и $Y_{1г} = c_n + (1 - c_n)\lambda$. С учетом этого выражения (3) и (10) принимают вид, соответственно,

$$K_{1м} = \frac{d - b_n\lambda}{1-\lambda}\sigma\sqrt{\pi l};$$
$$K_{1г} = \frac{0,407\left[c_n + (1-c_n)\lambda\right]}{(1-\lambda)^2}\sqrt{\frac{1-\lambda}{\lambda}}\sigma\sqrt{\pi l}. \quad (12)$$

При λ = 0,2 КЭ-значение $K_{1м} = 0,990\sigma\sqrt{\pi l}$, а для смежного значения λ = 0,3 $K_{1м} = K_{1г} = 1,054\sigma\sqrt{\pi l}$. С учетом этого и (12) получаем: b_n = 0,86; c_n = 0,77; d = 0,98. Таким образом, при поперечном изгибе образца с мелкой или глубокой трещиной для КИН получены решения, соответственно,

$$K_{1м} = \frac{0,98 - 0,86\lambda}{1-\lambda}\sigma\sqrt{\pi l};$$
$$K_{1г} = \frac{0,407(0,77 + 0,23\lambda)}{(1-\lambda)^2}\sqrt{\frac{1-\lambda}{\lambda}}\sigma\sqrt{\pi l}. \quad (13)$$

Оценка погрешностей Δ решений (13) по сравнению с КЭ-данными показала следующие результаты: λ → 0 Δ = –2 %; λ = 0,2 Δ = +2 %; λ = 0,4 Δ = –0,3 %; λ = 0,6 Δ = –0,1 %; λ = 0,8 Δ = +0,2 %; λ → 1 Δ → 0.

При исследовании критических КИН проводятся испытания на трещиностойкость в условиях чистого или поперечного изгиба партии образцов с предварительно выращенными трещинами исследуемых размеров l, определяются в соответствии с нормативными требованиями критические нагрузки P_c, вычисляются критические напряжения σ_c, а затем по предлагаемым соотношениям (5), (11) или (13) определяются характеристики трещиностойкости $K_{1мc}$ или $K_{1гc}$.

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Ehab S. M. M. Soliman, Mechatronics Department, Egyptian Russian University
Tamer A. El-Sayed, Mechanical Design Department, Faculty of Engineering, Mataria-Helwan University
Soheir A. R. Naga, Mechanical Design Department, Faculty of Engineering, Mataria-Helwan University

A NEW APPROACH TO DESIGN A COMPOSITE MATERIAL FOR LIGHT MONO LEAF SPRING USING FEA

Introduction

The weight reduction of automotive parts while keeping their strength is one of the main challenges in the vehicles industry. As leaf springs are widely used in many vehicles, it is important to investigate how to reduce their weight. The composite materials are widely used to reduce the leaf spring weight without any reduction in load carrying capacity [1]. The stored elastic strain energy in a leaf spring is a main factor for a comfortable suspension system. Since the strain energy in the spring is inversely proportional to the density and the young's modulus of the material, and the composite materials possess low density and modulus of elasticity, hence, they are suitable for a leaf spring [2, 3, 4]. The steel leaf spring was replaced by composite leaf spring whose thickness varied linearly from the middle to the end and the springs have been analyzed using FEA [5] and were designed and tested [6]. The composite leaf springs showed a reduction in weight 80 % to 88 % less than the steel spring. Designed simulation using finite element package (LUSAS) was put forward to calculate numerically elliptic woven roving glass/epoxy composite spring constants [7]. The investigation verified that composites can be utilized for vehicle suspension and meet the requirements, together with substantial weight saving. A comparative study between composite and steel leaf springs has been done [4] and inferred that composite materials reduce the weight of the leaf spring without any reduction on load carrying capacity and stiffness. A single leaf spring with variable thickness and variable width while keeping a constant cross sectional area of different composite materials was modeled and analyzed [8]. It was concluded that, the composite leaf spring has lower stresses for the same deflection, and the total leaf spring weight was nearly 78 % lower than the steel leaf spring. It has been showed [9] that the carbon composite leaf springs have high stiffness and offer substantial weight savings as compared to conventional steel leaf spring.

