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STAND TESTING OF SPRINGS FOR DRUM BRAKE SYSTEMS

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Abstract

The paper concerns the study of changes in responses of spring for drum brake systems, due to fatigue cycles conducted at operational regimes of vehicles with respect to the number of kilometers. Three types of springs from two manufacturers were examined i.e. double cylindrical, single cylindrical, and conical helical. The springs were subjected to a durability test up to 1×10^6 loading cycles, covering 300-500 thousand kilometers traveled. Tensile test was used for collecting differences between results for the tested object in the as-receive state and after fatigue. Values of the Pearson correlation coefficient were used to indicate differences between tested objects before and after loading cycles. They show that the obtained results expressed a very strong correlation, which means that the elastic response of the springs during operation over a distance of 300-500 thousand kilometers did not change significantly. Taking into account the recommendations of brake system manufacturers regarding the replacement of brake drums after 150,000 and 50,000 kilometers, respectively, it can be concluded that brake springs are the most durable and reliable element of such a brake system.

Keywords: drums, springs, fatigue, reliability

1. INTRODUCTION

Many branches of industry: automotive [1, 2, 3], aerospace and defence [1, 2], rail [1], fitness and prosthetics [1] use elastic components for carrying cyclic loading at a linear relationship between force and displacement [4, 5, 6]. This means the effects due to the operational conditions are possibly being gradually reduced or totally eliminated. By this, the influence of loading conditions is very small for other subcomponents. It means the durability of the elements is not changed and they may be operated without an earlier service stage. For this case, springs of vehicle suspension can be indicated because they carry out all loadings resulting from a road [26, 27]. Nevertheless, the role of these components is not only related to suspension [2, 3] but also follows braking systems for the front and rear brakes of vehicles [21, 24]. This kind of component reflects behaviour of brake lining in relation to discs and drums [21]. It is expected to create a full contact region between the component for acceptable braking force values. If some differences occur, then the stopping distance increases or a wheel is locked, leading to a road accident [22, 23]. Nevertheless, worth emphasizing, the springs of the front braking system can be very easily inspected while the springs used in the rear braking system are covered and their checking is not so easy [20]. The role of drum brakes can be indicated by this cited text of: "Trouble-free operation necessitates a braking system that is in perfect mechanical order".

Engineering and scientific efforts for the assessment of rear drum braking systems are concentrated on practical [9, 10, 11, 12, 17], experimental [7, 16], and numerical [8, 16, 19] approaches. From the practical point of view, this is represented by photos and figures [9, 12, 17] in detail as well as reasons for an unacceptable braking system i.e. no nominal stiffness or fracture of lining return springs [9]. This component is also indicated for obligatory inspection and selected for replacement if a fracture occurs [11]. The same comment is also addressed if the spring brakes do not operatefully [12]. The difference in position after the operational process with respect to the nominal one is the reason for the replacement of the component.

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Experimentally, this kind of component as a sub-element of drum brakes (made of ADC12 alloy, LM30 alloy) can be inspected at various rotation speed values: 700 RPM, 1000 RPM, and 1300 RPM following their direct role for expanding the lining. This was connected to braking analysis at the loading value regime, i.e. from 5 N to 20 N [7]. Results have indicated that a braking force equal to 20 N creates the drum braking system very efficiently because the distance braking values were represented by 45 m, 20 m, and 5 m for the highest, middle, and lowest of the RPM values used, respectively. This also means the springs of the rear drum braking system should enable to hold the force with respect to safety. For this case, rear drum brakes are taken into account by a lot of research groups for analysis using Finite Element Methods [8]. This is focused on the application of new material (aluminium alloy LM6) instead of grey cast iron based on stress state components in 3D, fatigue, and failure. Contrary to the alloy, this approach has enabled the indication of crack zone occurrence in the grey cast iron due to the relationship between stress state conditions and mechanical parameters. The same positive results for the alloy drum were collected from analysis focused on fatigue. In this case, values of life cycles have reached the following levels: $0.95 \cdot 10^7$ (grey cast iron), $1.4 \cdot 10^7$ (the aluminium alloy). The durability of a drum brake determined in experimental and numerical approaches is considered with respect to differences in the values captured [16]. This confirms the number of loading cycles taken for both methods is possible to be obtained and as it was shown in the paper this can be below 15%, i.e. 8.45×10^5 for test and $1 \times 10^4 \div 1 \times 10^6$ for finite element analysis. This kind of component is also considered in a 3D model, applying three different materials: grey cast iron, aluminium alloy, and composite AlSiC [13]. For this case, the results have indicated that maximum values of stress and strain have occurred for AlSiC and aluminium alloy, respectively.

Another kind of approach to the drum brakes is expressed by analysis of temperature distribution [13, 19]. This is represented by temperature changes with maximum values ranging by 550 °C ÷ 600 °C versus time limited by 25 seconds. In the case of braking pads [15] efforts of research teams are also focused on stress and strain distribution considering two different materials such as aluminium alloy and carbon steel. This approach is covered by the 3D component model. These kinds of results have informed maximum values of stress that occurred for the carbon steel, while the maximum value of strain appeared for the aluminium alloy [19].

