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# AN ANALYSIS OF THE RAILWAY VEHICLE SUSPENSIONS

Eugen GHITA[1], Lucia VÎLCEANU[2]

[1] "Politehnica" University of Timisoara, Faculty of Mechanical Engineering, ROMANIA
[2] "Politehnica" University of Timisoara, Faculty of Engineering Hunedoara, ROMANIA

**ABSTRACT**:
The paper presents an analysis regarding the metallic and rubber springs as parts of the suspensions of the railway vehicles. The working life-time of the rolling equipments depends on the life-time of the suspension which is correlated with the design and the state of stress in springs. This is the reason why a design improvement and an optimization process of the technical characteristics of railway springs is always needed.
**KEYWORDS**:
suspension, railway vehicle, spring parameters , stiffness, compression, optimization

## 1. INTRODUCTION

The comfort and the ride quality of the running process impose the improvement of the vehicle suspension design. Both the primary suspension (between the axle and the bogie frame) and the secondary one (between the bogie frame and the coach) include spiral metallic and rubber springs of different shapes and sizes.

## 2. PARTICULAR SPRINGS AND MASSES OF RAILWAY TRANSPORT VEHICLES

The vehicle masses are divided into two categories function of the elastic properties of the springs, as follows:

- suspended masses: the vehicle coach masse $m_c$ , the bogie frame mass and the masses which are elastically interconnected, $m_{sb}$
- non- suspended masses: the axle boxes and the spring masses, $m_{ns}$ (figure 1).

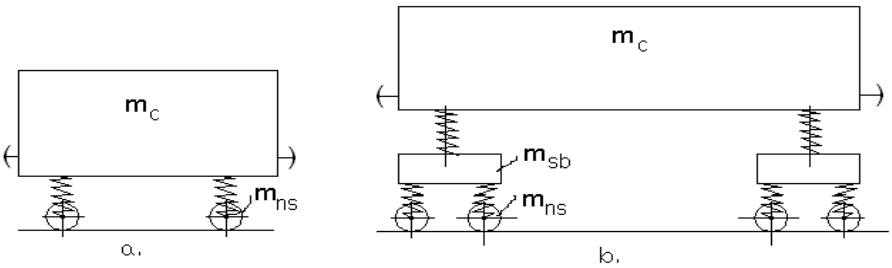


FIGURE 1. Single suspension (a) and double suspension (b) of railway vehicles

Both categories of masses interact through elastic forces inside the suspension springs:

$$\vec{F_e} = -k \cdot \vec{\Delta x} \tag{1}$$

where: k – the stiffness of the spring
Δx – the elongation from the equilibrium position

The acceleration of the suspended masses ($m_s$) represents a consequence of the supplementary elastic forces inside the springs and the mathematical expression is:

$$\vec{a}_s = \frac{\vec{F}_e}{m_s} \tag{2}$$

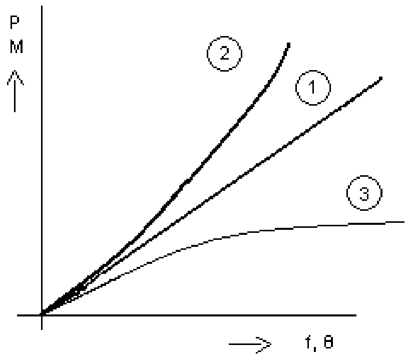


FIGURE 2. Elastic characteristic forms of particular railway springs

The role of the suspension is to reduce the dynamic load between wheel and rail and to increase the comfort and the safety during the running process. There is obviously compulsory to precise the parameters, the manufacturing technology and the running on conditions in order to design an efficient suspension. During the railway exploitation experience, the elastic elements of the suspension present always one of the following characteristics (figure 2), where:

1- linear characteristic - typical for metallic springs

2-non-linear progressive characteristic - typical for pneumatic and for the majority of rubber springs

3-non-linear regressive characteristic-typical for some rubber springs.

## 3. THE RAILWAY SPIRAL METALLIC SPRINGS

The spiral metallic springs are often used as primary and secondary suspensions of the railway vehicles, as well as the suspension of the electric traction engines, because of the following design advantages:

- a small volume
- a high coefficient of material volume utilization
- a high sensitivity because of the absence of friction
- a simple manufacturing process
- no need for lubrication.

