left and right) were performed on SP TERI (BladeSystem c7000 Enclosure, HP ProLiant BL260c G5, Infiniband: 4X DDR IB Mezzanine HCA for HP BladeSystem c-Class, CPU Quad-Core Intel Xeon 2.5 GHz, Memory 18.5 GB), located at the Technical University of Ostrava, Czech Republic. Each of them were computed as 20×40 separate simulations on 28 cores parallelly, it took 120 core hours, 4.2 hours of real time and 0.6 TB of memory.

4 Efficiency of the impact device

On the other hand it is easy to find a technological solution of the damping device that would make it possible to actively change the clearance width in dependence on the excitation frequency to get maximum attenuation of the rotor vibrations in a wide frequency interval. This is important especially in the cases when the excitation frequency is not constant but when it can slightly vary. This is evident from Fig. 8 showing the dependencies of the peak-to-peak vibration amplitudes of the rotor frame on the excitation frequencies for clearances $c_1 = c_2$ from 2 mm to 4 mm. Fig. 8(left) corresponds to $m_t = 22$ kg and Fig. 8(right) to $m_t = 25$ kg. In all these cases the impacts occur and the vibration attenuation is worthy of attention.

Figs. 9 and 10 show the efficiency of the intentional changing of the clearance width in dependence on the excitation frequency with the aim to get the maximum attenuation. Fig. 9 shows results for the case of no additional viscous damping between the impact element and the frame ($b_t = 0$ N s m^{-1}) while Fig. 10 depicts results for a viscous damping coefficient $b_t = 500$ N s m^{-1}. In both figures there is compared the efficiency of

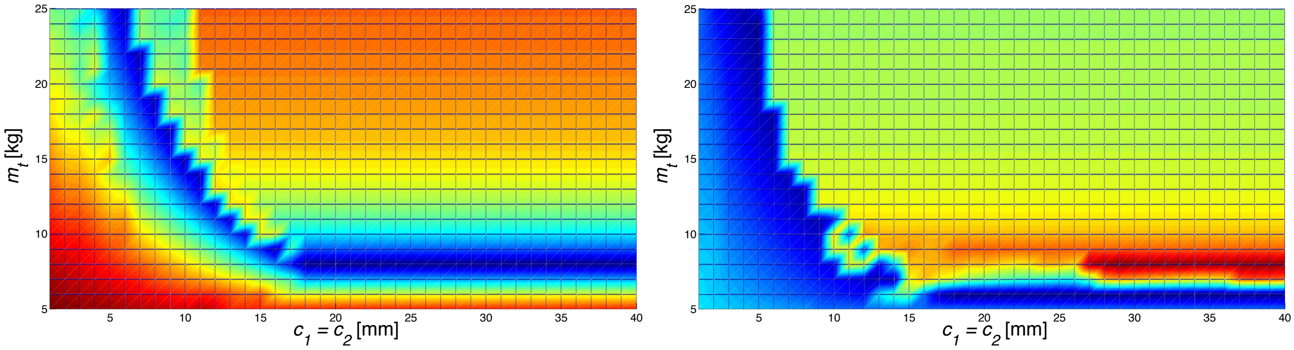

Fig. 5 Contour plot of the peak-to-peak vibration of the rotor frame dependent on the clearance $c_1 = c_2$ and the mass of the impact element m_t for the excitation frequency (left) $\omega = 100$ rad s^{-1} and (right) $\omega = 115$ rad s^{-1}.

