TO THE PROBLEM OF SELF-EXCITED VIBRATION SUPPRESSION

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The aim of this contribution is to present a basic survey of publications dealing with the means of vibration suppression in several kinds of self-excitation. The attention is given to passive and active means using parametric excitation. A special attention is given to the latter and the necessary steps for answering important questions are formulated to initiate further investigation and especially application to real systems.

Key words: *self-excited vibrations, passive and active means for vibration suppression*

1. Introduction

Self-excited vibrations represent a broad class of systems. There are various reasons for their initiation. The most common sources of self-excitation are as follows:

a) Medium flowing around elastic or elastically mounted body (e.g. pylon, high voltage lines, turbine blade, airplane wing, valves).
b) Oil or gas bearings, seals etc. in rotating systems.
c) The relative dry friction (e.g. between work-piece and tool).
d) The internal damping in rotors and etc.

The intensity of self-excited vibration is often so high that a save run is impossible and can result in a failure. Considering its occurrence from the historical point of view in the last century it may be said that this kind of vibrations is more frequent and their importance is higher. The technical progress has increased the output, velocity and rotation speed of machines, which resulted in the more frequent occurrence of self-excited vibrations. Thus, for instance self-excited vibration due to the action of oil bearings first appeared in our country first in the early 1950's – first in turbo-compressors, later in turbo-sets. This problem was solved relatively soon be replacing cylindrical bearings with more stable bearings. All this initiated more research into these phenomena. It appeared that the most efficient approach to solving the problem is when at least a basic analysis was made using a simple model. This enabled the analysis of the different means and their efficiency in vibration suppression.

2. Some passive means for self-excited vibration quenching

Considering different sources and character of self-excitation the means for vibration suppressing are different. The first measure, (this is not always possible), is to remove the source of self-excitation, e.g. by changing the flow round the body or by replacing the bearings, as already mentioned, with more stable ones.

But in most cases it is necessary to use additional means to suppress the vibration. The most usual way is passive, based on increasing the positive damping. Various possibilities

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exist. One of them is the dynamic absorber usually represented by a mass elastically mounted to the vibrating part of the structure. The motion of the damper mass is damped. As for the externally excited vibration the analysis of the dynamic absorber was presented at the beginning of the last century and can be found in most books dealing with vibration theory (see e.g. [1]). The analysis of the dynamic absorber for quenching vibration excited parametrically has been published relatively recently [2]. Also the analysis of dynamic absorber for self-excited vibration suppression has been published during the second half of the last century. It is necessary to stress that the efficiency differs because of various characters of self-excitation. A summary of the analysis can be found in the book [3] and the references cited therein. In this book several models of self-excitation are presented, i.e. not only models using negative linear damping (e.g. van der Pol model) but also another models (e.g. using additional differential equations). Most models use additional linear differential equation expressing that the dynamic force is retarded behind the static one. In the mentioned book are analyzed several systems using these models.

One analysis in this book deals with the effect of tuned absorbers. The most important difference in absorber application for different kinds of excitation – external, parametrical and self-excitation – lies in the effect of absorber damping. While for external excitation the absorber damping should be as small as possible, this is not valid for parametric and self-excitation. The optimal damping coefficient of the absorber motion is dependent on the mass ratio (absorber mass to basic system mass) (see [3]). The tuning is optimal when the partial natural frequency of the absorber is close to the vibration frequency of the basic system to which the absorber subsystem is attached. Usually the lateral or angular deflections of the absorber mass describe its motion but the absorber can have the form of a pendulum. In this case when the motion of the body to which the absorber is attached is perpendicular to the pendulum axis then the same rule is valid when the pendulum maximum deflection do not exceed a certain limit. When this limit is exceeded then a chaotic motion (a mixture of swinging and rotation motions) or rotating motion occurs. This type of absorbers has been used for suppressing flow-induced vibration of high slender structures and towers.

Another tuning is optimal for the case when the body motion is parallel to the equilibrium position of the pendulum absorber axis, i.e. the natural frequency of the absorber pendulum should be close to the half frequency of the basic subsystem (mass mounted on a spring). This system belongs to the class of so called auto-parametric systems. The theory of this type of auto-parametric absorbers, for the case of external excitation of the basic system, is treated in the book [4]. As for the case of self-excitation the analysis of this type of absorber has been presented later. The analysis results are presented the book [5]. Different auto-parametric systems are analyzed in this book and characteristic properties are discussed. The specific property is the 'saturation effect' which is especially utilized with this type of absorbers. This phenomenon is characterized by a non-proportional energy flow into the different parts of the whole system, e.g. when increasing the excitation intensity the vibration increases substantially only in one part of the system. It is interesting that the analysis of these auto-parametric systems has been presented as late as the end of the last century. The same is valid for systems being modeled as a pendulum (free or elastically mounted). Analysis of dynamic absorbers of these pendulum type systems excited by all the types of excitation can be found in the monograph [6].

