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COIL SPRINGS IN PASSENGER CARS – GENERAL THEORETICAL PRINCIPLES AND STRUCTURAL REQUIREMENTS

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Summary

The main role of the coil springs, as any kind of elastic elements is shock absorption caused by the movement of the so called unsprung masses (wheels, brakes, suspension components) on the uneven ground. This is to ensure not only driving comfort, but also protect the car from damage. The article presents the principles of strength calculations for coil springs used in the suspension of contemporary cars. Modern technological and structural solutions in contemporary cars are reflected also in the construction and manufacture of coil springs. A proper construction and the selection of parameters influence the strength properties, the weight, durability and reliability as well as the selection of an appropriate production method. An improper preparation of Finite Element Method calculation models consequently leads to wrong results. It is particularly difficult to interpret the results and to find an error if we do not have a comparative calculation base (such as results of fatigue tests, analytical strength calculations). Coil springs with linear and progressive characteristics were taken into consideration. In article was conducted a comprehensive calculation of these springs (analytically and FEM using) in order to show the differences and problems in the construction process. A proper selection of springs, that is also taking strength parameters into consideration, has a fundamental effect on correct functioning of the suspension of motor vehicles.

Keywords: coil springs in passenger cars, construction, calculation methods

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1. Introduction

Elastic suspension members transmit the forces caused by road surface irregularities to the vehicle frame, but they undergo deformations in the process, accumulating energy which is then released in the spring expansion phase. In this way, every deflection of the suspension results in a whole series of vehicle body's vertical vibrations which gradually cease because of the friction occurring in all reciprocally moving members [1, 2, 3, 8, 9]. However, the attenuating action of the friction is too slow to maintain the stability of vehicle motion and a permanent contact of its wheels with the road surface, particularly during greater driving speeds and more uneven road surfaces when the said vibrations may be additionally amplified by the resonance. Rapid and frequent vertical vibrations as well as longitudinal and lateral swaying of the vehicle body cause that the bodies of passengers undergo unpleasant and harmful accelerations, and the wheels, periodically losing contact with the road surface, transmit the vehicle driving, braking and steering forces less effectively.

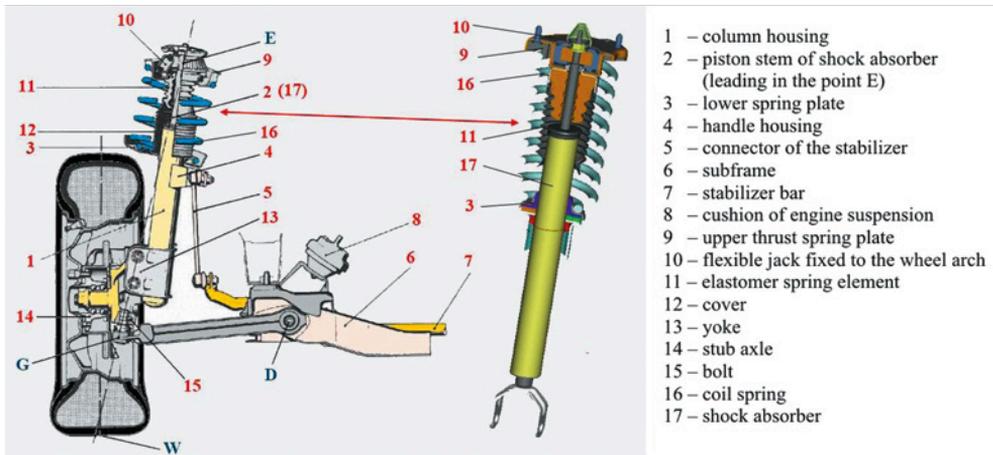


Fig. 1. Front suspension – McPherson strut [3, 5, 6, 10]

This is why shock absorbers the function of which is to more radically reduce the amplitude of cyclic motions of elastic members are used in car suspension systems [1, 2, 3, 8, 9].

A commonly used front suspension solution in cars is the McPherson strut suspension (Fig. 1). This suspension is a combination of a shock absorber and a spring which is attached in such a way that they constitute a fixed element. A shock absorber fulfils a damping function in this suspension as well as constitutes a guide member consisting in connecting the upper part of steering knuckles with a vehicle member (inner wheel arch) in a way enabling the shock absorber to turn [1, 2, 3, 8, 9]. Advantages of the MacPherson strut suspension:

- a compact and simple structure not taking up much space which may be used for the boot space or the engine compartment (Fig. 1),
- a lower weight in relation to the double wishbone suspension,
- cheaper to manufacture and kinematically simple,
- a large suspension stroke.

Disadvantages:

- the transmission of noise and vibration coming from the steering system to the vehicle body (e.g. due to unbalance of wheels),
- an unfavourable friction occurring in the socket of the shock absorber piston rod (caused by the rotation about its own axis),
- the need to use a piston rod with a large diameter (about 20 mm) in the shock absorber.

Ensure proper operation of McPherson struts require precise selection of coil springs (Fig. 2) together with shock absorbers. The mounting of springs (fastening with upper and lower resistive plate) (Fig. 3) is also an important part of the McPherson strut. The improper construction of fasteners most often leads to premature failure, fatigue fracture (Fig. 15). The proper design of the spring, determines the characteristic of springs (linear or progressive) (Figs. 7 and 8) referring to the required stiffness of the vehicle manufacturer does not automatically guarantee obtaining proper operation of the suspension. The requirements and design intents are often very difficult to obtain in terms of production. Therefore, already in the design phase it should take into account the production process (Fig. 4). In the article the authors describe with the example of two springs the linear (Figs. 5 and 6a, c) and progressive characteristics (Figs. 5 and 6b, d) showing the theoretical and practical guidance to assist in the design of coil springs.

2. Coil springs

Shown in Figs. 2 and 3 types of coil springs, formation of their ends depending on the choice of fixing plates are commonly used in modern motor vehicles. The design parameters such as the type – the type of springs, stiffness, and method attachment, building this coil springs in the McPherson column struts are usually imposed by car manufacturers. The design process should consider the requirements of car manufacturers, so that it does not have influence on the decrease in the required fatigue strength.

