MODERN AVIATION AND SPACE TECHNOLOGY

UDC 531.7

Mykola Kulyk[1]
Volodymyr Kvasnikov[2]

CONTROL OF RIGIDITY OF AN ELASTIC ELEMENT ACTIVE VIBROSUPPORT

National Aviation University
Kosmonavta Komarova avenue 1, 03680, Kyiv, Ukraine
E-mails:[1]kms@nau.edu.ua, [2]kvp@nau.edu.ua

Abstract. *The device and control system for reduction of influencing of vibration from the power-plant with bearing spirally on the fuselage of helicopter are offered. It is shown, that the management by inflexibility of resilient element provides possibility of more flexible and effective construction of algorithm of management by the system of active vibroprotection.*

Keywords: bearing spirally, control of rigidity, fuselage of helicopter, vibration, vibroprotection.

1. Introduction

Reducing dynamic response on the machines for bruises, ensuring reliable operation in complex dynamic loads on the working parts of precision equipment – this is not a complete list of current areas of theoretical and experimental research in the field of vibration protection and vibration isolation.

Usually, the implementation of specific technical and technological solutions, allowing almost completely isolate the protection from external vibrations and vibrations and greatly reduce the impact of external low-frequency vibration. But with the growth of technology and the growing requirements for vibration protection.

Based on the above, work towards building an active vibration protection system is relevant to many areas of technology.

2. Formulation of task

To develop the control system, that provides compensation of vibration from the power-plant with bearing spirally on the fuselage of helicopter and to build the algorithm of effective control by the active system of vibrosupport.

3. Solution of a problem

Let's observe passive dissipative system of a damping of vibrating oscillations as a part of active vibrosupport from the assumption that the active element is in a static condition. It can be presented as installation of protection in weight of M which is isolated from a basis by the device with a viscous friction which gives a tractive resistance and transforms a vibratory movement kinetic energy into warmth further – elastomer element (schematically shown in the form of in parallel connected: springs with pressure factor k and the impulse neutralizer with damping coefficient b). The system is represented on Fig. 1.

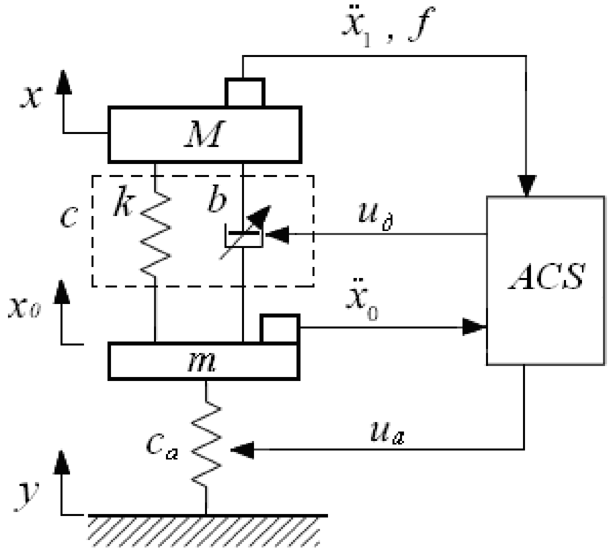


Fig. 1. The circuit design of management of rigidity of an elastic element:
y – vibrating moving of a basis (y=$h$sin(ωt), where h – amplitude of oscillations);
x – installation moving of a vibroprotection object in weight of M;
x_0 – moving of intermediate weight m;
k – pressure factor;
b – damping coefficient;
c – rigidity of elastomer element;
c_a – rigidity of the piezo actuator;
u_∂ – management of a passive elastic element;
u_a = Hsin(ωt) – management of actuator (H – amplitude of oscillations)

The working element of system of passive protection is characterised by pressure factor k=const and damping coefficient b which is in the following dependence with force of vibrating perturbation

$$F = b(V_y - V_x)\cdot$$

In an equilibrium state, in the absence of external vibrating perturbation y, force of pressure of a working element is counterbalanced by full weight M. In such condition we define dynamic rules in the counterbalanced condition

$$x = x_0 = y = 0,$$

and relative speed

$$V = \frac{(\Delta y - \Delta x)}{\Delta t} = 0\cdot$$

At emersion of vibrating perturbation at there are relative movings.

The dynamic force from an elastic element will be reaction to it:

$$F = k(y - x)\cdot$$

Relative speed also will be not zero, therefore in the impulse neutralizer there is a force directed against speed of perturbation:

$$F = b\,V.$$

Thus passive dissipative part of vibrosupport will be characterised by constant parametres k, b and M (Vibrations…1995).