For another group of means the use of elastic element with damping is characteristic. A typical example thereof is the elastic mounting with damping of the bearing body, e.g. by 'squeeze film bearings', or elastically mounted foundation with damping. Several examples (analytical analysis and experiments) of such means can be found in the book [7]. A further example is the elastic mounting with damping of the originally fix axis around which the spindle rotor was rotating (a machine for cotton yearn spinning). Rolling bearings were replaced by air-pressurized ones, which for the desired higher than double rotating running speed, resulted in self-excited vibration initiation. The elastic mounting with damping of the axis enable a substantial increase of the running speed even higher than the originally desired one. The theoretical analysis is presented in the second part of the book [3].

In all the above-mentioned publications it has been supposed that the self-excitation is due to the negative linear damping or can be expressed by linear terms in the governing differential equations. These models are mostly used because in real systems this kind of self-excitations prevails.

However, the linear model need not be always a comprehensive one expressing all effects of self-excitation. One example of such a nonlinear model is the model of a system excited by vortex shedding. In this model, an additional differential equation is used which contains a nonlinear term (see [3]).

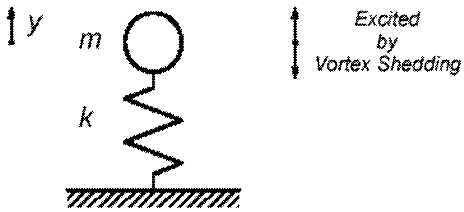

Fig.1: Schematic representation of basic system

The model of this system is governed by the following equations:

$$\begin{gathered} m\,\ddot{y} + b\,\dot{y} + k\,y = \gamma\,u\ , \\ \ddot{u} + \alpha_0\,\dot{u} + \omega_\mathrm{s}^2\,[u - \delta\ \mathrm{sgn}(\dot{y})] = 0\ , \end{gathered} \tag{2.1a}$$

ω_s – frequency of vortex shedding. These equations, after time transformation $\omega_0\,t = \tau$, ($\omega_0 = \sqrt{k/m}$), gets the form:

$$\begin{gathered} y'' + \kappa\,y' + y = \beta\,u\ , \\ u'' + \alpha\,u' + \omega^2\,[u - \delta\ \mathrm{sgn}(y')] = 0\ , \\ \kappa = \frac{b}{m\,\omega_0}\ , \qquad \beta = \frac{\gamma}{m\,\omega_0^2}\ , \qquad \alpha = \frac{\alpha_0}{\omega_0^2}\ , \qquad \omega = \frac{\omega_\mathrm{s}}{\omega_0}\ . \end{gathered} \tag{2.1b}$$

This model describes well the character of excited vibration especially the changing character of vibration when changing the vortex shedding frequency due to the change of flow velocity. In the region of vortex shedding frequency under the natural frequency of the system the vibration character is rather close to the character of the externally excited vibration, i.e. the vibration frequency is close to the vortex shedding frequency. But in the region of vortex shedding frequency over the natural frequency of the system the vibration frequency is close to the natural frequency of the system (see Fig. 2). This self-controlled system belongs to the mathematical models called in English literature bluff body models. The differential equations are so proposed to correspond to the considered system (see more in [3]).

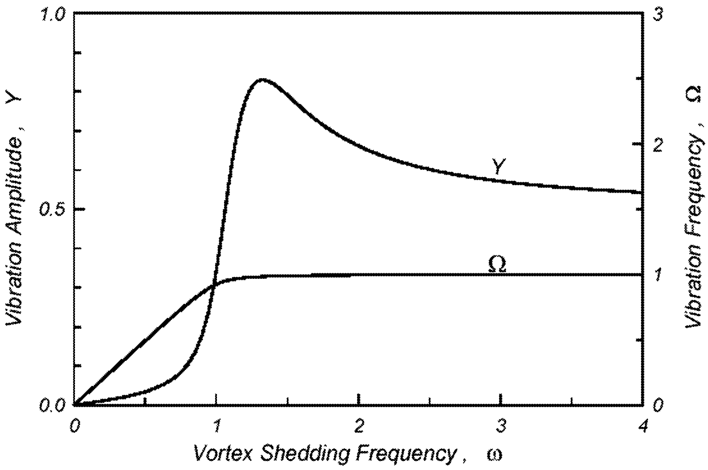

Fig.2: Analytically predicted vibration amplitude γ and frequency Ω as a function of vortex shedding frequency ω for basc system; the following parameters have been considered: $\kappa = 0.05$, $\alpha = 0.50$ and $\beta\,\delta = 0.02$