In the present work, proposed composite mono leaf spring models are designed to have the same stiffness and load capacity of a real selected conventional steel mono leaf spring. Due to the complexity of the adopted models it was found that the FEA is the most convenient for the analysis.

Finite element analysis (FEA)

FEA is adopting ANSYS 14 software. The composite mono spring model in the FEA can be reduced to a half model because the leaf spring is symmetric. The principal axes considered in the analysis are as follows: x axis along the leaf spring length and represents the longitudinal direction of the unidirectional fiber lamina, z and y axes represent the transverse in-plane and through the thickness directions respectively. A 3-D model of the leaf spring is used for the FEA analysis [10] and the loading conditions are assumed to be static. The element chosen is SOLID186, which is a layered version of the 20-node structural solid element.

An experimental validation of the used FEA has been carried out first for a steel flat specimen (250 mm × 40 mm × 6.5 mm) with Young's modulus = 200 GPa and Poisson's ratio 0.32.

Experimental validation has been also done on a unidirectional E-glass polyester composite flat specimen (250 mm × 25 mm × 6.5 mm). The burn out test [11] is carried out to determine the fiber volume fraction V_f in the composite material of the specimen and is found to be 0.329. The mechanical properties of a unidirectional E-glass polyester composite specimen (V_f = 0.329) [12] are given in Table 1.

Table 1. **Mechanical properties of the unidirectional E-glass/polyester composite, V_f = 0.329, flat specimen [12]**

Properties	E_X (MPa)	E_Y (MPa)	E_Z (MPa)	G_{XY} (MPa)	G_{YZ} (MPa)	G_{XZ} (MPa)
	24378	4704	4704	1731	1991	1731

The steel and composite flat specimens were experimentally tested under bending with span 140 mm using a WDW-300 computer controlled electronic universal testing machine. The obtained deflections for steel and composite specimens are presented in Figures 1 and 2 respectively and compared with that resulted from the FEA and theoretical analysis [13, 14] which show good agreement.

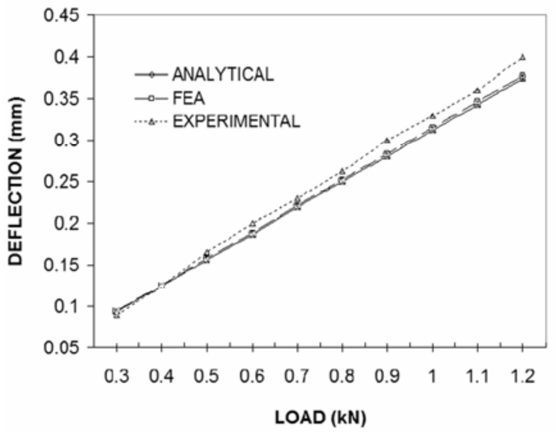

Fig. 1. Validation of FEM and experimental results, steel flat specimen beam

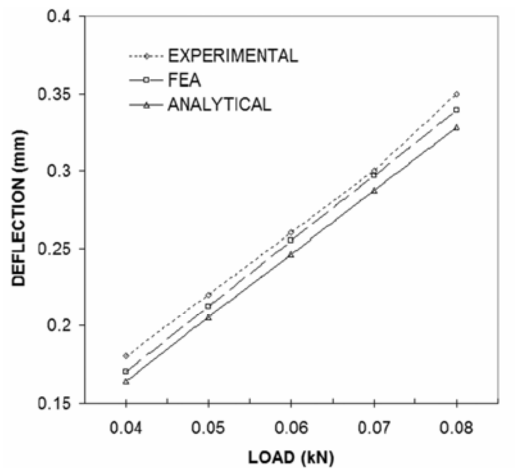

Fig. 2. Validation of FEM and experimental results, composite flat specimen beam

Design parameters of a steel parabolic mono leaf spring currently used in a passenger vehicle are given in Table 2 as a reference for analysis and comparison [15].