However, to observe this system more widely it is necessary to increase a set of parametres for observation:

– a resonant frequency which presents system free oscillation in absence of any friction:

$$\omega_0 = \sqrt{\frac{c}{M}}, \tag{1}$$

where c – rigidity:

$$c = f\,(k, b);$$

– good quality which characterises fading properties of system

$$Q = \frac{\omega_0}{b}\cdot$$

Transfer ratio T that is a relationship between basis and object of vibroprotection:

$$T = \frac{x}{y}\cdot$$

From Fig. 2 it is visible that for system of passive vibroprotection on low frequencies transfer ratio $T \approx 1$, i.e. moving of object of protection is identical to moving of a basis.

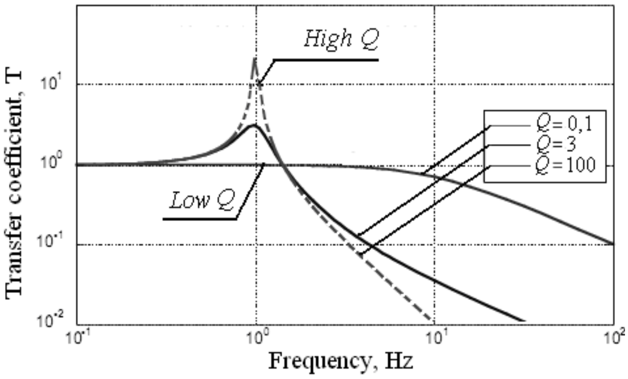


Fig. 2. Dependence of a transfer ratio of an idealised passive element on frequency and good quality

The passive element does not work as a vibroprotection.

Close to a resonance (transfer ratio T > 1), moving of the object of protection is more than basis moving. In this area of frequencies the system is characterised by good quality Q (the more Q, the more amplitude of movings).

Above a resonant frequency the moving of the object of protection decreasing that shows the work of a passive dissipative element: the higher is frequency the more effective is suppression of oscillations.

In general the vibration damping system a passive elastic element can be comparable to the filter of low frequencies. It has two big deficiencies: close to a resonant frequency which always exists, the increase in perturbations is observed, and in lower frequencies the system does not provide isolation from oscillations.

The analysis of constant forced oscillations of weight M and additional weight m showed that weight M has a regime of dynamic clearing of oscillations

$$\omega_г = \sqrt{\frac{c_a + c + c_a c + \frac{cH}{h}}{m}}\,. \tag{2}$$

The additional weight m has a regime of dynamic clearing of oscillations

$$\omega_г^0 = \sqrt{\frac{c_a + \frac{c_a h}{ch + H}}{M}}\,. \tag{3}$$

Analysing expressions (1), (2), (3) it is possible to draw following leading-outs that management of rigidity of an elastic element gives the chance to change own frequency of system and regimes of dynamic clearing of oscillations of the object of vibroprotection.

Amplitude – frequency characteristic of vibroprotection system (Fig. 1) resulted on Fig. 3.