An analysis of the primary and of the secondary suspension of the electric locomotive LE 4300 KW. is presented below. The main parameters of the locomotive are:

- the vertical allowed loading on axle: 182 kN
- the maximum speed:130 km/h
- the width:3000 mm.

After a classical calculus, it results to improve of the following spring parameters:

d - the diameter of a wire

D - the average wrapping up diameter

$L_b$ - the length of the spring when the wires are in contact

L - the length of the spring under nominal loading

$L_0$ - the free length of the spring without loading

c - the ratio between diameters ( $c = \frac{D}{d}$ )

k - the correction coefficient of stresses ( $k = \frac{c + 0{,}2}{c - 1}$ )

$f_b$ - the displacement under nominal loading

($f_b = 2\ f = 2x\frac{8D^3nP}{Gd^4}$, n-the number of active wires)

$\tau_{b\max}$ - maximum shearing stress in a wire

( $\tau = \frac{8DP}{\pi d^3}$ , $\tau_{b\max} = k\tau$ )

### 3.1. THE PRIMARY SUSPENSION

The vertical force on a primary suspension point is P=45.510 N and the stiffness of a spring is k=787,5 N/mm. After a classical strength calculation,the results are presented in table 1:

TABLE 1 . Parameters of the primary suspension springs

| d[mm] | D[mm] | $L_b$ [mm] | L[mm] | $L_0$ [mm] | c | k | $f_b$ [mm] | $\tau_{b\max}$ [MPa] |
|---|---|---|---|---|---|---|---|---|
| 40 | 221,2 | 160 | 218 | 276 | 5,53 | 1,26 | 116 | 100,93 |
| 41 | 228,6 | 164 | 222 | 280 | 5,57 | 1,26 | 116 | 96,86 |
| 42 | 236,1 | 168 | 226 | 284 | 5,62 | 1,26 | 116 | 93,06 |
| 43 | 243,6 | 172 | 230 | 288 | 5,66 | 1,26 | 116 | 89,47 |
| 44 | 251,2 | 176 | 234 | 292 | 5,70 | 1,25 | 116 | 85,43 |
| 45 | 258,8 | 180 | 238 | 296 | 5,75 | 1,25 | 116 | 82,28 |
| 46 | 266,5 | 184 | 242 | 300 | 5,79 | 1,25 | 116 | 79,32 |
| 47 | 274,3 | 188 | 246 | 304 | 5,83 | 1,25 | 116 | 76,54 |
| 48 | 282,1 | 192 | 250 | 308 | 5,87 | 1,25 | 116 | 73,90 |
| 49 | 301,1 | 196 | 254 | 312 | 5,92 | 1,25 | 116 | 70,63 |

### 3.2. THE SECONDARY SUSPENSION

The vertical force on a secondary suspension point is P=39.160 N and the rigidity of a spring is k=333,3 N/mm. After a classical strength calculation ,the results are presented in table 2:

TABLE 2. Parameters of the secondary suspension springs

| d [mm] | D [mm] | $L_b$ [mm] | L [mm] | $L_0$ [mm] | c | $f_b$ [mm] | $\tau_{b\,\max}$ [MPa] |
|---|---|---|---|---|---|---|---|
| 40 | 234,2 | 280 | 398 | 516 | 5,85 | 236 | 90,42 |
| 41 | 242,1 | 287 | 405 | 523 | 5,90 | 236 | 86,83 |
| 42 | 250,0 | 294 | 412 | 533 | 5,95 | 236 | 83,45 |
| 43 | 257,9 | 301 | 419 | 537 | 5,99 | 236 | 80,25 |
| 44 | 266,0 | 308 | 426 | 544 | 6,04 | 236 | 77,22 |
| 45 | 274,1 | 315 | 433 | 551 | 6,09 | 236 | 73,76 |
| 46 | 282,2 | 322 | 440 | 558 | 6,13 | 236 | 71,07 |
| 47 | 290,4 | 329 | 447 | 565 | 6,17 | 236 | 68,52 |
| 48 | 298,7 | 336 | 454 | 572 | 6,22 | 236 | 66,10 |
| 49 | 307,0 | 343 | 461 | 579 | 6,26 | 236 | 63,49 |

