Table 2

Regimes implemented in a system with dampers with different numbers of optimized parameters depending on the exciting force frequency at P=800 N

ω, rad·s^{-1}	6.2	6.3	6.4	6.5	6.7	7.0
m_2=39.7 kg						
with 2 optimized parameters	T,3,3	Transient chaos; *T*,2,4	*T*,2,3	Chaotic	Chaotic	*T*,1,2
with 7 optimized parameters	T,1,2	T,1,2	T,0,2	T,0,1	T,0,1	T,0,1
m_2=62.0 kg						
with 2 optimized parameters	Chaotic	Chaotic	Chaotic	Transient chaos; *T*,2,2	Chaotic	Chaotic
with 7 optimized parameters	Chatter	T,3,3	Chatter	T,2,2	T,2,2	T,2,1

For these dampers with 7 optimized parameters, the system movement is “calmed down”; irregular modes are replaced by regular periodic regimes. For the damper m_2=39.7 kg, we see the modes without direct damper impacts on the primary structure. Thus, different optimization procedures produce different damper parameters, so analysis is necessary to define, which variant is the most effective.

3.3. Selecting the optimal damper parameters

A logical question arises: which of these four options, determined by the optimization procedures, is preferable? Which option provides the best damper performance?

Let us compare the efficiency of four damper variants with 7 optimized parameters, that is, let us see how they attenuate the primary structure energy.

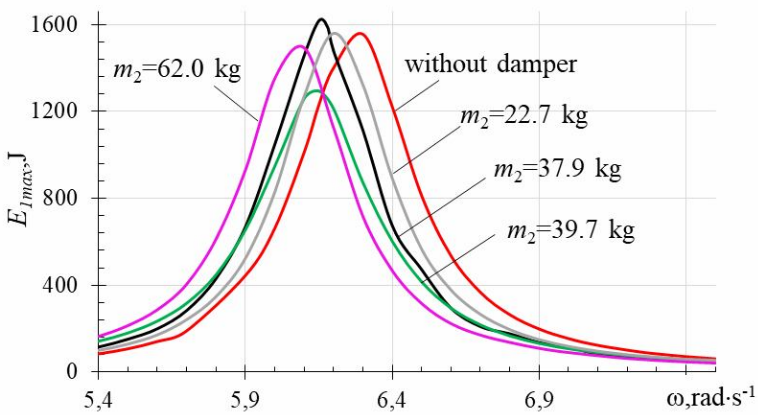


Fig. 4. The maximum total energy of the primary structure coupled with different SSVI NES depending on the exciting force frequency

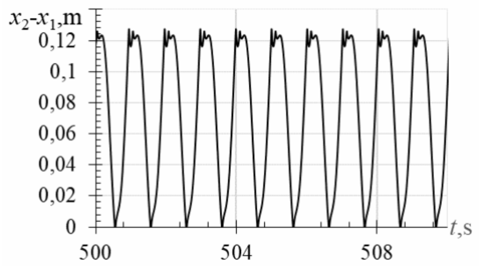
Fig. 4 shows that all dampers shift the resonant peak of the maximum total energy of the primary structure to the left towards the low exciting force frequencies. Only a damper with a mass of m_2=39.7 kg (green curve) significantly reduces it; a damper with a mass of m_2=62.0 kg (lilac curve) reduces it a little. The energy mitigation occurs at fairly high frequencies. At low frequencies, the primary structure energy not only does not attenuate, but, on the contrary, increases. We believe that from these 4 variants proposed by the optimization procedures, it is worth choosing a damper of mass m_2= 39.7 kg (green curve in Fig.4). Firstly, it reduces the energy resonant peak quite well, better than other dampers. Secondly, it attenuates the primary structure energy at high frequencies also better than other dampers. Thirdly, it ensures regular periodic motion over a wide frequency range, as shown in Table 2. Its mass is 4% of the primary structure mass. It is worth noting that at high frequencies a damper with mass m_2= 62.0 kg mitigates the energy better. However, it is too heavy (its mass has 6% of the primary structure mass) and provides a high resonant peak. Table 3 shows the attenuation of the maximum total energy of the primary structure coupled with the dampers of mass m_2= 39.7 kg and mass m_2= 62.0 kg.

Table 3

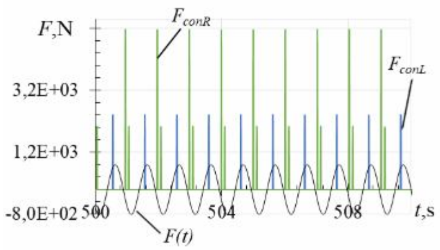
Attenuation of the maximum total energy of the primary structure coupled with the dampers of masses m_2= 39.7 kg and m_2= 62.0 kg, depending on the exciting force frequency

ω rad·s^{-1}	**6.2**	**6.3**	**6.4**	**6.5**	**6.7**	**7.0**
$\boldsymbol{E_{1max}}$wane **%** for m_2=39.7 kg	13.7	44.1	50.8	49.0	41.2	31.4
$\boldsymbol{E_{1max}}$wane **%** for m_2=62.0 kg	20.1	54.2	61.9	61.4	53.9	42.5