Table 2. **Design parameters of steel parabolic mono leaf spring [15]**

Parameter	Maximum load on spring (N)	Total length (eye to eye) (mm)	Free camber (at no load conditions) mm	Width of leaf spring (mm)	Thickness of leaf spring (mm)
Parabolic mono leaf spring	2500	965	68	50	10

The material properties of the steel [16], the epoxy matrix [17] and the unidirectional E-glass/epoxy composite (UEC) for (0.3, 0.6) fiber volume fractions [12, 18] for the springs are listed in Table 3.

First the FEA is used to determine the sections of the spring which are safe to withstand the applied bending load without fiber reinforcement i.e. the applied stress is less than the epoxy strength. The length and the width of the composite mono parabolic leaf spring are taken as that of the conventional steel mono leaf spring, Table 2. The thickness varied in order to obtain the same deflection (i.e. same stiffness) and same load carrying capacity as that of the steel mono parabolic leaf spring under the same applied load .The investigated mono leaf springs in the present work have been partially reinforced by fibers in areas where the stress exceeds the allowable strength of the epoxy matrix given in Table 3 and with volume fraction depending on the strength of the UEC given in Table 3. Figure 3 represents the bending stress across the composite leaf spring cross section for a one layer parabolic mono composite leaf spring with uniformly distributed fibers UEC, V_f = 0.6 with thickness 16.6 mm, design 0 in Figure 5. From the hatched area in Figure 3 it can be seen that the stress in the thickness from 3.8 mm to 10.2 mm is less than the allowable strength of the epoxy matrix (50.7 MPa – 128.5 MPa), Table 3, in tension and compression respectively. Hence it is safe to design a spring using epoxy only without fibers in this region. Similarly it is safe to use a UEC with V_f = 0.3 in the whole section, in which the stress is less than the allowable stress of composite V_f = 0.3, (538 MPa – 316 MPa), Table 3.

Table 3. **Mechanical properties of steel [16], epoxy [17] and unidirectional E-glass/epoxy composite [12, 18]**

Material	E_X MPa	E_Y MPa	E_Z MPa	PR_{XY}	PR_{YZ}	PR_{XZ}	G_{XY} MPa	G_{YZ} MPa	G_{XZ} MPa	Tensile strength MPa	Compressive strength MPa	Factor of safety	Allowable Tensile strength MPa	Allowable Compressive strength MPa
Steel	210000	–	–	0.266	–	–	–	–	–	1158	1158	1.4	827	827
Epoxy	4600	–	–	0.4	–	–	–	–	–	71	180	1.4	50.7	128.5
E-glass unidirectional / epoxy composite Vf = 0.3	25420	7915	7915	0.36	0.34	0.36	2781	3198	2781	754	442	1.4	538	316
E-glass unidirectional / epoxy composite Vf = 0.6	46240	14966	14966	0.31	0.29	0.31	5690	6544	5690	1438	815	1.4	1027	582