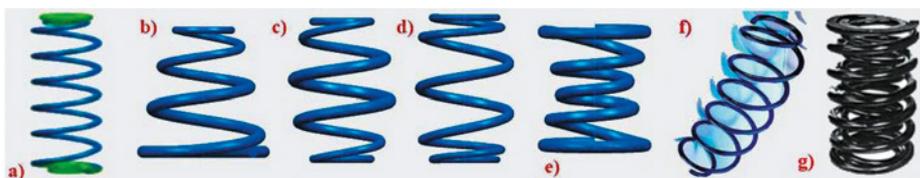


Fig. 2. The coil springs used in passenger cars [1, 2, 3]

a) linear cylindrical spring
b) progressive conical spring

c) progressive double truncated cone spring
d) barrel spring

e) hourglass spring
f) C - spring / banana spring
g) gradual layered spring

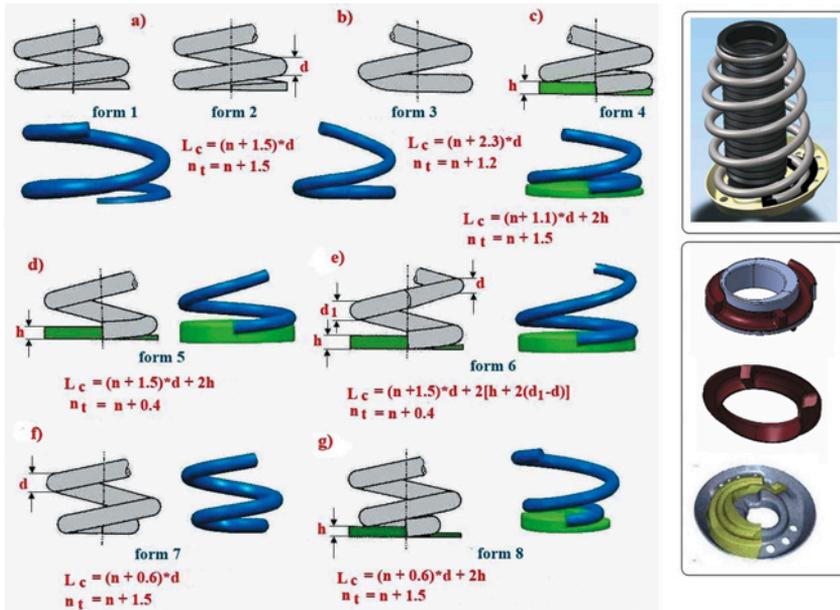


Fig. 3. End shapes of coil springs [1, 2, 3, 5, 9], upper and lower resistive plate

- a) grinding or forged and build
- b) build planar
- c) build to gradient of spring plate
- d) build to approx. 0.2 windings
- e) bar thickening and build to approx. 0.2 windings
- f) build planar
- g) build to gradient of spring plate

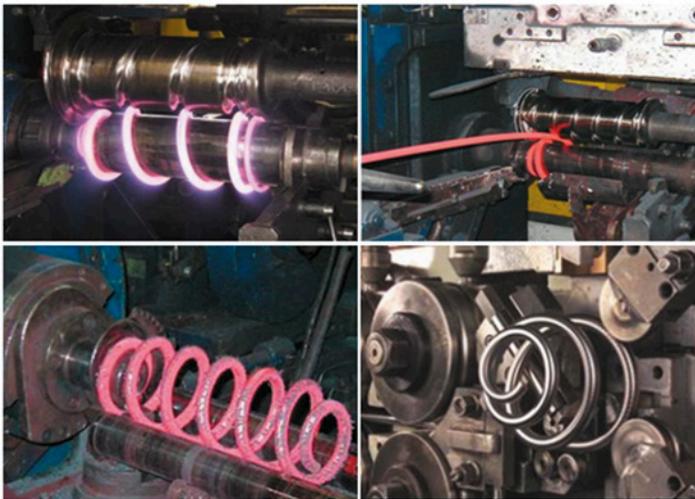


Fig. 4. The hot and cold winding of coil springs and shaping of ends (pigtail) (source TKF&S)

2.1. Formulary for coil springs

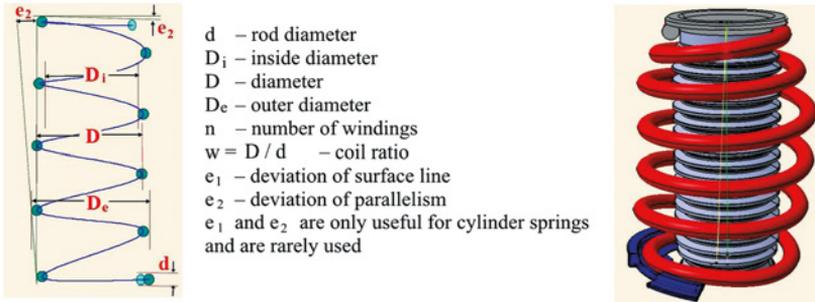


Fig. 5. The geometric parameters of automotive cylindrical coil springs [3, 4, 11]

The specified formulas are used for an approximate calculation of coil springs; all formulas are derived for linear cylindrical springs which are not or only to a limited extent suitable for springs with a different shape or for springs with a progressive characteristic [1, 2, 3, 4, 7, 9, 10] (Fig. 5).