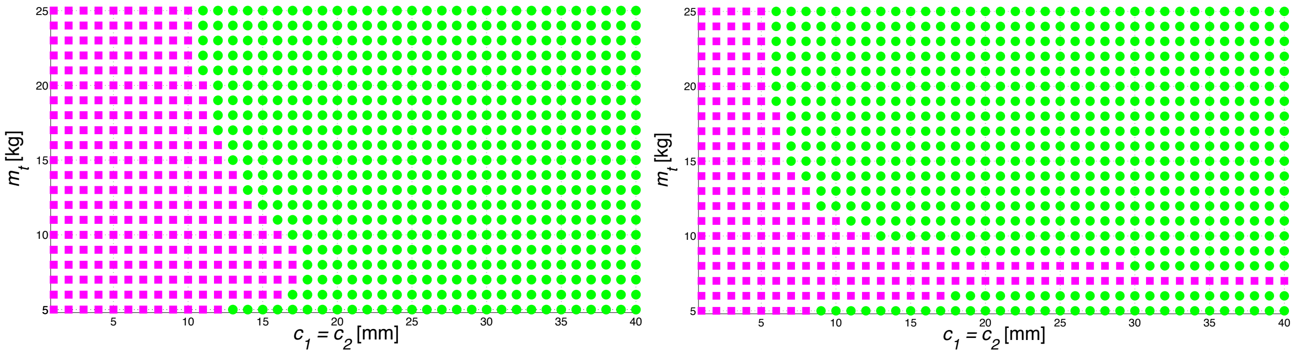

Fig. 6 Impact occurrence dependent on the clearance $c_1 = c_2$ and the mass of the impact element m_t for the excitation frequency (left) $\omega = 100$ rad s^{-1} and (right) $\omega = 115$ rad s^{-1}. Here, *red box* = impact occurs, *green dot* = no impacts.

the impact element (of the impact damping device) with the regime when the impact body works as an inertia damper with no impact occurrence. It is evident that

- the impact regime considerably extends the frequency interval in which the peak-to-peak vibration amplitude of the rotor frame is lower than those of the excitation (2 mm),
- the viscous damping element reduces efficiency of the damping device working both in the inertia and impact regimes and
- the system sensitivity to the baseplate excitation goes down for rising frequencies so that for frequencies higher than approximately 146 rad s^{-1} no impacts are needed to reduce the rotor frame vibration amplitude below the excitation one. Figs. 11 and 12 show the optimum setting of the clearance to get the maximum vibrations attenuation of the rotor frame in the individual ranges of the excitation frequencies.

5 Dynamics of the impact damper

The character of the vibrations of the system with the damping element working in the impact regime in dependence on the baseplate excitation frequency is evident from the bifurcation diagrams drawn in Figs. 13 ($m_t = 25$ kg, $b_t = 500$ N s m^{-1}) and 14 ($m_t = 25$ kg, $b_t = 5$ N s m^{-1}). The individual relative displacements $\mathrm{Y} - \mathrm{Y}_t$ are referred to the magnitude of the clearance between the impact body and the rotor casing for which vibration amplitude of the latter is lowest. It is evident that in the case with additional viscous damping between the impact body and the rotor frame the resulting steady state motion is 1T periodic (T denotes the period of

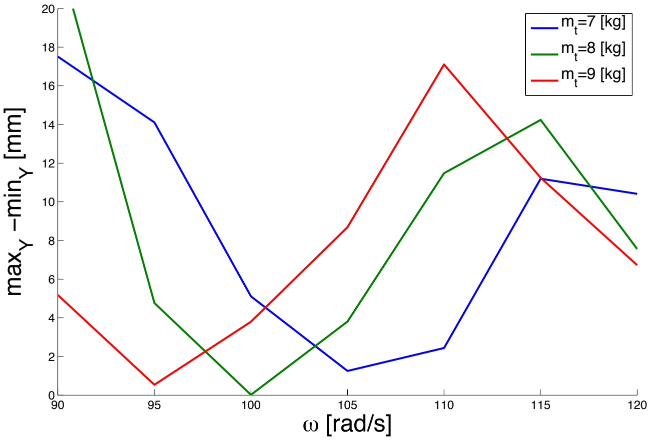

Fig. 7 Peak-to-peak vibrations amplitudes of the rotor frame dependent on the excitation frequency for the mass of the impact element m_t from 7 kg to 9 kg. There are no impacts in all these cases.