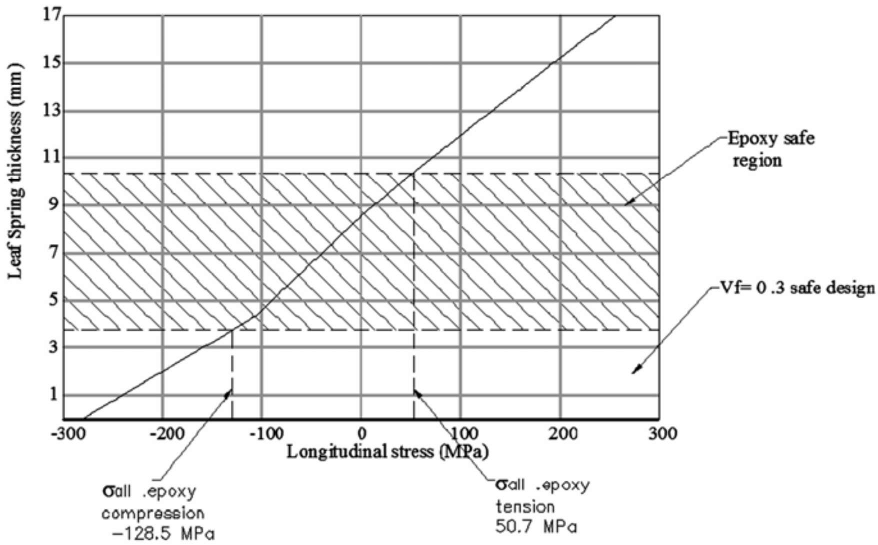

Fig. 3. Maximum bending stress distribution across the thickness of UEC (V_f = 0.6) curved leaf spring

Figure 4 represents the bending stress across the half symmetric span of the leaf spring. Figure 4, *a* represents the maximum compressive stress along the leaf spring (at the lower face of leaf spring) and Figure 4, *b* represents the tensile stress along the leaf spring (at the upper face of leaf spring). Figure 4 shows that it is safe to design springs using epoxy only at a minimum distance of about 380 mm from the center of the spring (point of loading) where the maximum bending stress is less than the allowable strength of the epoxy matrix (50.7 MPa – 128.5 MPa).

In the current work, five models 0, I, II, III, IV are designed as shown in Figure 5, *a–e* for half leaf spring. Design model 0 is one layer UEC Vf = 0.6 of thickness 16.6 mm [19], Figure 5, *a*. The rest of models are for layered composite leaf spring of thickness 17 mm. Figures 5, *b–e*. Design I is a UEC parabolic mono leaf spring with thickness 17 mm consists of three continuous fibers layers (VF0.6/epoxy/VF0.6). Design II is a UEC parabolic mono leaf spring with thickness 17 mm consists of five continuous fibers layers (VF0.6/VF0.3/epoxy/VF0.3/VF0.6). Design III is a UEC parabolic mono leaf spring with thickness 17 mm consists of three discontinuous fibers layers (VF0.6/epoxy/VF 0.6). Design IV is a UEC parabolic mono leaf spring with thickness 17 mm consists of five discontinuous fibers layers (VF0.6/VF0.3/epoxy/VF0.3/VF0.6). The models 0, I, II, III and IV are designed to achieve the same deflection (stiffness) of the conventional steel leaf spring. Since the currently investigated models are complicated and the analytical solution is difficult for such cases, therefore, FEA analysis is used to analyze the proposed designs. Several trials should be done to obtain the thickness of each layer in assigned design, to achieve the required stiffness.

Results and discussion

The results of parabolic composite (UEC) models 0, I, II, III, IV and steel mono leaf springs are listed and compared in Table 4.

From the FEA results the bending stresses in layers thickness for parabolic mono leaf springs (design I, II, III, IV) are less than the allowable strength of the composite material in Table 3 and the estimated maximum shear stress (1.5F/A) due to the maximum load 2500 N which is equal 4.412 MPa, is less than the allowable shear stress of the epoxy middle layer (35 MPa/1.4 = = 25 MPa). The safety of each layer in each design has been checked and Table 5 is given as an example, for design IV.