Spring deflection:

$$s = \frac{8}{G} * \frac{D^3 * n * F}{d^4} \quad (1)$$

Spring force:

$$F = \frac{G}{8} * \frac{d^4 * s}{D^3 * n} \quad (2)$$

where:

G – the modulus of shear or modulus of rigidity [MPa]

Shear stress ideally:

$$\tau = \frac{8}{\pi} * \frac{D * F}{d^3} \quad (3)$$

Spring rate:

$$R = \frac{F * G * d^4}{8 * s * D^3 * n} \quad (4)$$

Winding ratio:

$$w = D / d \quad (5)$$

Bar diameter:

$$d = \sqrt[3]{\frac{8}{\pi} * \frac{D * F}{\tau}} \quad (6)$$

Correction factor:

$$k = \frac{4 * w - 1}{4 * w - 4} + \frac{0,615}{w} \quad (7)$$

Spring coils:

$$n = \frac{G}{8} * d^4 * \frac{s}{D^3 * F} \quad (8)$$

Corrected shear stress:

$$\tau_k = \tau * k \quad (9)$$

Spring work:

$$W = \frac{1}{2} F * s \quad (10)$$

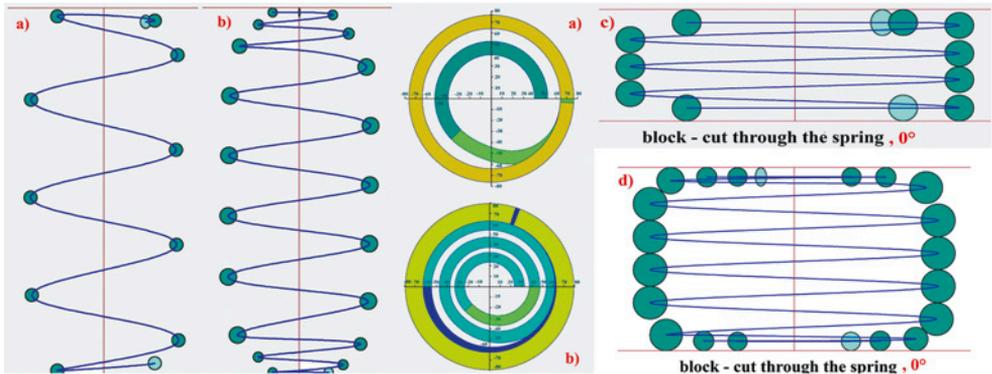


Fig. 6. a) Cylindrical coil spring with linear characteristic (Fig. 7), end shapes of coil springs [3, 4, 11]
 b) Barrel coil spring with progressive characteristic (Fig. 8), end shapes of coil springs [3, 4, 11]
 c) Maximum deflection (block) of cylindrical coil spring with linear characteristic [3, 4, 11]
 d) Maximum deflection (block) of barrel coil spring with progressive characteristic [3, 4, 11]

The rate of linear spring (Fig. 5, 6a):

$$R = \Delta F / \Delta s \quad (11)$$

A natural frequency (single mass oscillator, ratio of coil spring $i_{spring} = 1$):

$$f = \frac{1}{2\pi} \sqrt{\frac{R}{m}} = \frac{1}{2\pi} \sqrt{\frac{g * R}{F}} \tag{12}$$

where: R – spring rate [N/mm] m – loading mass of coil spring [kg] $F = mg$ [N]

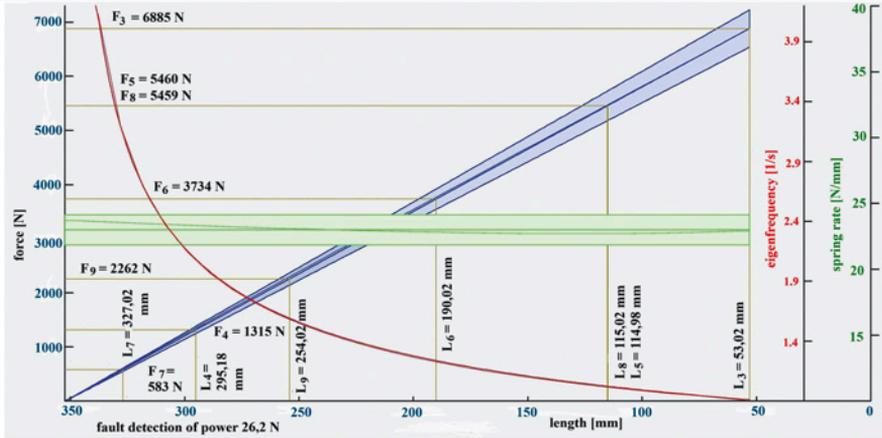


Fig. 7. Linear characteristic of cylindrical helical coil spring (load, stiffness, natural frequency) (own calculations)

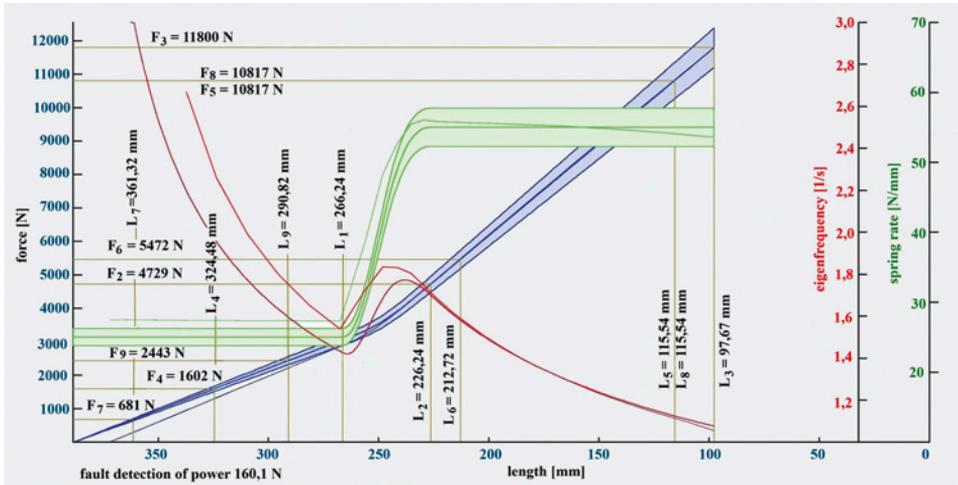


Fig. 8. Progressive characteristics of the coil spring type MiniBlock (load, stiffness, natural frequency) (own calculations)

The vertical movement of the suspension elements (z – direction) can be analyzed using the simplified model with one degree of freedom. This model consists of mass m equal to the weight of the car and spring, which stiffness R is equal to the entire stiffness of the suspension (with the rigidity of the tire). This model under the action of disposable, instantaneous force will vibrate with own frequency strictly defined with its dynamic parameters (m and R). Since a higher statistic spring force corresponds to a higher mass, the natural frequency decreases with increasing force or spring compression. This presentation allows evaluating e.g. a change in the natural frequency with a load (Fig. 7, 8) [1, 2, 3, 9, 12].