An analysis of the references indicates a gap for spring testing, while this kind of component type is strongly related to safety because it should create contact between the brake lining to the drum during braking and moreover if this component breaks then the breaking system may be locked. This means the component should not be omitted in research but taken into account with respect to its mechanical feature covering durability. Therefore, this problem is discussed in the paper in the form of data from static and fatigue tests.

In the approach to the problem considered the following stages were used:

a) proposition of method for testing the springs for a drum brake by means of an electro-dynamic testing machine at its grips and servo-valve actuator used in stand tests,
b) determining fatigue response of the springs for drum brakes at the number of cycles equal to 1×10^6 covering a long-term operation i.e. up to 500 000 kilometres traveling,
c) checking the influence of the cyclic loading on the static characteristic of the springs tested,
d) determination of differences between the same types of springs at known (W) and unknown (B) quality.

2. EXPERIMENTAL PROCEDURE AND TESTED OBJECT

The experimental procedure included as below:

1) static tests of various types of coil springs selected for drum braking systems at tensile loading,
2) durability test up to 1×10^6 cycles at sinusoidal function using the number of braking operations for 300-500 thousand kilometers traveled,
3) subsequent static tests the tested objects subjected to fatigue one,
4) comparison and analysis of the obtained results in terms changes in the performance properties of springs in drum brake systems during operation.

a) 
b) 
c) 

Fig. 1. Photos of coil springs: (a) double cylindrical, (b) single cylindrical, (c) conical helical

The object of the tests were double-cylindrical (Fig. 1a), single-cylindrical (Fig. 1b) and conical helical (Fig. 1c) coil springs, and from two manufacturers denoted Manufacturer No. 1 (W) and Manufacturer No. 2 (B) (Fig. 1).

Coding of individual types of springs is presented below:

- W - Manufacturer No. 1,
- B - Manufacturer No. 2,

- 1÷3 - Spring type (1- double-cylindrical coil spring, 2- single-cylindrical coil spring, 3- conical helical coil spring),
- 1÷4 - Spring number from the set.

All springs were measured collecting their outside diameter, wire diameter, total and operational lengths, as well as turns number, Tab. 1.

Table 1. Dimensions and geometrical features of spring from Manufactured No. 1

Spring type	Ø diameter of wire spring [mm]	Total length [mm]	The number of turns and their lengths [mm]		$Ø_t$ turns diameter [mm]	
			1	2	1	2
W.1	2.00	84.26	7 14.27	7 14.25	10.35	10.35
B.1	1.99	84.56	7 14.24	7 14.22	10.45	10.40
W.2	1.22	62.50	24 29.79		5.57	
B.2	1.28	62.10	18 23.83		6.62	
W.3	1.60	34.97	6 24.99		12.95	10.90
B.3	1.59	35.36	6 23.93		13.23	11.17

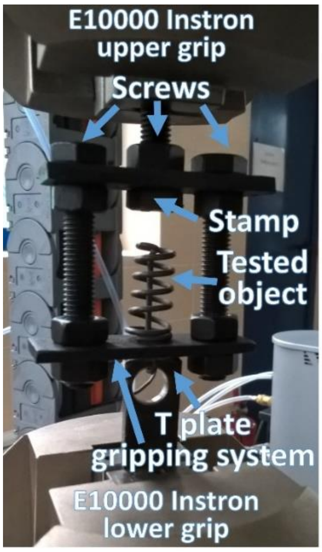

Fig. 2. Conical helical spring in the gripping system used in the E10000 Electropuls INSTRON testing machine before static test

The tests were carried out using an E10000 Electropuls INSTRON testing machine (Fig. 2) and ±10kN Saginomiya servo-hydraulic actuator at room temperature applying displacement signal in a form of monotonic and sinusoidal cyclic functions, respectively. Worth noticing, the Instron testing machine is controlled using a digital controller and is recommended for small specimens because it covers piezo-electric components and its loading capacity does not reach 10 kN. The servo-motor was also operated by an IST (Instron Structural Testing) controller and Saginomiya power pack. Changes in values of displacement and force were collected to obtain the force - elongation/shortening relationship as the final result. Such tests were carried out for springs in the as-received state and after fatigue testing.

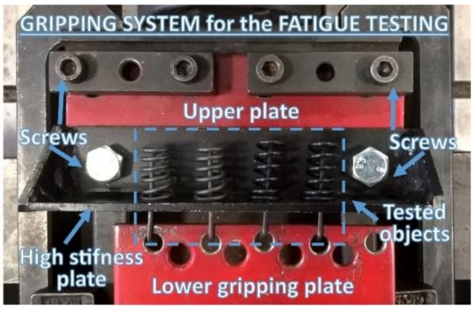

Fig. 3. Manufacturer No. 1 and Manufacturer No. 2 coil springs in the mounting system before the fatigue test

Table 2. Parameters of the fatigue test

Spring type	Minimum (H_{min}) and maximum (H_{max}) values of springs's high and amplitude of tension-compression cycles		
	H_{min} [mm]	H_{max} [mm]	Amplitude [mm]
Double-cylindrical coil	92	97	±2.5
Single-cylindrical coil	71.5	72.0	±0.25
Conical helical coil	18.25	18.75	±0.5

Single and double cylindrical springs were mounted in the testing machine by means of special gripping system for axial loading and with respect to accordance with the operational, Fig. 2. Springs were mounted coaxially concerning an axial force direction on a lower plate. An upper plate of the gripping device covers the movement of a stamp in the form of nuts employing a hole. The both screws were used for the stamp location with respect to a displacement loading.