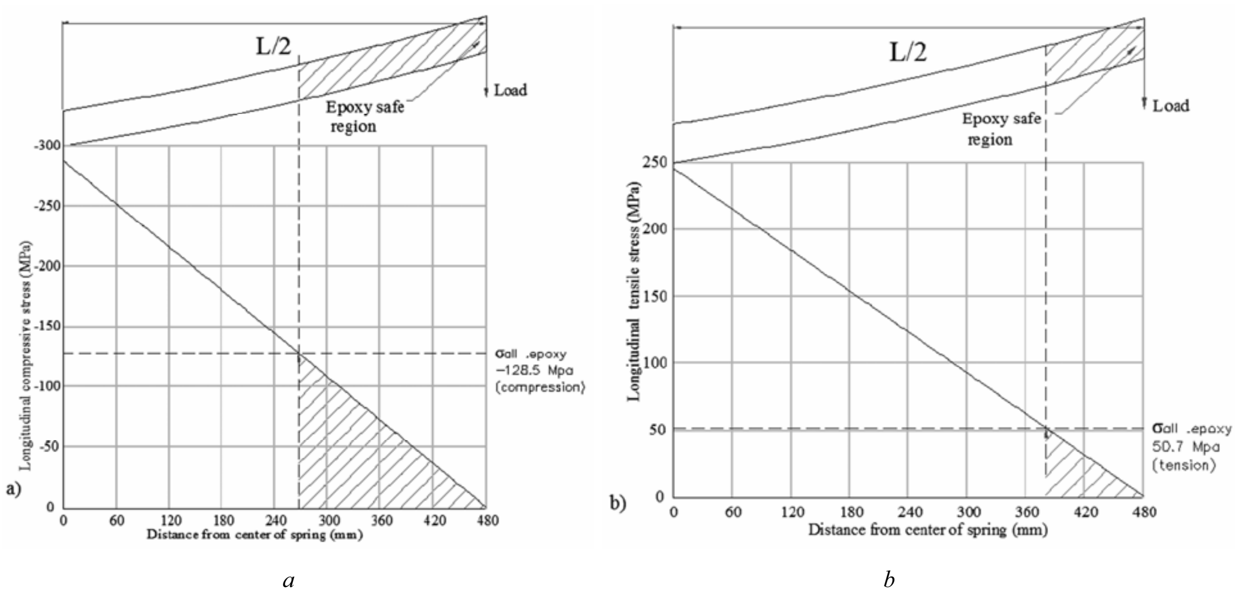

Fig. 4. Bending stress versus the distance measured from the center of the composite leaf spring with $V_f = 0.6$:
a – compression; *b* – tension