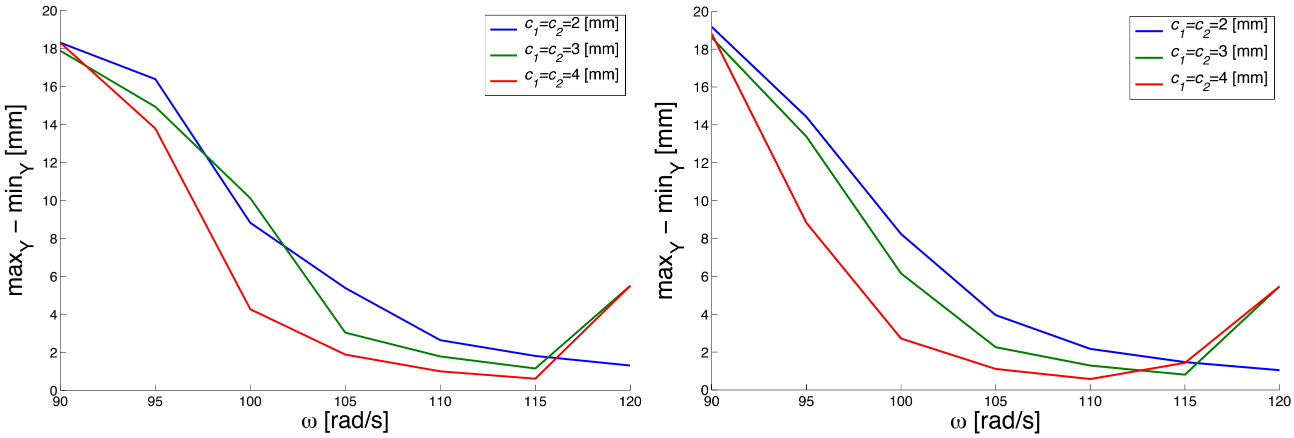

Fig. 8 Peak-to-peak vibrations amplitudes of the rotor frame dependent on the excitation frequency for clearances $c_1 = c_2$ from 2 mm to 4 mm where (left) corresponds to $m_t = 22$ kg and (right) corresponds to $m_t = 25$ kg. In all these cases impacts occur.

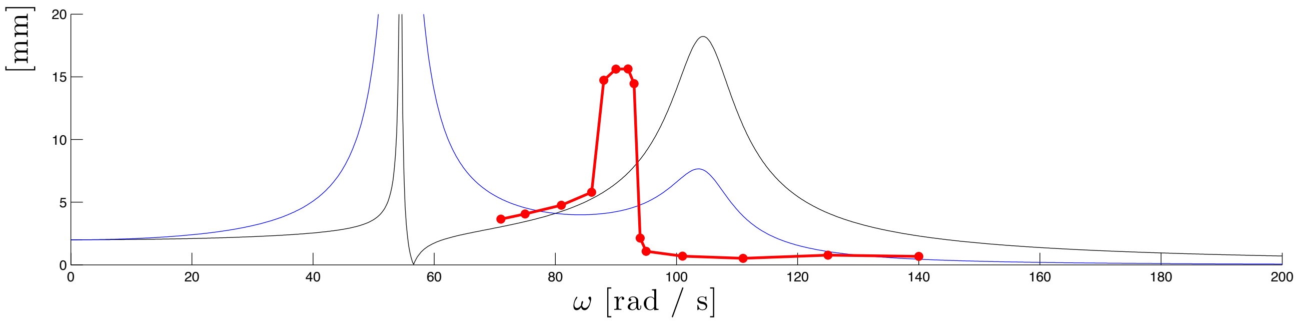

Fig. 9 *Black and blue:* the dependence of the peak-to-peak vibration amplitude of the rotor frame and impact element, respectively, on angular frequency of excitation of the baseplate for unlimited movement of the impact body; *red:* the dependence of the peak-to-peak vibration amplitude of the rotor frame on angular frequency of excitation of the baseplate for the movement of the impact body limited by clearances; for parameters $m_t = 25$ kg and $b_t = 0$ N s m^{-1}.

excitation) in the whole extent of the investigated excitation frequencies. For lower excitation frequencies the motion has a quasi periodic character as no impacts occur and the transient vibration is not reduced because of