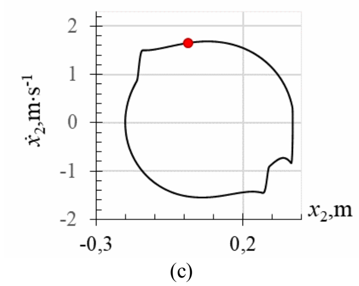
As Table 2 shows, a vibro-impact system with damper of mass m_2=39.7 kg performs the regular periodic motion even without direct damper impacts on the primary structure at higher exciting force frequencies. Fig. 5 shows the movement characteristics of the T,1,2 regime at the same exciting force frequency ω = 6.2 rad·s^{-1}.



(a)



(b)



(c)

Fig.5. The characteristics of the periodic regime T,1,2 for the system with a damper of mass m_2= 39.7 kg with 7 optimized parameters at the exciting force frequency ω = 6.2 rad·s^{-1}. (a) The damper relative displacements. (b) The contact forces during damper impacts against the primary structure in blue and against an obstacle in green. (c) The phase trajectory with Poincaré map in red

The relative damper displacements in Fig. 5 (a) show one direct impact between the damper and the primary structure at (x_2-x_1) = 0 and two damper impacts on an obstacle at (x_2-x_1) = C = 0.1224 m. The contact forces graph in Fig. 5(b) shows one force per cycle when the damper directly impacts the primary structure in blue and two forces per cycle when the damper hits an obstacle in green. The exciting force is also shown in this graph. Fig. 5 (c) presents the phase trajectory with Poincaré map in red for a damper. This is a closed curve with one point of the Poincaré map, which is typical for T-periodic movement.

Table 2 shows that the vibro-impact system with a heavy damper of a mass m_2=62.0 kg also performs regular periodic motion, but at some exciting force frequencies the movement deviates somewhat from the periodic one and becomesa motion that can be called “chatter”. It is interesting to compare the types of its characteristics with both periodic and chaotic regimes. Fig. 6 demonstrates the characteristics of regime “chatter” for a system with the heavy damper of mass m_2=62.0 kg with 7 optimized parameters at the same exciting force frequency ω = 6.2 rad·s^{-1}.

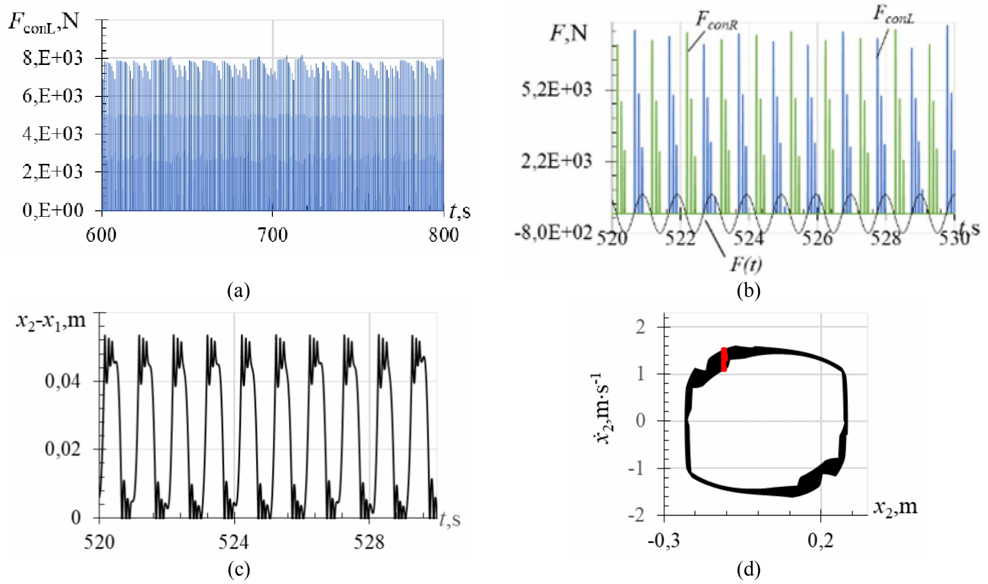


Fig. 6.The characteristics of "chatter" mode for the system with the damper of mass m_2= 62.0 kg with 7 optimized parameters at the exciting force frequency ω = 6.2 rad·s^{-1}. (a) Contact forces during damper impacts on the primary structure directly. (b) Contact forces when the damper hits both the primary structure directly in blue and an obstacle in green over a narrower exciting force range. (c) The relative damper displacements. (d) Phase trajectories with Poincaré map in red for the damper

4. Conclusions

The research results described above allow us to draw the following conclusions.

• It is necessary to optimize the VI NES parameters in order to select such its design that will ensure the most efficient operation.

• Different optimization procedures give different results, so the obtained values of the VI NES parameters must be carefully analyzed in order to choice those values that will ensure the most efficient operation.

• It is necessary to pay attention to the synergistic effect of multiple parameters, so as many parameters as possible should be optimized.