In this book is also presented the analysis of the effect of the linear dynamic absorber showing that for this system a full suppression cannot be achieved. The same situation was found by the analysis of other systems (e.g. a two-mass system where the upper mass is excited by vortex shedding (see [8]). For this two-mass system two vibration modes are possible. Three different vibrations can be initiated: single-mode vibration with the first or second mode and the double-frequency vibration. When the single-mode vibration is suppressed, it results in no case in vibration suppression but to the change in other mode vibration.

Different nonlinear models of self-excitation are analyzed for a two-mass system, where the upper mass is self-excited and the lower mass is damped considering different damping models (see [9]). The main result can be formulated as follows: A full suppression of the self-excited vibration can be achieved in the case as follows:

The component of foundation damping has the same form as the negative component of the self-excitation of the basic sub-system and, furthermore, certain conditions are met. With nonlinear self-excitation of a two-mass system deals a paper [61]. The analysis results show that for system with nonlinear self-excitation several steady state solutions can exist. The stable equilibrium position need not mean that the system is safe for self-excited vibration. Some disturbances can lead to violent vibration. This means that the linear approach (using only linear terms) need not guarantee the save run of the considered system.

A passive means for a valve system using the model with additional linear differential equation is treated in [10]. Fig. 3 shows the scheme of the considered system. This is governed by the following equations:

$$\begin{aligned} m_1\,\ddot{y}_1 + b_1\,\dot{y}_1 + k_1\,(y_1 - y_2) &= P\ , \\ m_2\,\ddot{y}_2 + b_2\,\dot{y}_2 - k_1\,(y_1 - y_2) - k_2\,(y_2 - y_0) &= 0\ , \\ \dot{P} - \alpha_0\,[F(y_1) - P] &= 0\ . \end{aligned} \tag{2.2}$$

The latter equation of the first order is expressing the retardation of the dynamic force (acting on the valve) behind the static one. The idea of these models is treated in Part III

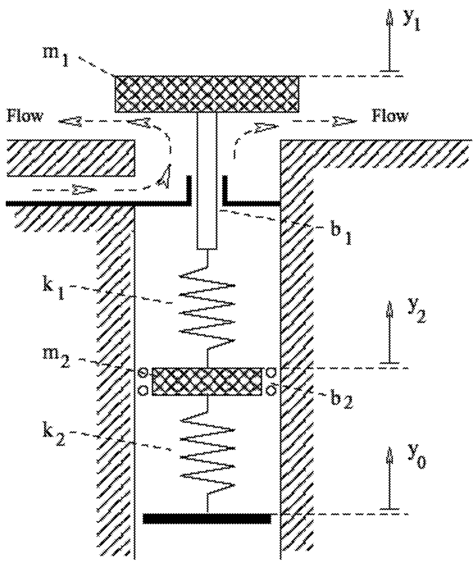

Fig.3: Scheme of the system

of the book [3]. The analysis results of this system are presented in [10] and illustrated by diagrams of stability boundaries for both vibration modes and different parameter values.

3. Parametric excitation as an active means for self-excited vibration suppression

Before dealing with the active means using parametric excitation let us mention two papers dealing with the effect of external excitation (without parametric excitation) on self-excited system (see [11], [12]). When the passive means is not able to suppress the self-excited vibration then the external excitation can, under certain conditions, reduce the vibration.

Parametric excitation is due to periodic variation of system parameters (spring stiffness, damping coefficients, mass or inertial moment). The motion of such a system is governed by differential equations with periodic coefficients (having period $2\pi/\omega$). In most cases the effect of the parametric excitation is undesirable because it results in the instability of the equilibrium position in some intervals of the frequency ω and initiation of parametric resonance. This can occur in certain interval of frequency close to ω

$$\omega = \frac{\Omega_j \pm \Omega_k}{N} \qquad (j,k = 1,2,\dots,n;\ N = 1,2,\dots) \tag{3.1}$$

Ω_j, Ω_k – natural frequencies of the system.