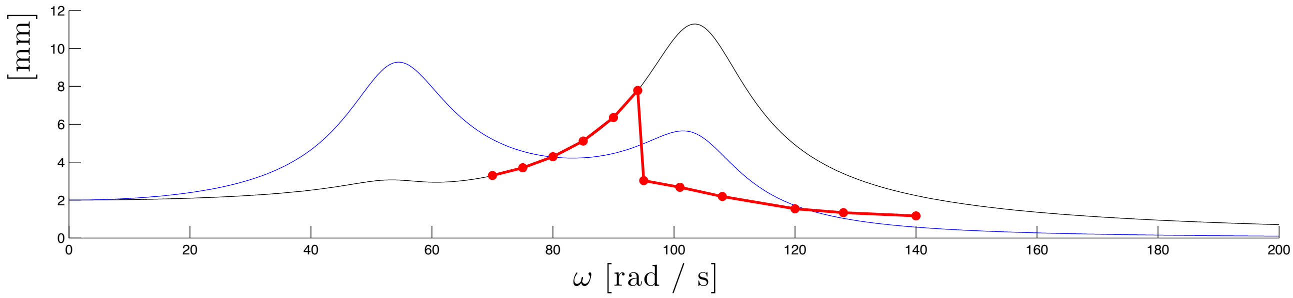

Fig. 10 *Black and blue:* the dependence of the peak-to-peak vibration amplitude of the rotor frame and impact element, respectively, on angular frequency of excitation of the baseplate for unlimited movement of the impact body; *red:* the dependence of the peak-to-peak vibration amplitude of the rotor frame on angular frequency of excitation of the baseplate for the movement of the impact body limited by clearances; for parameters $m_t = 25$ kg and $b_t = 500$ N s m^{-1}.

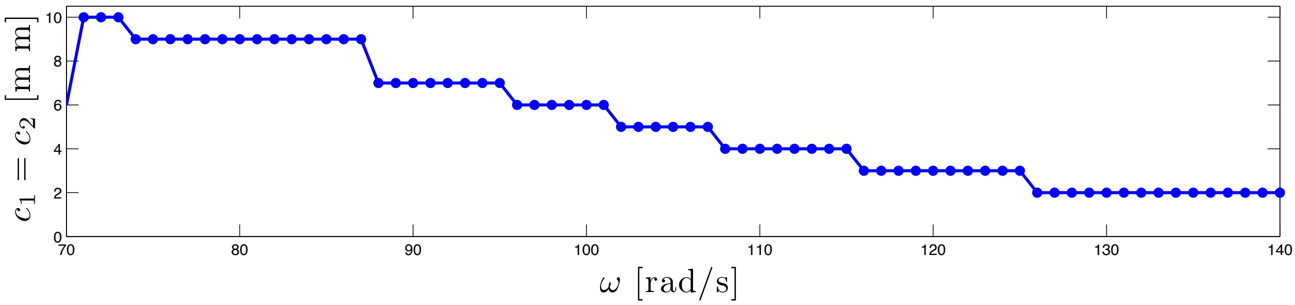

Fig. 11 Optimal choice of the clearance $c_1 = c_2$ for the maximal attenuation of vibrations of the rotor body corresponding to the *red* curve in Fig. 9, for parameters $m_t = 25$ kg and $b_t = 0$ N s m^{-1}.

no damping between the impact body and the rotor casing. For excitation frequencies higher than about 90 rad s^{-1} the impacts occur and the motion has a periodic character, mainly 1T. Nevertheless, in a short frequency range around the frequency of 149 rad s^{-1} a 3T periodic movement can be observed, see Fig. 14.

As the analysis carried out shows that influence of the impact damper on vibration attenuation of the rotor frame is more efficient for the arrangement with no additional viscous damping device placed between the impact element and the rotor frame, the further and more detailed study is referred only to this design case ($b_t = 0$ N s m^{-1}).

For the simulation parameters of the angular frequency $\omega = 100$ rad s^{-1} and the gaps between the rotor frame and the impact element $c_1 = c_2 = 6$ mm, $\omega = 150$ rad s^{-1} and $c_1 = c_2 = 1$ mm, respectively the Fourier spectra and phase trajectories show that the movement is formed only by the components having the basic and ultra harmonic frequencies as given in Fig. 15 and 16. The character of these diagrams shows that the system performs a periodic movement on these conditions.

The Fourier spectrum of the system response referred to the excitation frequency of $\omega = 110$ rad s^{-1} and the gaps between the rotor frame and the impact element $c_1 = c_2 = 5$ mm (Fig. 17) shows the side bands around the frequency peaks related to the principal and ultra harmonic components of the motion that is characteristic for quasi-periodic oscillations. The corresponding phase trajectory is depicted in Fig. 17(left).

Finally, the Fourier spectrum of the motion excited by the baseplate vibrating with the frequency of $\omega = 185$ rad s^{-1} and the gaps between the rotor frame and the impact element $c_1 = c_2 = 1$ mm (Fig. 18) has a band character between the distinct frequency peaks which confirms a chaotic pattern of the system oscillations. The high degree of irregularity of the motion is also evident from the phase trajectory depicted in Fig. 18(left).