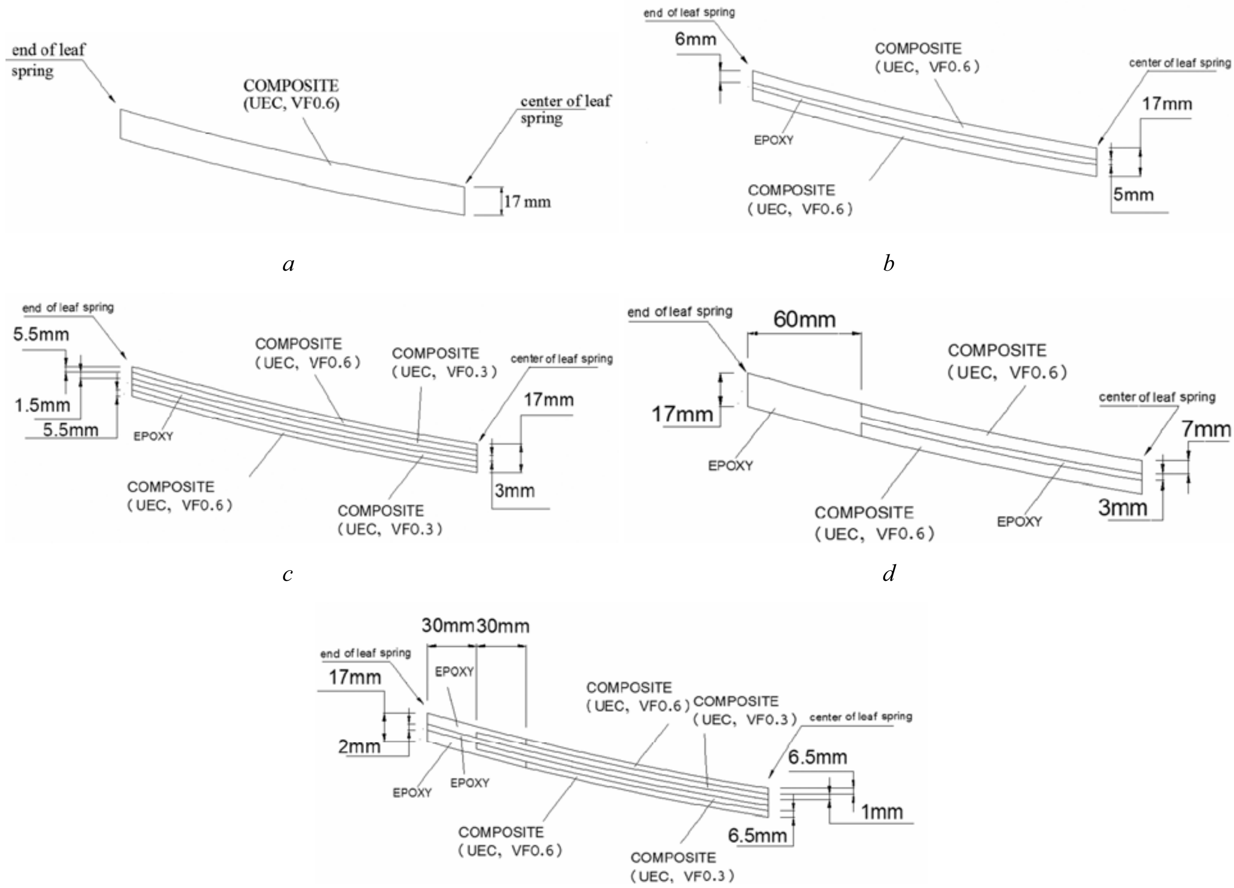

Fig. 5. The proposed composite mono leaf spring models: *a* – Design 0; *b* – Design I; *c* – Design II; *d* – Design III; *e* – Design IV

Table 4. **FEA results of steel & standard design, I, II, III, IV parabolic mono leaf springs**

	Steel	Composite with one layer (UEC, $Vf = 0.6$)	Composite with continuous layers		Composite with discontinuous layers	Composite with discontinuous layers
		Design 0	Design I	Design II	Design III	Design IV
Load (N)	2500	2500	2500	2500	2500	2500
Thickness (mm)	10	16.6	17	17	17	17
Thickness (mm) of layers	10 mm Steel	16.6 mm VF 0.6	6 mm VF 0.6+ 5 mm EPOXY+ 6 mm VF 0.6	5.5 mm VF 0.6+ 1.5 mm VF 0.3+ 3 mm EPOXY+ 1.5 mm VF 0.3+ 5.5 mm VF 0.6	7 mm VF 0.6+ 3 mm EPOXY+ 7 mm VF 0.6	6.5 mm VF 0.6+ 1 mm VF 0.3+ 2 mm EPOXY+ 1 mm VF 0.3+ 6.5 mm VF 0.6
Deflection (FEA) (mm)	51.8	51.8863	51.8506	51.8011	51.8298	51.8788
Von Mises stress (MPa)	748.4	259.9	274.016	276.594	261.994	264.926
Max.Bending stress (MPa)	+783.4 –788.7	+259.7 –265.2	+266.374 –301.455	+265.698 –303.756	+260.347 –289.859	+260.513 –292.536
Stiffness (N/ mm)	48.2	48.2	48.2	48.2	48.2	48.2
Strain energy (J)	32.4	32.4	32.4	32.4	32.4	32.4
Weight (kg)	3.79	1.63	1.47	1.49	1.48	1.46
% Weight reduction over steel leaf spring	–	57	61.2	60.7	60.9	61.5

Table 5. **Results of FEM bending stress in layers thickness in parabolic mono leaf spring (Design IV)**