The spring rate of a progressive spring (Fig. 6b) is not a constant value and is calculated using:

$$R = dF/ds \quad (13)$$

In case of the force, rate and frequency characteristics, the theoretical (desired) characteristics and the characteristic values probably to be expected are presented. In case of a full spring compression, the reduction of the rate may be identified by the increase in the coil radius during spring compression. The progression (increase in the spring rate) is generated by blocking coil parts, namely by [1, 2, 3]:

- putting on the plate (typically: barrel springs),
- blocking by the coil contact (typically: progressive cylindrical springs).

2.2. Stress of the coil springs

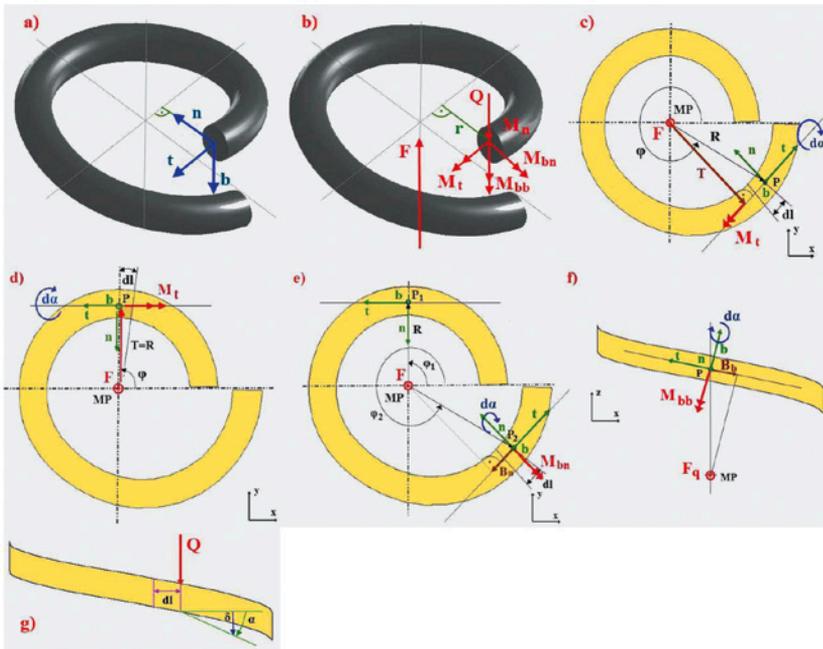


Fig. 9. Geometric assumptions to calculate the strength of coil springs [1, 2, 3, 4, 5, 7, 10]

Load on the springs – designations of direction

- a) The designation of direction of moments and stresses takes place by means of a rotating coordinate system with the directions (Fig. 9a) [1, 2, 3, 4, 5, 7, 9, 10]:
 - Tangential (t) in the direction of the bar,
 - Normal (n) perpendicular to the bar axis in the direction of the centre of curvature,
 - Binormal (b) perpendicular to t and n .
- b) In the examined spring cross-section, the moments and lateral forces occur in the designated directions (Fig. 9b).
- c) The main load results from the torsional moment M_t (Fig. 9b).
- d) The moments about the axes t and b , however, must also be taken into consideration in order to obtain an accurate calculation (Fig. 9b).

Torsion – twist of cross-sections

At the point MP the force F acts perpendicular to the plane of projection in the axis of the spring – shock absorber (McPherson spring strut Fig. 11a). The torsional moment M_t is generated by the lever arm T which is positioned perpendicular to the tangent t (Fig. 9c) [1, 2, 3, 4, 5, 7, 9, 10]:

$$M_t = T * F \quad (14)$$

The incremental angle of torsion is:

$$d\alpha = \frac{M_t}{G * I_p} dl \quad (15)$$

Torsion – cylindrical special case

At the point MP the force F acts perpendicular to the plane of projection. The torsional moment M_t is generated by the lever arm $T=R$ (Fig. 9d) which is positioned perpendicular to the tangent t [1, 2, 3, 4, 5, 7, 9, 10]:

$$M_t = R * F \quad (16)$$

The incremental angle of torsion – see (15).

Bending about the normal – bending causing spring deflection

At the point MP the force F acts perpendicular to the plane of projection with the lever arm B_n on the point P_2 , the lever arm is positioned perpendicular to the normal n , the bending moment M_{bn} is generated. In the circular area at the point P_1 the bending arm B_n has the zero length, and no bending occurs (Fig. 9e) [1, 2, 3, 4, 5, 7, 9, 10]:

$$M_{bn} = B_n * F \quad (17)$$

The incremental bending angle about the normal is:

$$d\alpha = \frac{M_{bn}}{E * I_x} dl \quad (18)$$

Bending about the binormal – radius changing bending

At the point MP the force F_q acts perpendicular to the plane of projection with the lever arm B_b on the point P , the lever arm is positioned perpendicular to the binormal b , and the bending moment M_{bb} is generated. If the coil part has no pitch, then B_b has the zero length, and no bending occurs (Fig. 9f) [1, 2, 3, 4, 5, 7, 9, 10]:

$$M_{bb} = B_b * F_q \quad (19)$$

The incremental bending angle about the binormal is:

$$d\alpha = \frac{M_{bb}}{E * I_x} dl \quad (20)$$

Shear – shift of cross-sections

A "displacement" of the cross-sections against one another is generated by the shear force $Q=F$, and the angle arising as a result can be calculated as (Fig. 9g) [1, 2, 3, 4, 5, 7, 9, 10]:

$$\alpha = \kappa * \frac{Q}{G * A} \quad (21)$$

The displacement δ is:

$$\delta = \alpha * dl \quad (22)$$

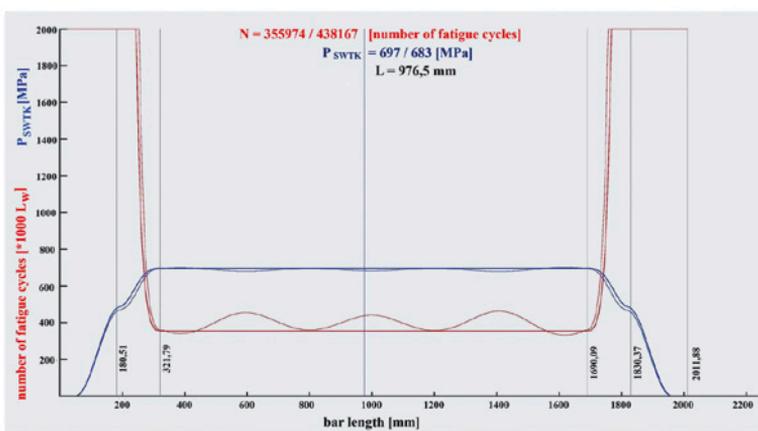


Fig. 10. Fatigue strength (number of fatigue cycles), parameter of destruction P_{SWTK} and stress distribution in coil springs with linear a) and progressive characteristic b) (own calculations)

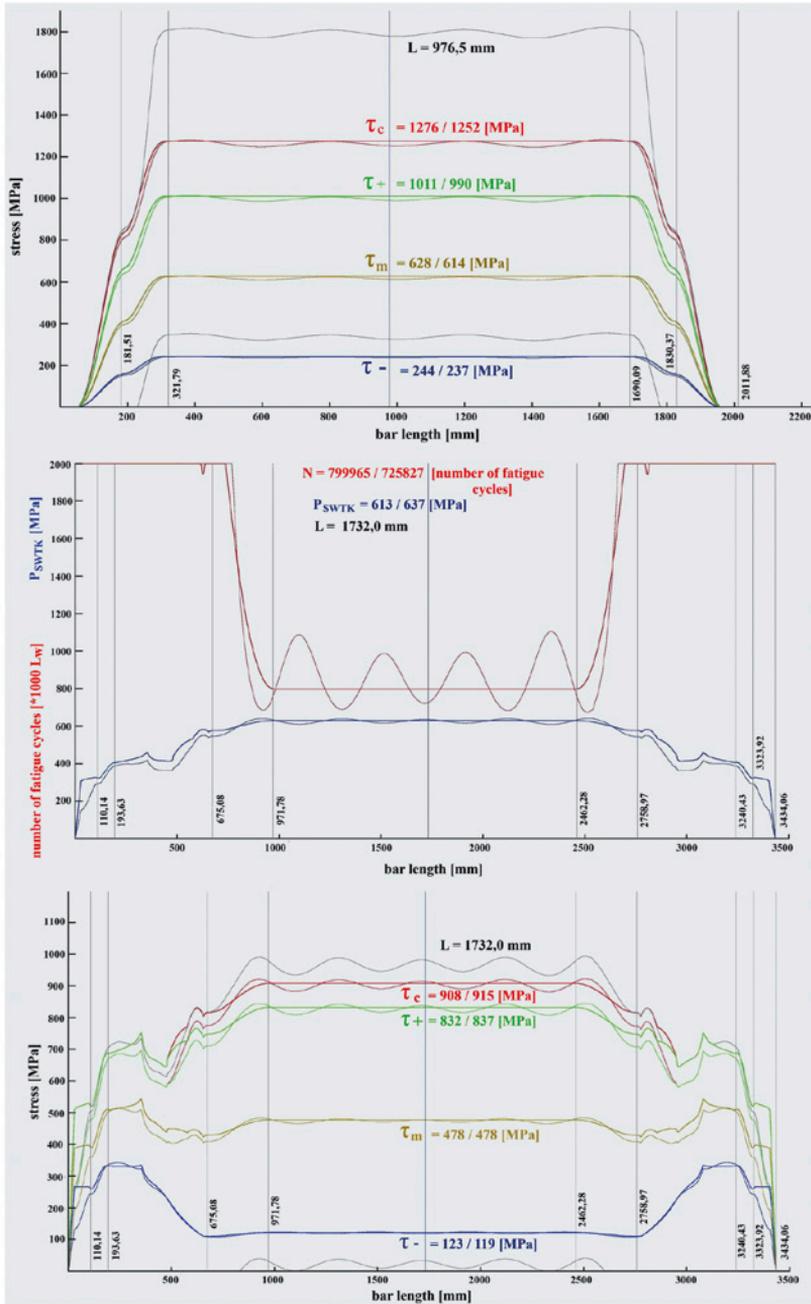


Fig. 10. Fatigue strength (number of fatigue cycles), parameter of destruction P_{SWTK} and stress distribution in coil springs with linear a) and progressive characteristic b) (own calculations)

2.3. Strength calculations of coil springs – general comments

The calculations of fatigue strength (number of fatigue cycles), parameter of destruction P_{SWT} and distribution of stresses (Fig. 10) in the coil springs with linear and progressive characteristic (Figs. 6, 7, 8) using the calculation program from ThyssenKrupp Federn & Stabilisatoren GmbH reflect the differences between these coil springs. The formulas (1 ÷ 22) are fully applicable to the calculation and determination of the characteristic of cylindrical coil springs (linear characteristic). They are an integral part of the above-mentioned calculation program. These formulas in the case of barrel coil springs (with progressive characteristic) are applicable to a limited extent. The calculation and determination of the characteristics of the barrel coil springs are more complex and comprehensive. These calculations are an important output base for the calculation using FEM (static and dynamic calculation). They can also, along with fatigue tests be used for validation and verification of computational of FEM models.

2.4. Spring and shock absorber system

The wheel vertical force F_A , the rocker arm force F_L and the body force F act on the spring and shock absorber system (Fig. 11a).