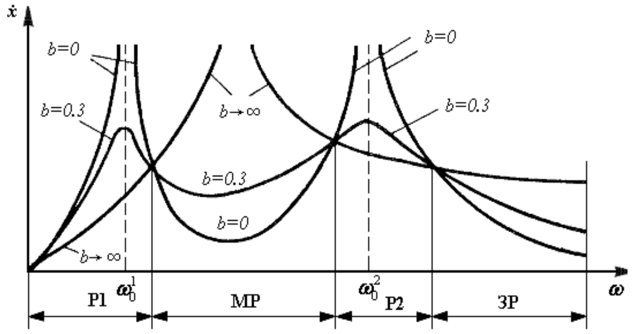


Fig. 3. Frequency response of active vibroprotection system

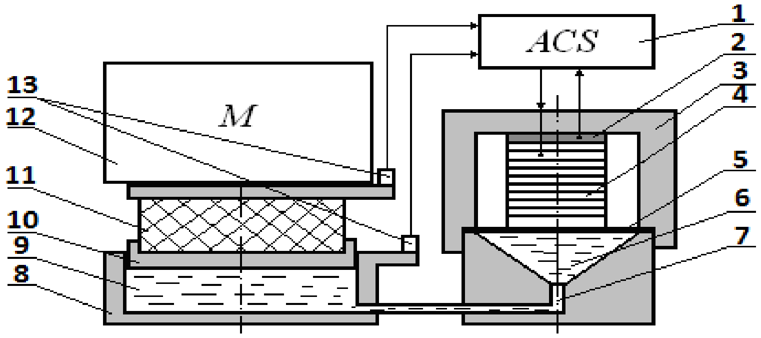
The principle of work of a control system (Fig. 1) magnitude of a damping (rigidity) of an elastic element, consists that measure frequency of forced oscillations of object of vibroprotection. Then, by means of band filters resonant and nonresonant produce matching signals, compare them on magnitude and depending on result of comparison instal magnitude of force which acts on an elastic element and changes magnitude of a damping. Magnitude of a damping operate as follows: if the signal from resonant filters (zones P1 and P2 on AFC Fig. 3) exceeds a signal with not resonant (MP and 3P) magnitude of a damping increase, otherwise reduce.

The offered way of management is realised by means of the special device which circuit design represented on Fig. 4, *a*. The given device is protected by the patent of Ukraine for useful model № 52666 (Kulyk et al. 2010).

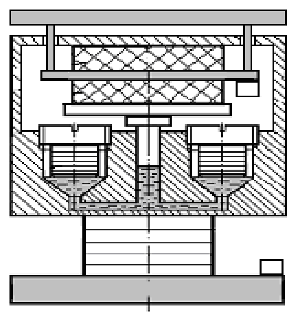
Management of rigidity is carried out thus. Necessary nominal rigidity of an elastic element 11 at the expense of its trimming operation by pressure of a liquid which moves in the chamber *9* of the chamber *6* by means of actuator *4* on the diaphragma *5*, through a highway *7* is preliminary installed.

The regulator operated a control system *4* holds pressure liquids in chambers *9* and *6*. Rigidity of an elastic element *11*, so, and frequency of natural oscillations and installation of protection of trimming operation of an elastic element of *12* proportional extents *11* and magnitude of pressure of a liquid in the chamber *9*.

Magnitude of pressure of a liquid *6* is measured in the chamber by the pressure unit *2* and arrives in a control system *1* which on magnitude of pressure defines matching value of frequency of natural oscillations of installation of protection *12*. Forced oscillations of installation of protection are measured by the sensing transducer of frequency *13*. The measured value of frequency of forced oscillations is transferred in a control system *1* in which the management signal by actuator *4* is produced.



a



b

Fig. 4. The circuit design of a control mean rigidity of an elastic element of vibroprotection system (*a*) and its implementation in a design of active vibrosupport (*b*):
1 – control system; *2* – pressure unit; *3* – case; *4* – piezoactuator; *5* – diaphragm; *6* – hydraulic chamber; *7* – hydraulic highway; *8* – basic basis; *9* – hydraulic chamber; *10* – piston; *11* – elastic element of the controlled rigidity; *12* – installation of protection; *13* – sensing transducer of frequency

Apparently from AFC on Fig. 3 amplitude of oscillation, at the same value of damping coefficient *b*, different in the observed zones on a frequency axis. A particular interest, from the practical point of view, at construction of active vibrosupports under the circuit design of two weights (Fig. 1) the zone between resonant frequencies *MP* (represents Fig. 3). The width of zone *MP* (a frequency range between two resonances) is defined by magnitudes of weights *M* and *m*. Whereas the weight of *M* is object of vibroprotection and can adopt a value from $M_{\min}$ (work at no-load installation of measurements) to $M_{\max}$ (work with the maximum loading installation of measurements), the weight *m* is one of constructive elements of vibrosupport. Thus at designing of vibrosupport there is a possibility to count the minimum interresonant interval *MP*. Practical researches showed that for decrease in resonant perturbations in a zone of frequencies of the second mechanical resonance it is necessary that condition $M / m > 50$ was satisfied.

The realised way of management of rigidity of an elastic element *11* automatically, with high reliability to avoid resonant regimes of oscillations of installation of protection in a wide range of frequencies at the expense of timely and sufficient on magnitude of change of frequency of natural oscillations of installation.

Management of rigidity of an elastic element gives the chance more flexibly and to build effectively a control algorithm of active vibroprotection system. If u_a (Fig. 1) provides management on an entry applying u_∂ was possibly to provide management on a system exit. Management on an exit is observed considering constructive construction of installation of protection which inertia masses create additional agencies on oscillatory system.