### 4. THE RAILWAY RUBBER SPRINGS

The Hooke's law $\sigma = \varepsilon \cdot E$ may be admitted for the rubber elements only for the small deformation domain $\varepsilon$ <10...12%. The temperature has an important influence on the elastic parameters of the rubber springs because the hardness decreases when increasing the temperature. The transformation of the energy because of the internal frictions and also the damping process are influenced by the behavior of the rubber under traction-compression loading as in figure 3:

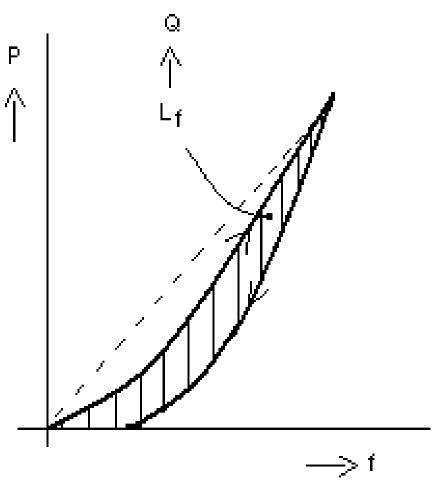


FIGURE 3. Stress-strain dependence under traction- compression loading of railway rubber springs

The rubber springs, as parts of the railway vehicles, must present higher ratio values of "natural rubber" than 60 %, but the traction loading has to be eliminated always when assembling.

A lot of advantages impose a large scale utilization of the rubber for the railway vehicle design ,[1]:

- a high amount of elastic energy stored (four times than the energy stored by the metallic springs)
- the considerable high internal frictions eliminate the third part of the mechanical energy.

Unfortunately, the rubber springs present some disadvantages as follows: the rubber became hard at low temperatures domain and it lose all the elastic properties at higher temperatures domain (> 240 °C).

The Poisson`s ratio of steel $\mu$ = 0,5 drive to the following dependence of Young`s modulus E and of the cross-section elastic modulus G function of the hardness $\chi$.

The behavior of the rubber springs under compression loading depends on the quality of the contact between the rubber and the metal on boundary. The free deformation is presented in figure 5:

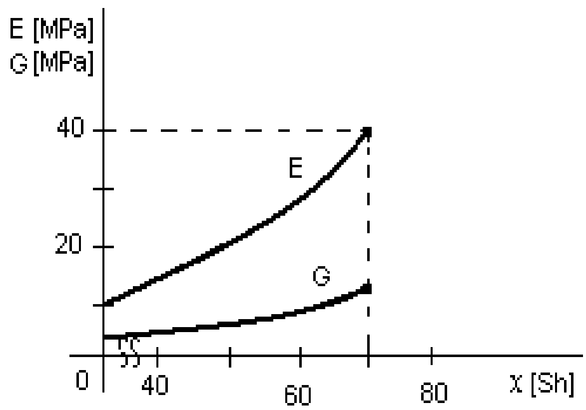


FIGURE 4. Elastic modulus-hardness dependence of railway rubber springs

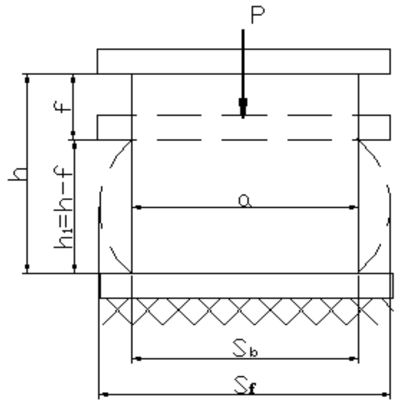


FIGURE 5. The free deformation of railway elastic rubber springs under compression loading

The Hooke`s law imposes:

$$\sigma = E \cdot \varepsilon = \frac{P}{S_f} \text{ and } \varepsilon = \frac{f}{h} \tag{3}$$

The rubber volume is constant, so:

$$S_b \cdot h = S_f (h - f) \tag{4}$$

It results,

$$P = 3G \cdot a \cdot b \frac{f}{h - f} \ . \tag{5}$$

As it was presented in figure 3, for the small deformation domain, f<<h, the spring characteristic is linear, but for the large deformation domain it becomes progressive,[1].
The stiffness under compression conditions is:

$$k_\sigma = \frac{dP}{df} = 3G\, ab \frac{h}{(h-f)^2} = \frac{3G \cdot a \cdot b}{h} \frac{1}{\left(1 - \frac{f}{h}\right)^2} \tag{6}$$