If $j = k$ we speak about instability intervals and resonance of the first kind and Nth order, if $j \neq k$ we speak about resonance of the second kind or combination resonance. These of the first order are most important especially in the case of harmonic parametric excitation. In the analyzed systems for periodic variable stiffness only one sign (see (3.1)) is valid for the combination resonance, which was proved for harmonic stiffness variation (see [13]). When the system is self-excited and also the parametric excitation is acting, the self-excited vibration can be synchronized by the action of parametric resonance of the first and second kind as well (see [14]). By simulating on analogue computer in a broader interval of parametric excitation frequency the following phenomenon was found: If parametric combination resonance occurs in a certain interval at $\omega = \Omega_j + \Omega_k$ $(j \neq k)$ then at $\omega = |\Omega_j - \Omega_k|$ the self-excited vibration can be partly or even fully suppressed. This phenomenon was called **parametric anti-resonance**. The conditions for the full suppression were mathematically formulated first in [15] and [16]]. Now there exist numerous publications dealing with the use of this phenomenon (see [16] to [60] and [62]).

Several simple models of systems self-excited by flow are analyzed in [17], [18], [19], [22] where the van der Pol model is used (variable stiffness as parametric excitation is applied).

For illustration let us show an example (see more in [22]). Fig. 4 shows the scheme of the system which is described by the following differential equation:

$$\begin{gathered} m\,\ddot{y} + [b - \beta_0\,U^2\,(1+\gamma\,y^2)]\,\dot{y} + k\,y - \frac{1}{2}\,k\,y_0 = 0\;, \\ m_0\,\ddot{y}_0 + b_0\,\dot{y}_0 + k_0\,(1+\varepsilon\,\cos\omega t)\,y_0 - \frac{1}{2}\,k\,(y-y_0) = 0\;. \end{gathered} \tag{3.2}$$

These after time transformation: $\Omega_0\,t = \tau$ $(\Omega_0 = (k/m)^{1/2})$ get the form:

$$\begin{gathered} y'' + [\kappa - \beta\,V^2\,(1-\gamma\,y^2)]\,y' + y - \frac{1}{2}\,y_0 = 0\;, \\ y_0'' + \kappa_0\,y_0' + q^2\,(1+\varepsilon\,\cos\eta\tau)\,y_0 + \frac{1}{2}\,M\,(y_0-y) = 0\;, \\ \kappa = \frac{b}{m\,\Omega_0}\;,\quad \beta = \frac{\beta_0\,U_0^2}{m\,\Omega_0}\;,\quad V = \frac{U}{U_0}\;,\quad \kappa_0 = \frac{b_0}{m_0\,\Omega_0}\;, \\ q^2 = \frac{\omega_0^2}{\Omega_0^2}\;,\quad \omega_0^2 = \frac{k_0}{m_0}\;,\quad M = \frac{m}{m_0}\;,\quad \eta = \frac{\omega}{\Omega_0}\;. \end{gathered} \tag{3.3}$$

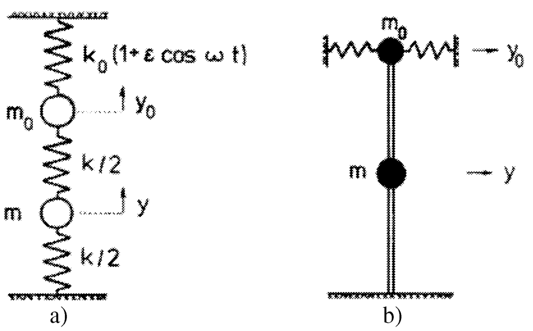

Fig.4: Scheme of the system

The results of numerical analysis of this for given parameters are presented in Fig. 5. It can be seen that a full suppression can be achieved in a certain interval of parametric excitation frequency ω round $\omega = \Omega_2 - \Omega_1$.

The linear parametric excitation can be also successfully used for the following systems: An linear additional differential equation expressing the idea that the dynamic force is retarded behind the static one is added to the governing differential equations (see [41]).

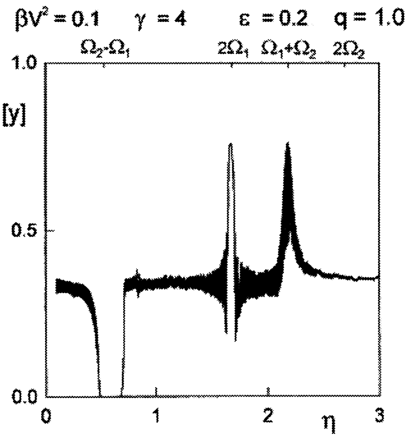

Fig.5: Extreme of y, y_0 in dependence on parametric excitation frequency ω

In the case when vibration consists of self-excited and externally excited components then the self-excited component can be suppressed with the parametric excitation. The external excitation does not influence the quenching ability of the parametric excitation on the self-excited vibration component (see [29]).

More complicated systems are treated in [24], [33] and also the models of rotor systems [20], [21], [35], [37].