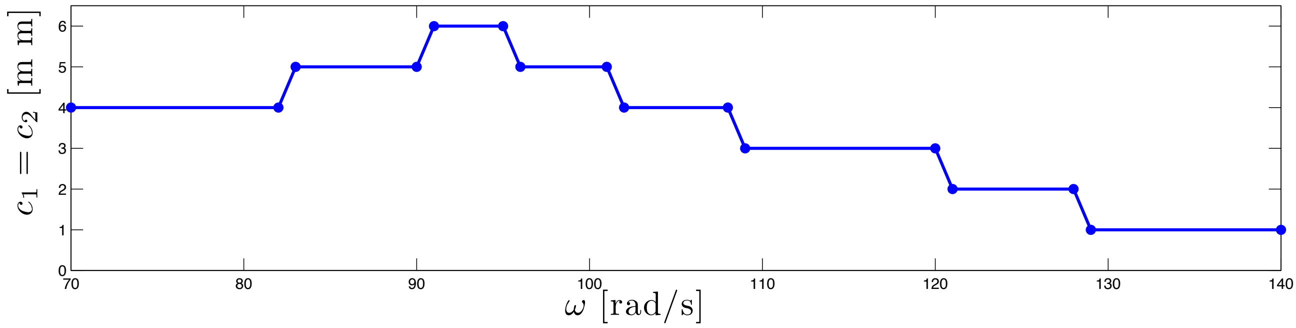

Fig. 12 Optimal choice of the clearance $c_1 = c_2$ for the maximal attenuation of vibrations of the rotor body corresponding to the *red* curve in Fig. 9, for parameters $m_t = 25$ kg and $b_t = 500$ N s m^{-1}.

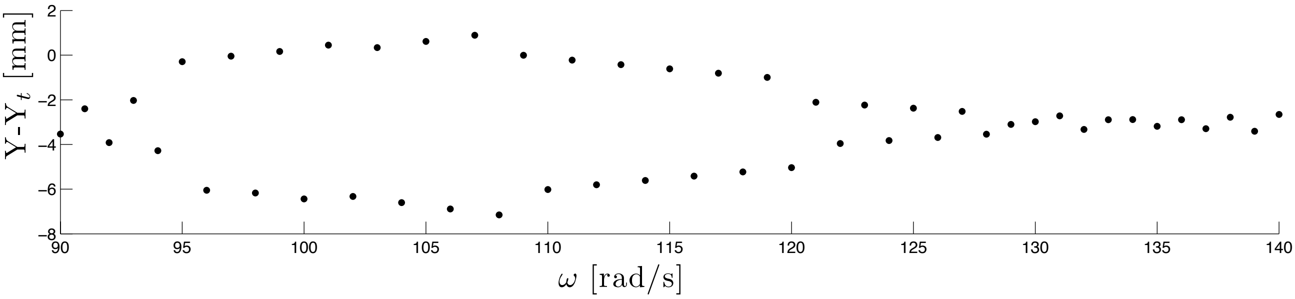

Fig. 13 Bifurcation diagram with respect to the angular frequency ω for parameters $m_t = 25$ kg and $b_t = 500$ N s m^{-1}.

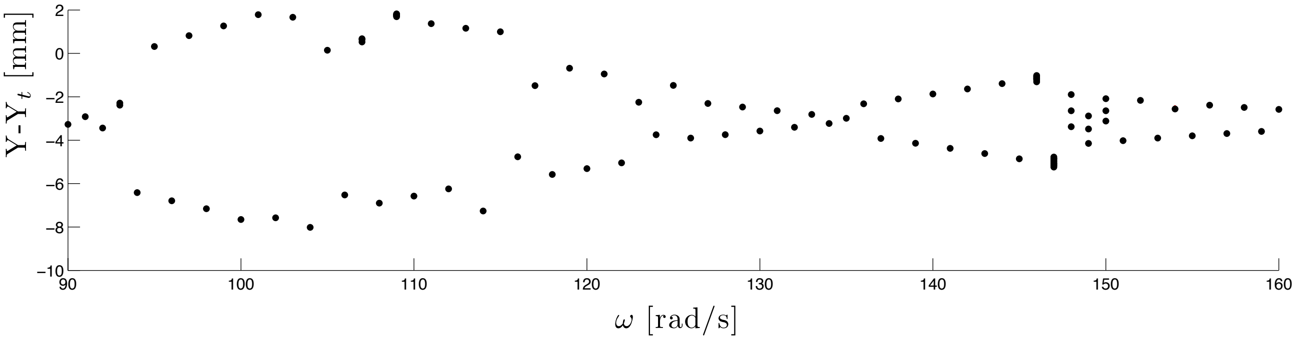

Fig. 14 Bifurcation diagram with respect to the angular frequency ω for parameters $m_t = 25$ kg and $b_t = 0$ N s m^{-1}.