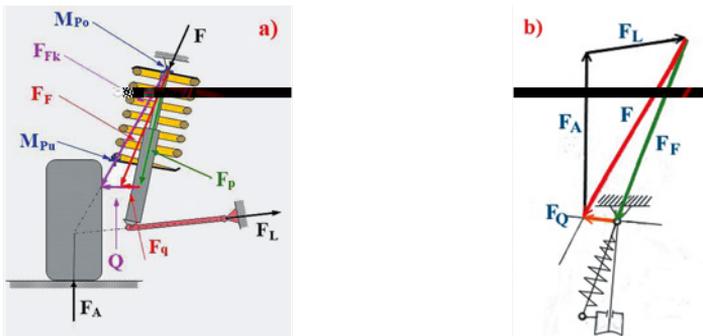


Fig. 11a and b The distribution of forces in the spring – shock absorber (McPherson spring strut) [5, 6, 9, 10]

The sum of these three forces is zero. Since the total moment must also be zero, the force F must be directed in such a way that its extension encounters the intersection of the extensions of both other forces. If the spring force F_F acts, then it can be split into portions F_p , parallel to the shock absorber and F_q . The difference between the required shear force Q and the shear force F_q absorbed by the spring is the shear force which the shock absorber must bear. If and only if the spring force F_{Fk} runs through the points M_{po} and M_{pu} with its line of action, the spring force is equal to the shear force required in the system, thus the shock absorber is free of shear force. The points M_{po} and M_{pu} are the force centres to be maintained by the spring (Fig. 11a, b and 12).

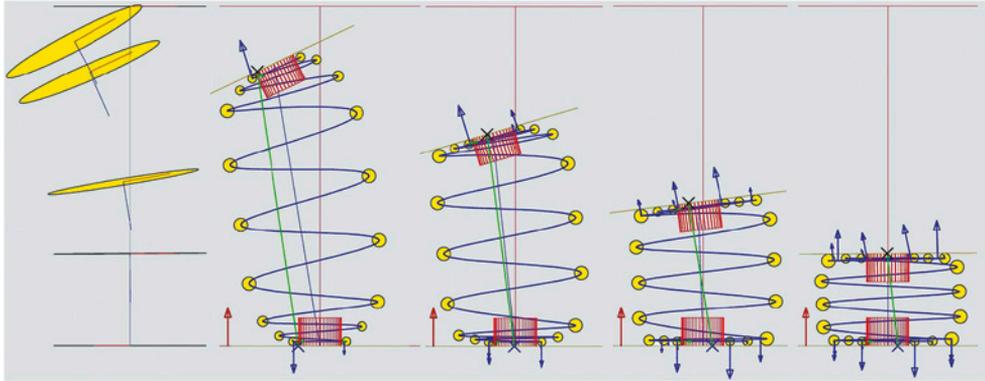


Fig. 12. The phases of deflection of coil springs in the system: coil spring - shock - absorber [3, 4, 5, 11]

2.5. Analytical formulas for spring

Table 1. Analytical formulas for spring with variable shapes, circular wire cross - section of diameter [3, 4, 5, 10]

	spring rate	mass
<i>A = const.</i> <i>D = const.</i>	$R = \frac{G\pi d^4}{8nD^3}$	$m = \frac{1}{4}\pi^2 n d^2 \rho D$
<i>A = const.</i> <i>D ≠ const.</i>	$R = \frac{G\pi d^4}{4 \int_0^{2\pi n} D^3(\varphi) d\varphi}$	$m = \frac{1}{8}\pi d^2 \rho \int_0^{2\pi n} D(\varphi) d\varphi$
<i>A ≠ const.</i> <i>D = const.</i>	$R = \frac{\pi G}{4D^3 \int_0^{2\pi n} d^{-4}(\varphi) d\varphi}$	$m = \frac{1}{8}\pi \rho D \int_0^{2\pi n} d^2(\varphi) d\varphi$
<i>A ≠ const.</i> <i>D ≠ const.</i>	$R = \frac{\pi G}{4 \int_0^{2\pi n} D^3(\varphi) d^{-4}(\varphi) d\varphi}$	$m = \frac{1}{8}\pi \rho \int_0^{2\pi n} d^2(\varphi) D(\varphi) d\varphi$

R - spring rate [N/mm]

n - number of active coils []

D - mean of coil diameter [mm]

d - Diameter of wire [mm]

G - shear modulus [MPa]

A - cross - sectional area [mm²]

ρ - mass density [kg/m³]

m - mass of the spring [kg]

φ - angle of winding of coil springs [°]

2.6 FEM calculations of coil springs

Commonly used analytical springs calculation programs are based on years of experience from manufacturers and are still an essential tool in the application design and strength calculations. However, there is an increasing use of FEM particular in the calculation of the buckling of coil springs and their fatigue strength (Fig. 13, 14, 15).

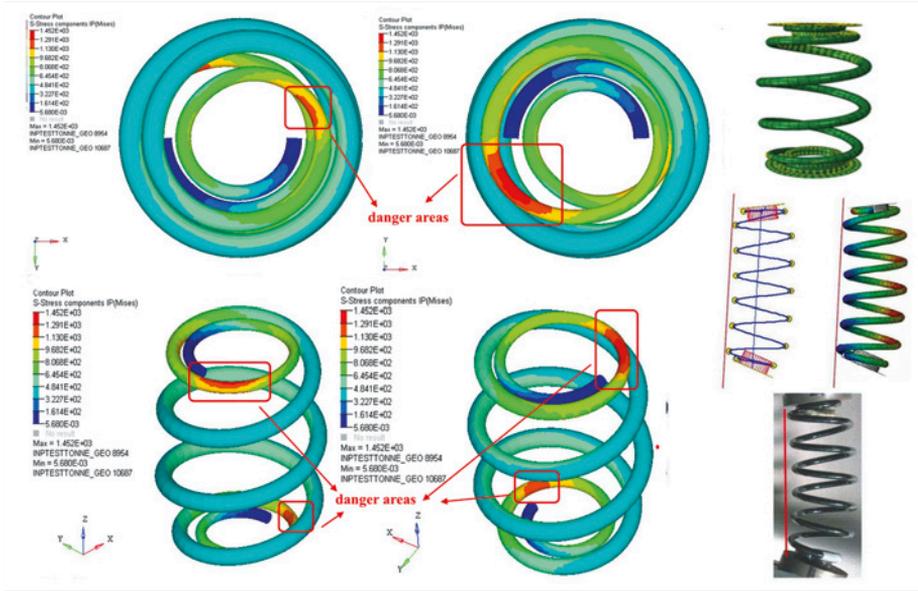


Fig. 13. Strength calculations of loaded barrel springs using FEA (critical areas) (own calculations)

The article presents a calculation example of the barrel spring with the parameters set out in the table below. Static strength calculations were performed using Finite Element Method (HyperWorks with Radioss) (Fig. 13) and calculate the fatigue strength (nCode DesignLife 8) (Fig. 14 and 15). The calculations were verified by fatigue tests. The results of fatigue tests coincide with the results of numerical simulation (Table 2, item 7 and Fig. 15b).