The results of systems with more degrees of freedom can be summarized as follows: The self-excited vibration can be fully suppressed in the case when the equilibrium to a single vibration mode is unstable (i.e. only the single-frequency vibration can occur) and certain conditions are met. In the case when several vibration modes can be self-excited then a full vibration suppression cannot be achieved even if a double-frequency parametric excitation is used (see [36], [40]).

In the mentioned simple systems the parametric excitation is considered as acting in a single spring. In some analyzed systems more parametric excitations are considered, e.g. in some rotor system or in the mentioned more complicated systems. The question arises about the phase between the excitations. The optimization of the multi-location parametric excitation by stiffness variation is analyzed in [44].

Up to now only the action of linear parametric excitation in systems where the self-excitation is expressed by linear terms in differential equations has been analyzed. Of course, not all models of self-excitation are expressed by linear terms.

Although for most real systems the self-excitation can be described by linear terms there exist systems where the self-excited vibration is better described by nonlinear terms. This is valid e.g. the modeling of a system excited by vortex shedding (see [3], [8]).

Up to now mostly as parametric excitation the stiffness variation has been used. First step to parametric excitation by variation of mass or moment of inertia presents the paper [57] and [59]. In the latter the system described by the lateral and angular deflections is analyzed. One important result was found: The parametric anti-resonance exists close to the parametric combination frequency of the sum type. The research in this way should be continued also as for the question of efficiency and realization possibilities. In paper [62] a two-mass chain system is investigated. Two alternatives are considered: Self-excitation acts on the upper mass (a), on the lower mass (b). For both alternatives the parametric excitation due to the periodic mass variation is considered. The suppression effect occurs at parametric excitation frequency of difference type for both alternatives similarly as for parametric excitation due to the stiffness variation.

The combination of passive and active means need more detailed analysis as for the optimal tuning of the system (e.g. of the absorber) because the optimal tuning for the system with passive means need not be the same as for combination of both means as it is shown in [58].

The investigations having been made in this field are not only limited on analytic investigations but also experimental research have been made, which is important not only for comparison reason with analytic results but also for initiation applications in real systems (see [48], [49], [51]). All experimental investigations have been made in the Institute of Mechanics and Mechatronics, Vienna University of Technology.

4. Questions and problems to be answered and analyzed

As for the passive means the analysis of the stability in the large, i.e. for arbitrary disturbances, for systems where more locally stable vibrations can exist should be analyzed. This problem concerns for example the systems excited by vortex shedding (see [8]). For solving this problem the method using the disturbance defined by two quantities (e.g. an external force pulse defined by the amplitude and interval of action – see [55], Chapter 7) could be used. Such a disturbance can better correspond to reality. Main advantage is: the results can present two-dimensional diagrams.

Up to now the active means have been considered for systems where the parametric excitation is also linear. It would be reasonable to consider also the action of 'quasi-linear' parametric excitation for example characterized by the terms:

$$\varepsilon\, k\,[(1+\delta\, y^2)\,\cos\omega t]\, y\ , \tag{4.1}$$

ε, δ are small parameters. This can be the case when the 'stiffness' of quasi-linear spring is varied. Of course, the variation need not be harmonic but periodic only, e.g. instead of $\cos\omega t$ the term $\mathrm{sgn}(\cos\omega t)$, which in some real systems can be realized easier, e.g. by using reversible periodic change of the leaf spring length.

As it was mentioned in the previous chapter not all self-excitation types can be described by linear terms, e.g. the mentioned system excited by vortex shedding or other nonlinear excitation types (see [9]).

The question can read:

Is it possible to use parametric excitation? Can this be linear or must it be nonlinear or combination of both? What kind of parametric excitation is efficient (stiffness, damping, mass/moment of inertia)?

Can efficient nonlinear parametric excitation also be achieved?

As regards the model describing the systems excited by vortex shedding, it is necessary to consider the following fact: Passive means using damping fail to achieve full vibration suppression (in the systems that have so far been analyzed). It is necessary to answer the question whether this is generally valid for systems having more than two natural vibration modes. In cases where this would be really true, the active means using linear parametric excitation can only have limited success because, as has been stressed, only one single vibration mode can be unstable in order that full suppression could be achieved. Substantial suppression of vibration would be considered a good result. Similarly it can hardly be expected that by using linear parametric excitation full vibration suppression could be achieved for other types of nonlinear self-excitations.

To apply only the passive means it does need not bring satisfying results. Combination of both passive and active means is the best way, but then it is important find the best possible arrangement and tuning of the system.

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Received in editor's office: July 7, 2008
Approved for publishing: August 11, 2008