6 Conclusions

In this paper, there was developed and analyzed a new rotating system damped by impact element with soft stops dependent on parameters, namely on its weight, on the width of the clearance between the impact body and rotor casing and finally on the excitation frequency. The used system is important because many technological devices in industry consist of a rotating machine powered by an electric motor and are excited by the rotor unbalance and the ground oscillations. The equations of motions were solved numerically by application of the explicit Runge-Kutta method. The computational simulations showed that the vibration of the baseplate plays a key role here and proved that application of the impact body arrived at a significant decrease of the rotor frame vibration amplitude. It was observed that for the given parameters, the damping element can work as inertia (without impacts) or as an impact damper (impacts occur). Both situations were investigated and are commented here. The ranges of parameters for which the attenuation is meaningful were detected. In the case of the inertia damper, the range of the excitation frequency where the damper works, is very narrow. On the

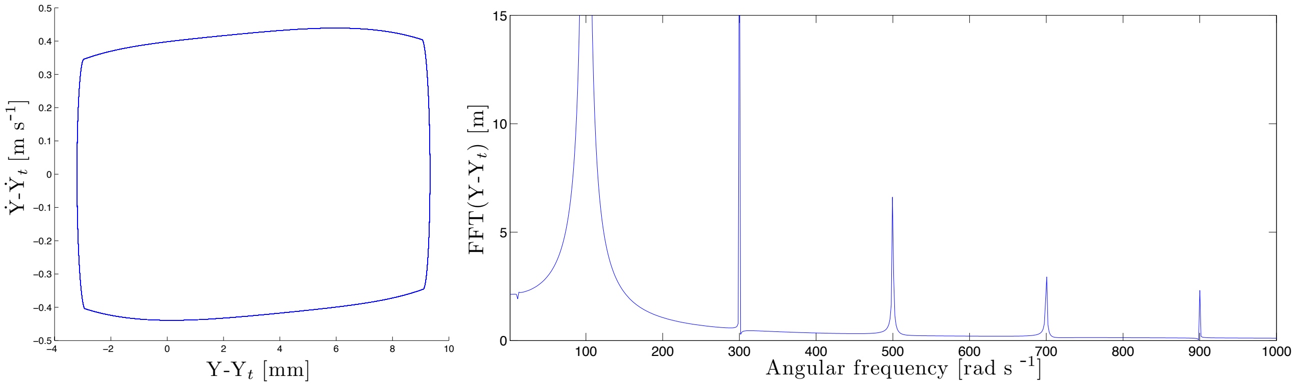

Fig. 15 Phase portrait (left), $\mathrm{Y}-\mathrm{Y}_t$ versus $\dot{\mathrm{Y}}-\dot{\mathrm{Y}}_t$ and Fourier spectra (right) for the angular frequency $\omega = 100$ rad s^{-1} and the gaps between the rotor frame and the impact element $c_1 = c_2 = 6$ mm.

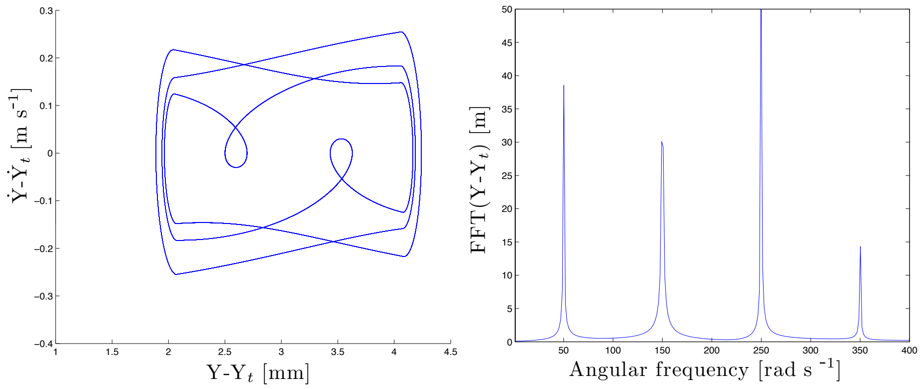

Fig. 16 Phase portrait (left), $\mathrm{Y}-\mathrm{Y}_t$ versus $\dot{\mathrm{Y}}-\dot{\mathrm{Y}}_t$ and Fourier spectra (right) for the angular frequency $\omega = 150$ rad s^{-1} and the gaps between the rotor frame and the impact element $c_1 = c_2 = 1$ mm.