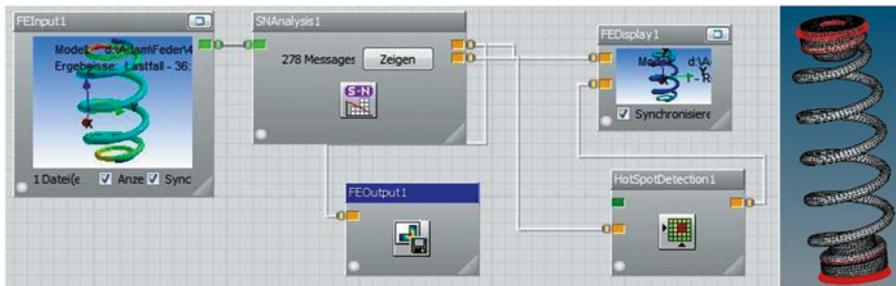


Fig. 14. Algorithm of fatigue strength calculations of loaded barrel springs using HBM nCode8 software

Table 2. Computing data and results of fatigue test or simulation of barrel spring

1	kind of springs		barrel spring
2	kind of production		hot or cold winding
	heat treatment – oil hardening [°C] and middle tempering [°C]		830 / 230
3	steel grade		spring steel 55Cr3
	ultimate tensile strength	R_m [MPa]	1400 – 1700
	yield strength	R_e [MPa]	1200
4	winding temperature	[°C]	930
5	required rate of coil spring	[N/mm]	53
6	max. equivalent stress	[MPa]	1450,0 (1477,0)
7	fatigue strength (number of fatigue cycles) []		min. 3850 max. 600000 (min. 3871 max. 620700)
8	conducted the fatigue tests (fatigue machine type Franke 2)		yes
9	conducted the fatigue simulations (nCode) (Fig. 14, 15b)		yes
10	conducted the static calculation of equivalent stress with FEM (Fig. 13, 15a)		yes
11	conducted the verification and validation of computational models		yes

The compression coil springs can bend by critical laterally force (see Fig. 16a). The limit for the buckling force can be calculated for the elastic buckling. The buckling force depends on the slenderness ratio of the spring and the nature of their storage. The slenderness ratio λ is by the ratio L_0 to D defined. The Mounting is by the factor ν described (Fig. 16a). The limit curve of the buckling is illustrated in Fig.16b. The boundary curve separates the buckling area from the non – buckling safe area. For springs with a $\nu\lambda < 2.633$ there is no risk of buckling the entire load range [3, 5, 10].

$$\lambda = \frac{L_0}{D} \quad (23)$$

$$\frac{s_k}{L_0} = 0,808 \left[1 \pm \sqrt{1 - 6,935 \left(\frac{1}{\lambda\nu} \right)^2} \right] \quad (24)$$

where: s_k – spring deflection [mm]

L_0 – unstrained length of coil spring [mm]

D – average diameter of coil spring [mm]

ν – coefficient characterizing the way of the seating of coil springs (Fig. 16a) []

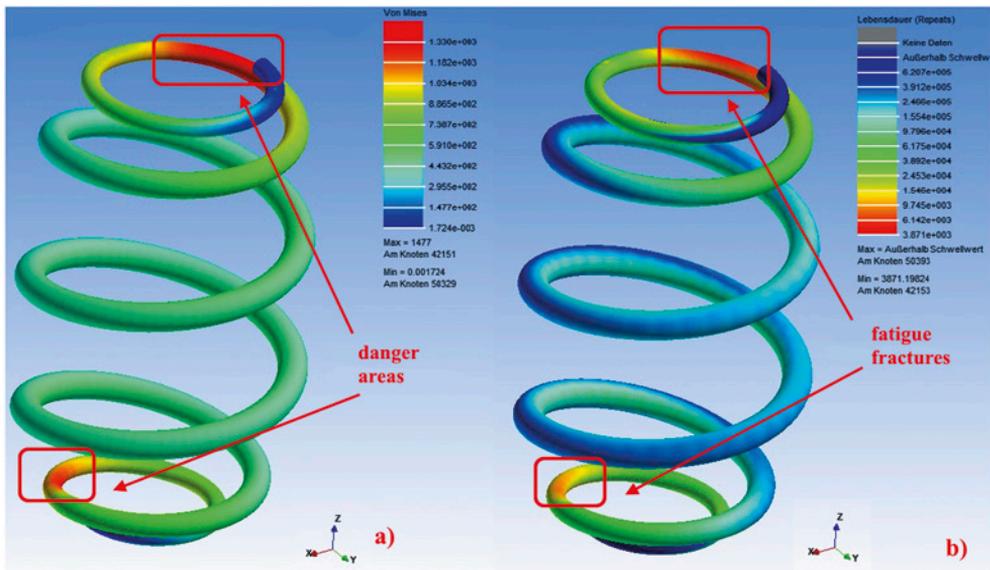


Fig. 15. Static strength calculations a) and fatigue strength of dynamically loaded coil springs b) (HBM nCode 8) (own calculations)