The management signal design u_∂ is spent taking into account dependence of deformation of an elastic element on moving of the piston *10* (Fig. 4), pressures *P* in the hydraulic chamber *9*, and dependences of factor of change of rigidity on change of linear dimensions of an elastic element. For small movings of the piston *10* in a hydraulic system and pressure *P* there is a linear dependence. Therefore further the accepted linear dependence of rigidity on magnitude of compression (model of Treloara for a hyperelastic incompressible material) (Kobets et al. 2008).

The control system for formation u_∂ carries out the analysis of amplitude of movings of weight of *M* scales it and defines a management step. Managements it is carried out in a following regime because in an operating mode as a result of a vibration damping, the elastic element heats up.

Application of the device for decrease of vibrations in a pulse-jet helicopter fuselage is shown on Fig. 5.

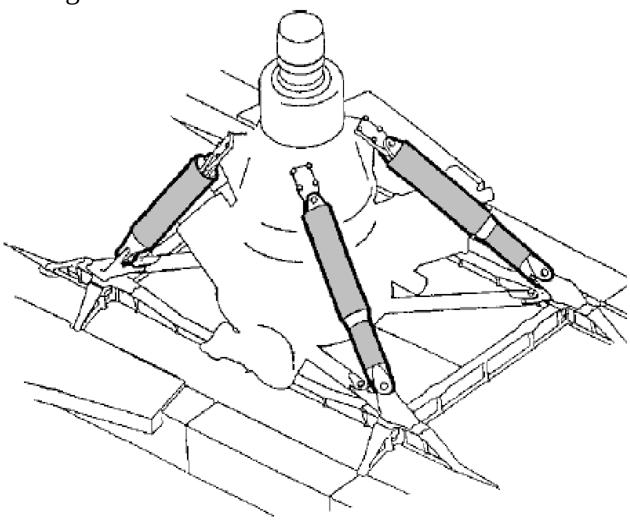


Fig. 5. Application of the device for decrease of vibrations in a pulse-jet helicopter fuselage

The control algorithm is resulted by rigidity of an elastic element at Fig. 6. The Control system for formation u_∂ carries out the analysis of amplitude of upright conveyances of weight of *M* scales it and defines a management step.

Management is carried out in a watching regime because in an operating mode as a result of a vibration damping, the elastic element heats up. At heating of the characteristic of an elastic element change on the nonlinear dependences, therefore watching management gives the chance to avoid calculations of corrections for their compensation.

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4. Conclusions

It is shown, that forming an additional connecting link in system of fastening of the rigid rotor to a fuselage, the offered device and a control system can be applied to decrease of agency of vibration from the propulsion system with the rigid rotor on a fuselage of the helicopter.