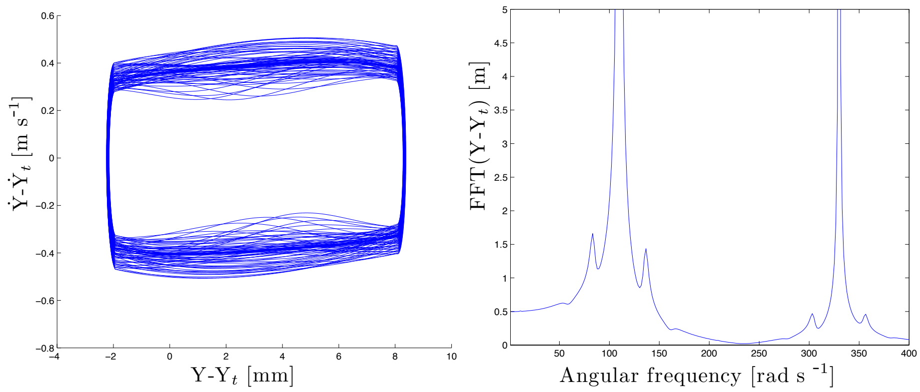

Fig. 17 Phase portrait (left), $\mathrm{Y}-\mathrm{Y}_t$ versus $\dot{\mathrm{Y}}-\dot{\mathrm{Y}}_t$ and Fourier spectra (right) for the angular frequency $\omega = 110$ rad s^{-1} and the gaps between the rotor frame and the impact element $c_1 = c_2 = 5$ mm.

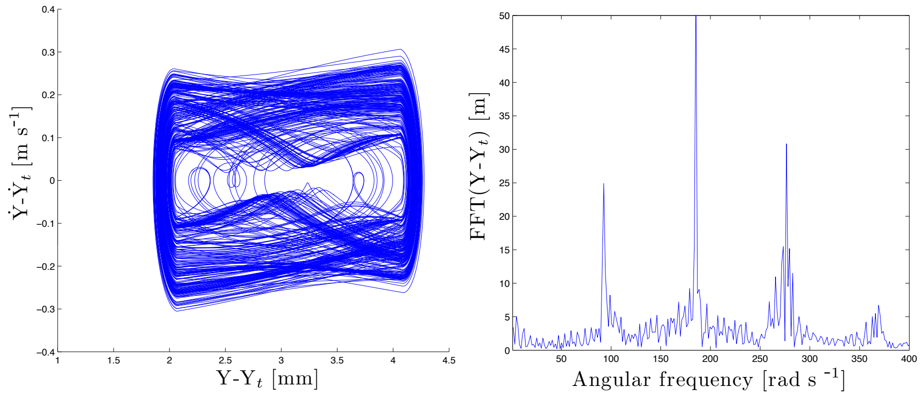

Fig. 18 Phase portrait (left), $\mathrm{Y}-\mathrm{Y}_t$ versus $\dot{\mathrm{Y}}-\dot{\mathrm{Y}}_t$ and Fourier spectra (right) for the angular frequency $\omega = 185$ rad s^{-1} and the gaps between the rotor frame and the impact element $c_1 = c_2 = 1$ mm.

other hand, it is quite wider in the case when the damping device works in the impact regime. An active control of the damping element can be achieved by changing the width of the clearance between the damping body and the rotor casing dependent on frequency of the baseplate excitation. The sensitivity of the system to the ground excitation decreases with rising frequency of the baseplate oscillations which means that amplitude of the induced vibrations of the rotor frame goes down and no damping device is needed. On the contrary, presence of the impact damper in these cases can arrive at elevation of the vibration amplitude and at irregular character of the system motion as was proved by the simulations.

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