• A well-designed damper in combination with the primary structure shifts the resonant peak of its energy to the left towards low frequencies and reduces it. A damper mitigates the primary structure energy well at the exciting force frequencies above the resonant one. On the contrary, it not only does not attenuate its energy, but increases it at low exciting force frequencies lower than the resonant one.

• The dynamics of a vibro-impact system "the primary structure – VI NES" is rich and complex. We see many different irregular regimes. However, the complex dynamics practically does not change the damper efficiency, because the oscillatory amplitudes and velocities of the heavy primary structure change little under these regimes, but the contact forces change strongly.

• Direct impacts between the primary structure and the damper occur in almost all regimes. This means that the single-sided VI NES operates like a double-sided one, in which the primary structure itself is the second barrier.

Summarizing, we want to highlight the limitations of VI NES. They do not attenuate the primary structure energy at the exciting force frequencies below the resonant one. The choice of its optimal design is complex and ambiguous.

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ВИБІР ОПТИМАЛЬНОГО ДИЗАЙНУ ДЛЯ ВІБРО-УДАРНОГО НЕЛІНІЙНОГО ПОГЛИНАЧА ЕНЕРГІЇ

Ефективність віброударного нелінійного поглинача енергії (vibro-impactnonlinearenergysink - VINES), тобто вібро-ударного демпфера, значною мірою визначається його конструкцією. Оптимальний дизайн демпфера можна підібрати за допомогою процедур оптимізації. Однак результат їхньої роботи неоднозначний, їхні різні варіанти показують різні значення оптимальних параметрів демпфера. Ретельний аналіз отриманих значень параметрів дозволяє підібрати оптимальний варіант за певним критерієм. Проводячи цей аналіз, ми спостерігали багато цікавих явищ, а саме синергетичний ефект багатьох параметрів, багату комплексну динаміку VI NES, наявність прямих ударів між демпфером і головним тілом, залежність повної енергії від параметрів збуджуючої сили. Аналіз також дозволяє сформулювати обмеження VI NES. Всі ці проблеми відображені в цій статті.

Ключові слова: віброударний, первинна структура, демпфер, нелінійний поглинач енергії, синергетичний ефект.

Lizunov P.P., Pogorelova O.S., Postnikova T.G.

SELECTION OF THE OPTIMAL DESIGN FOR A VIBRO-IMPACT NONLINEAR ENERGY SINK

The efficiency of a vibro-impact nonlinear energy sink (VI NES), that is, a vibro-impact damper, is largely determined by its design. The optimal damper design can be found through optimization procedures. However, the result of their work is ambiguous, their various options show different values of the optimal damper parameters. A thorough analysis of the obtained parameters values allow you to select the best option according to a certain criterion. While carrying out this analysis, we observe many interesting phenomena, namely, the synergistic effect of multiple parameters, rich complex dynamics of the VI NES, the presence of direct impacts between the damper and the main body, the dependence of the total energy on the exciting force parameters.The analysis also allows us to formulate the limitations of the VI NES. All these problems are reflected in this article.

Keywords: vibro-impact, primary structure, damper, nonlinear energy sink, synergistic effect.

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Ефективність віброударного нелінійного поглинача енергії (vibro-impact nonlinear energy sink -VI NES), тобто вібро-ударного демпфера, значною мірою визначається його конструкцією. Оптимальний дизайн демпфера можна підібрати за допомогою процедур оптимізації. Однак результат їхньої роботи неоднозначний, їхні різні варіанти показують різні значення оптимальних параметрів демпфера. Ретельний аналіз отриманих значень параметрів дозволяє підібрати оптимальний варіант за певним критерієм. Проводячи цей аналіз, ми спостерігали багато цікавих явищ, а саме синергетичний ефект багатьох параметрів, багату комплексну динаміку VI NES, наявність прямих ударів між демпфером і головним тілом, залежність повної енергії від параметрів збуджуючої сили. Аналіз також дозволяє сформулювати обмеження VI NES. Всі ці проблеми відображені в цій статті.

Табл. 3. Рис. 6. Бібліогр. 22 назв.

UDC 539.3
Lizunov P.P., Pogorelova O.S., Postnikova T.G. **Selection of the optimal design for a vibro-impact nonlinear energy sink**//Strength of Materials and Theory of Structures: Scientific-and-technical collected articles. – K.: KNUBA. 2023. – Issue111. – P. 13-24.

The efficiency of a vibro-impact nonlinear energy sink (VI NES), that is, a vibro-impact damper, is largely determined by its design. The optimal damper design can be found through optimization procedures. However, the result of their work is ambiguous, their various options show different values of the optimal damper parameters. A thorough analysis of the obtained parameters values allow you to select the best option according to a certain criterion. While carrying out this analysis, we observe many interesting phenomena, namely, the synergistic effect of multiple parameters, rich complex dynamics of the VI NES, the presence of direct impacts between the damper and the main body, the dependence of the total energy on the exciting force parameters. The analysis also allows us to formulate the limitations of the VI NES. All these problems are reflected in this article.

Tabl. 3. Fig. 6. Ref. 22.

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