Design IV		
Layers	Bending stress (MPa), ANSYS	Allowable strength (MPa) = Strength / Factor of safety
Layer 0-6.5 mm UEC, VF 0.6	–292.54 → –28.748	Tensile = 1027 Compressive = –582
Layer 6.5-7.5 mm UEC, VF 0.3	–28.748 → –5.8934	Tensile = 538 Compressive = –316
Layer 7.5-9.5 mm EPOXY	–5.8934 → 5.8527	Tensile = 50.7 Compressive = –128.5
Layer 9.5-10.5 mm UEC,VF 0.3	5.8527 → 30.168	Tensile = 538 Compressive = –316
Layer 10.5-17 mm UEC,VF 0.6	30.168 → 260.51	Tensile = 1027 Compressive = –582

Conclusion

The FEA technique used herein can be applied on complicated design mono leaf spring of composite material to satisfy specific requirements of stiffness and load carrying capacity.

The proposed composite leaf springs, of varying volume fraction with continuous and discontinuous fibers, achieved a weight reduction from 57 % to 61.5 % compared with a steel mono leaf spring of same stiffness and load capacity.

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Р. А. Мусарский, доктор технических наук, профессор, Нижегородский государственный технический университет имени Р. Е. Алексеева

АНАЛИЗ И ОПТИМИЗАЦИЯ ХАРАКТЕРИСТИК СОПРОТИВЛЕНИЯ ГИДРАВЛИЧЕСКИХ АМОРТИЗАТОРОВ *

В настоящее время применяются амортизаторные стенды, позволяющие осуществлять над амортизатором как периодическое, так и импульсное перемещение. В результате стендового испытания получают характеристику в виде гистерезисной петли (рис. 1) [1].

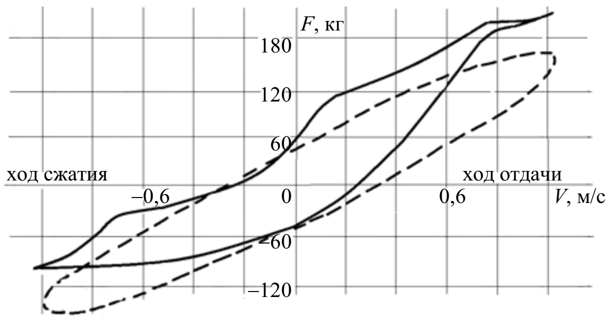

Рис. 1. Зависимость силы сопротивления гидравлического амортизатора легкового автомобиля среднего класса от скорости перемещения его элементов, полученная на стенде

Из рис. 1 видно, что характеристика сопротивления нелинейная, гистерезисная. Пунктиром обозначена аппроксимация петли гистерезиса, если имитировать силу амортизатора в виде суммы сил двух линейных элементов: только упругого и только демпфирующего (модель Фойгта – Кельвина). Более точно характеристику сопротивления гидравлического амортизатора можно представить в виде следующей аналитической аппроксимации:

$$F\left(x,\dot{x},\Delta t^{\mathrm{o}}\right)=\eta\left(\Delta t^{\mathrm{o}}\right)\left[F_{\text{д}}\left(\dot{x}\right)+F_{\text{у}}\left(x\right)\right]. \qquad (1)$$

Здесь x и $\dot{x}$ – относительное перемещение и скорость перемещения элементов амортизатора; Δt^{o} – разность температуры поверхности амортизатора и окружающей среды; $F_{\text{д}}\left(\dot{x}\right)$ – демпфирующая составляющая силы сопротивления амортизатора; $F_{\text{у}}\left(x\right)$ – упругая составляющая силы сопротивления амортизатора; $\eta\left(\Delta t^{\mathrm{o}}\right)$ – тепловой коэффициент, учитывающий падение силы сопротивления при повышении вязкости амортизаторной жидкости при ее нагревании.

Ниже приведены рис. 2, 3 и 4, иллюстрирующие отдельно каждую из этих характеристик.

Заметим, что обычно исследователи не принимают во внимание сложную гистерезисную характеристику сопротивления гидравлических амортизаторов и ограничиваются рассмотрением только демпфирующей характеристики. Более того, большинство исследований рассматривают линейную демпфирующую характеристику и ограничиваются коэффициентом демпфирования амортизатора.

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