When the rubber spring is connected on a metallic support, a calculus longitudinal elastic module $E_c$ has to be taken into account. Generally, $E_c$>E and it depends on the shape coefficient $\lambda = \frac{S_b}{S_l}$, where: $S_b$ – the surface of basis, $S_l$ – the lateral surface. The calculus shape coefficient $\lambda_c$ depends on the theoretical one $\lambda$, as it is presented in figure 6:

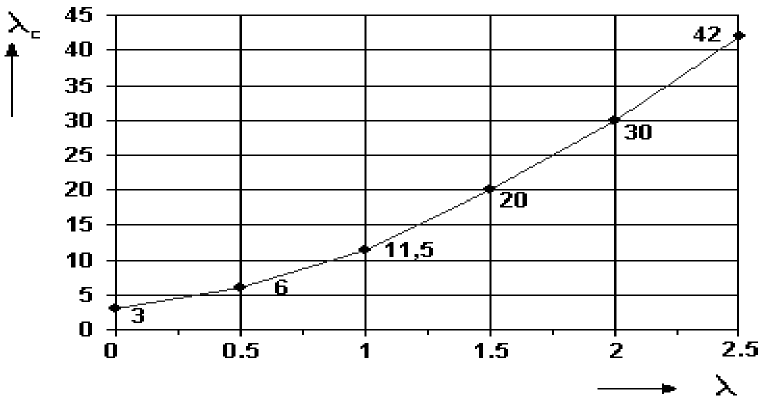


FIGURE 6. The graphical dependence between the theoretical and the calculus shape coefficient of railway rubber springs

For rubber connected on a metallic support it may be admitted the following dependence:

$$E_c = \lambda_c \cdot G \tag{7}$$

The compression force is:

$$P = \sigma \cdot S_f = \varepsilon \cdot E_c \cdot S_f \tag{8}$$

The same constant volume condition leads to:

$$P = \frac{f}{h} \cdot E_c \cdot S_b \cdot \frac{h}{h-f} = E_c \cdot S_b \cdot \frac{f}{h-f} \tag{9}$$

For a “brick” shape element, the following relations are valid :

$$\left.\begin{aligned} S_b &= a \cdot b \\ S_l &= 2 \cdot (a+b) \cdot h \end{aligned}\right\} \quad \text{şi} \quad \lambda = \frac{a \cdot b}{2 \cdot (a+b) \cdot h} \tag{10}$$

which conduce to the graphical solution for $\lambda_c$ and $E_c = \lambda_c \cdot G$.

It results:

$$P = \lambda_c \cdot G \cdot a \cdot b \cdot \frac{f}{h-f} = \frac{\lambda_c}{3} \cdot E \cdot a \cdot b \cdot \frac{f}{h-f} \tag{11}$$

The non-linear dependence imposes a derivative solution of the stiffness:

$$k_\sigma = \frac{dP}{df} = \lambda_c \cdot G \cdot a \cdot b \cdot \frac{h}{(h-f)^2} = \frac{\lambda_c}{3} \cdot E \cdot a \cdot b \cdot \frac{h}{(h-f)^2} \ . \tag{12}$$

## 5. CONCLUSIONS

According to tables 1 and 2 , the theoretical analysis of the primary and secondary suspensions of the railway vehicle leads to the following conclusions for the metallic springs:

- the level of the shearing stress decrease when the number of wires increase. It conduce to a technological alternative to decrease the level of stresses function of the designing free volume in accordance with the railway European regulation U.I.C. 505,[4].
- the different values of the spring length are in accordance with the number of wires and with the diameter of a wire
- the displacement of the ends of spring until the wires are in contact has a double value in comparison with the displacement under nominal loading conditions. For a small wire diameter the shearing stress level exceeds 100 MPa, when the contact between wires is present.

For the rubber elastic springs ,the stiffness under compression conditions increases when decreasing the length h of the element, the cross-section elastic modulus G ,the hardness $\chi$ ,the cross-section area $S_b$=a⋅b, and also the displacement of the ends f,usually resulting a progressive characteristic.

Nowdays,different combinations between metallic and rubber springs with dry friction dampers are usually used for the modern railway vehicles suspension desing solutions,[2],[3].


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