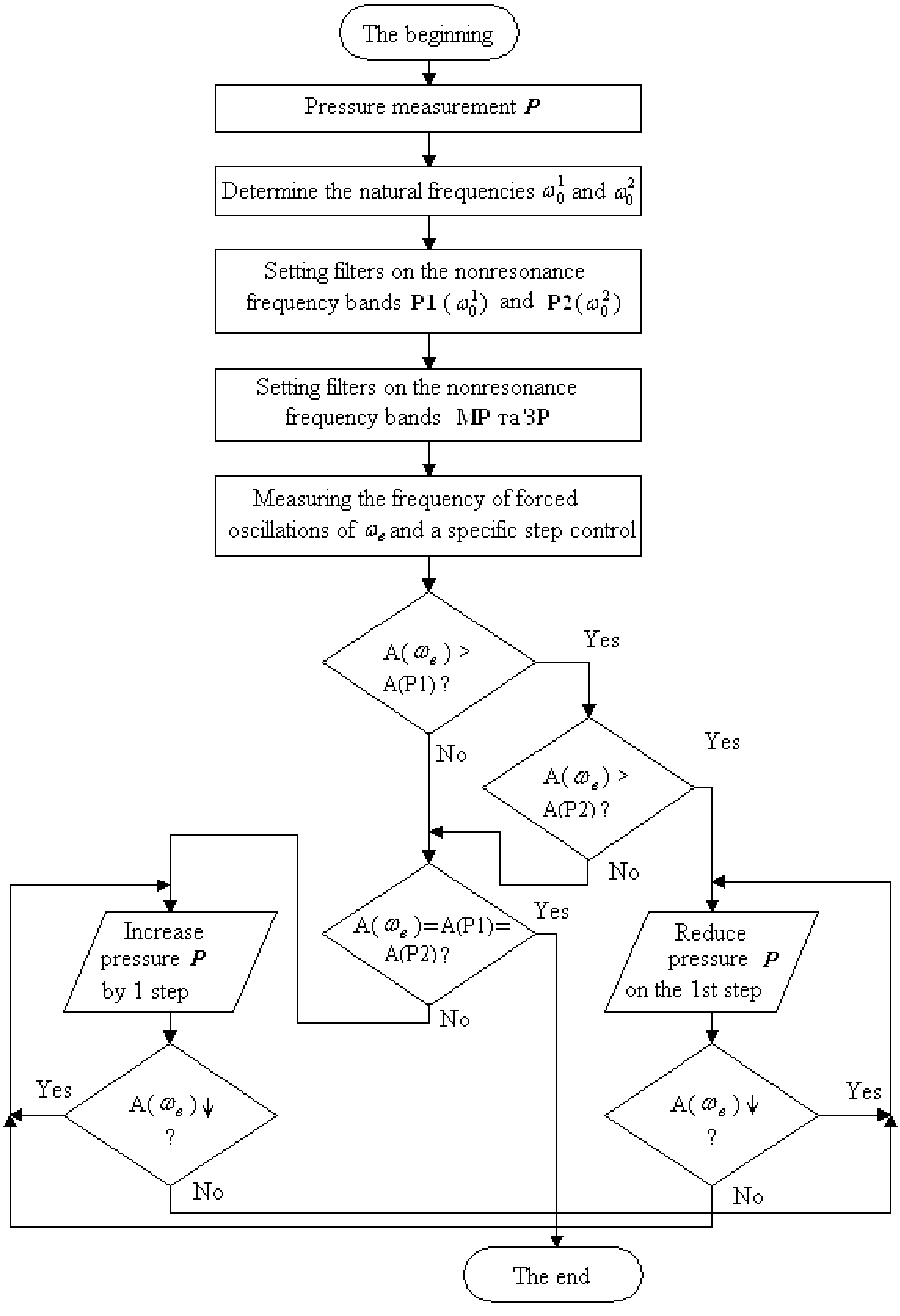

Fig. 6. The control algorithm of the elastic stiffness of the element active antivibration mountings

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Received 26 March 2013.

М.С. Кулик[1], В.П. Квасніков[2]. Керування жорсткістю пружного елемента активної віброопори
Національний авіаційний університет, проспект Космонавта Комарова, 1, Київ, Україна, 03680
E-mails:[1]kms@nau.edu.ua, [2]kvp@nau.edu.ua

Запропоновано пристрій та систему керування для зменшення впливу вібрації від силової установки з несучим гвинтом на фюзеляж вертольоту. Показано, що керування жорсткістю пружного елемента надає можливість більш гнучкої та ефективної побудови алгоритму керування системою активного вібрзахисту.

Ключові слова: вібрація, вібрзахист, несучий гвинт, керування жорсткістю, фюзеляж вертольоту.

Н.С. Кулик[1], В.П. Квасников[2]. Управление жесткостью упругого элемента активной виброопоры
Национальный авиационный университет, проспект Космонавта Комарова, 1, Киев, Украина, 03680
E-mails:[1]kms@nau.edu.ua, [2]kvp@nau.edu.ua

Предложены устройство и система управления для уменьшения влияния вибрации от силовой установки с несущим винтом на фюзеляж вертолета. Показано, что управление жесткостью упругого элемента предоставляет возможность более гибкого и эффективного построения алгоритма управления системой активной виброзащиты.

Ключевые слова: вибрация, виброзащита, несущий винт, управление жесткостью, фюзеляж вертолета.

Kulyk Mykola. Doctor of Engineering. Professor. Holder of a State Award in Science and Engineering of Ukraine (2003). Winner of a State Prize of Ukraine in Science and Engineering (2005).
Rector of the National Aviation University, Kyiv, Ukraine. Head of the Department of Aircraft Engines, National Aviation University, Kyiv, Ukraine.
Education: Kyiv Institute of Civil Aviation, Kyiv, Ukraine (1978).
Research area: definition of the technical state of aircraft engines.
Publications: over 200.
E-mail:kms@nau.edu.ua

Kvasnikov Volodymyr. Doctor of Technical Sciences. Professor.
Education: Odessa State University (1976), Kyiv Polytechnical Institute (1982).
Head of Department of Information Technology, National Aviation University, Kyiv, Ukraine.
Research area: co-ordinate-measuring machines, the integrated robotics measuring complexes.
Publications: over 230.
E-mail: kvp@nau.edu.ua