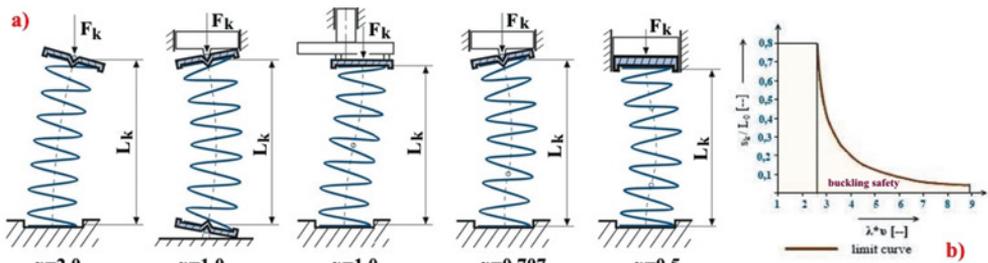


Fig. 16 Buckling of the coil spring, depending on the applied forces and the clamping of springs a) buckling characteristic of coil springs b) [5, 9, 10]

2.7. Formulation of the optimization problem

The increasing demands on coil springs, e.g. in terms of static and dynamic strength and the long - term behavior, the desire of customer, mass and space to save, require from springs manufacturers increasing efforts.

The spring dimensioning is the basis for choice of production process of the coil springs. Both the spring geometry and the material data are determined, to achieve a required force - displacement characteristic, as well as data concerning on the production of springs. The

dimensioning and manufacturing process in many cases are distinguished by an iterative sequence of actions. So it is not possible due to the mathematical basis to determine the geometry values of a coil spring in a single calculation run. The formulas 25 ÷ 33 concern the optimization calculations of the barrel springs (Table 2).

- compliance with certain prescribed characteristic (spring deflection – force – behavior),
- adherence to the defined spring force – line of action (lateral forces, moments),
- fulfillment of the installation conditions „packaging”,
- weight optimization under conditions of fatigue strength and resistance to environmental influences and corrosion.

Formulation of the optimization problem for non – cylindrical coil spring [3, 5, 7, 10, 11]

Design conditions:

$$\text{spring force} \equiv \text{Target force} \Rightarrow F_{max.} = F_{max.,target} \quad (25)$$

$$\text{stress} \leq \text{Permissible stress} \Rightarrow \tau_{id,max.} \leq \tau_{adm.} \quad (26)$$

$$\text{spring rate} \equiv \text{shall rate spring} \Rightarrow R = R_{shall} \quad (27)$$

Optimization goal (quality criterion):

$$\text{optimal mass of the coil spring} \Rightarrow m \rightarrow \min. \quad (28)$$

Solution of the optimization problem for non – cylindrical coil spring

$$\text{spring force} \equiv \text{Target force} \Rightarrow F_{max.} = F_{max.,target} \quad (29)$$

$$\text{stress} \leq \text{permissible stress} \Rightarrow \tau_{id,max.} \leq \tau_{adm.} \Rightarrow \tau_{id,max.} = \frac{M}{W_T} = \quad (30)$$

$$= \frac{8F_{max.} * D(\varphi)}{\pi d^3(\varphi)} \leq \tau_{adm.} \quad d(\varphi) = \left(\frac{8F_{max.} D(\varphi)}{\pi \tau_{adm.}} \right)^{1/3}$$

$$\text{spring rate} \equiv \text{shall rate spring} \Rightarrow R = R_{shall} \Rightarrow R_{shall} = \quad (31)$$

$$= 4\pi G \left(\frac{F_{max.}}{\pi \tau_{adm.}} \right)^{4/3} \left(\int_0^{2\pi n} D^{5/3}(\varphi) d\varphi \right)^{-1}$$

$$\text{optimal mass of the coil spring} \Rightarrow m \rightarrow \min. \Rightarrow m_{opt.} = \quad (32)$$

$$= \frac{1}{2} \pi \rho \left(\frac{F_{max.}}{\pi \tau_{adm.}} \right)^{4/3} \int_0^{2\pi n} D^{5/3}(\varphi) d\varphi$$

$$m_{opt.} * R_{shall} = \frac{2\rho G F_{max.}^2}{\tau_{adm.}^2} \quad (33)$$

In order to ensure proper functioning of the damping elements of suspension, the strength calculations must be properly performed. The theses included in the tables 3 and 4, are intended to avoid incorrect construction assumptions, which lead to erroneous strength calculation.

Table 3. The theses for proper selection of the coil springs

1	The more coils a spring has, the stronger it is! The spring rate is crucial here.	False!
2	If the spring is shorter, the car lies lower! The wire size and the spring type are decisive.	False!
3	Shock absorbers are responsible for the „suspension“ of the vehicle! The suspension, as the name also already implies, is provided by the spring. The shock absorbers reduce the occurring vibrations. Therefore, if new shock absorbers are mounted, the ride quality changes to a limited extent only.	False!
4	Coil springs are either intact or defective! Springs can break or a material fatigue occurs. The driver does not directly notice this gradual wear. We have the same problem here as in the case of a defective shock absorber.	False!
5	Old slacked springs are shorter than new springs! If dismantled, possibly no height difference can be found. The wear becomes apparent under stress only.	False!
6	Shock testers are shock absorber testers! Shock testers check all unsprung masses, thus the complete chassis.	False!

Table 4. The theses for proper selection of the coil springs

			spring rate
wire diameter	grows	→	grows
outer diameter	grows	→	decrease
number of windings	grows	→	decrease

3. Conclusions

The purpose of this article was to demonstrate the complexity of the problems associated with the design and strength calculations of coil springs being a very important element of suspended vehicles. Applied computational methods are an important part of the design process and facilitate the work of the engineer. The properly performed strength

calculations, taking into account the requirements of car manufacturers are not a guarantee for production of the optimal product in the manufacturing process. Therefore, the choice of the manufacturing process of coil springs is an important element in the construction process. The choice of the manufacturing process depends on the results of the calculation and the actual strength parameters of the pattern spring. This article is an aid and a guide in the design development.

The full text of the Article is available in Polish online on the website <http://archiwummotoryzacji.pl>.

Tekst artykułu w polskiej wersji językowej dostępny na stronie <http://archiwummotoryzacji.pl>.

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