In the case of fatigue tests the other griping system was used, Fig. 3. This was represented by lower and upper griping plate while the first one was connected with the servo-motor. This has enabled to applied a loading to the lower plate directly. The upper mounting plate containing a rigid zone enabled the loading to be transferred to the tested object.. Fatigue tests were carried out up to 1×10^6 cycles were obtained (which was estimated to correspond to the number of braking operations over a distance of 300-500 thousand kilometres of vehicle operation) or to fracture, whichever occurred first. Parameters of fatigue experiment and courses of displacement are shown in Fig. 4.

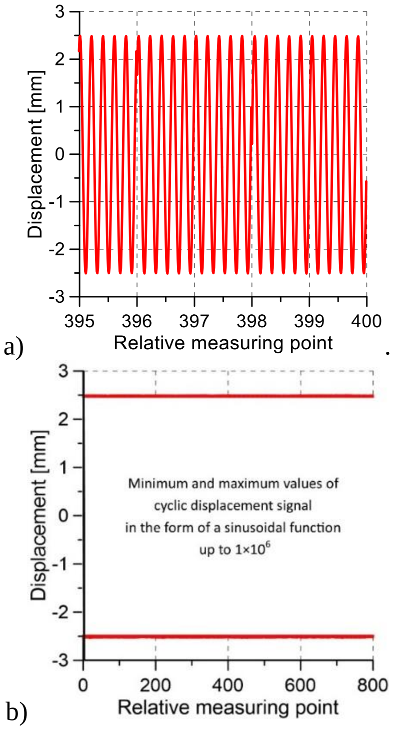

Fig. 4. Tension-compression cycles versus the relative measurement point from the double helical spring test: (a) at ½ of the number of final cycles; (b) in the range from 0 to about ¾ of the total number of cycles, represented by the value 1×10^6 (represented by the values of relative measuring point)

3. RESULTS

The first stage of the static tests has enabled to collect relationship between force and displacement as well as their maximum values, Fig. 5. This was done employing four springs of the same type from the same manufacturer. A correlation matrix was then created for each spring group, Tab. 3. This means the tested object can be selected easy without the influence of the spring quality on the results.

Table 3. Correlation matrix for the W.1 springs series

	W.1.1	W.1.2	W.1.3	W.1.4
W.1.1	1	0.9987	0.999	0.989
W.1.2	0.999	1	0.998	0.999
W.1.3	0.999	0.998	1	0.995
W.1.4	0.989	0.999	0.995	1

Table. 4. Correlation coefficients between pair of the same spring type of both manufacturers

Springs pair	Pearson correlation coefficientt
B.1 – W.1	0.997
B.2 – W.2	0.996
B.3 – W.3	0.992

The obtained results from the matrix approach indicate their very strong correlation, which means high repeatability of spring properties for one group. It was therefore possible to use only one spring from each series of the tested object for testing. Then, the correlation coefficients between the spring responses of the same spring type from both manufacturers were determined. The results are summarized in Tab. 4, while the graphical comparison represented by force-displacement relationship is shown in Fig. 6.

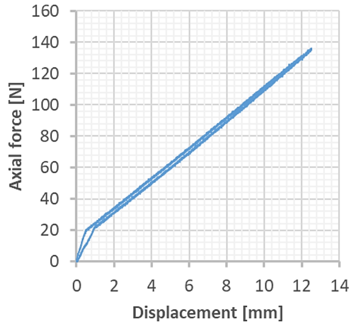

Fig. 5. Axial force versus displacement for the W1.2-2 spring from static test

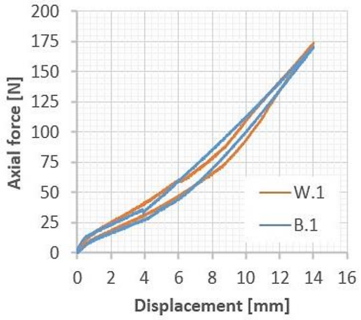

Fig. 6. Force-displacement from static testing the spring denoted as W.1 and B.1 in the as-received state

The correlation matrix approach was also used for analysis of fatigue results in a form of relationship between force and displacement. For this case B.1, B.2, B.3 (Tab. 5) and W.1, W.2 and W.3 (Tab. 6) ones represented the reference objects i.e. in the as-received state and after fatigue test. Results from this approach focused on response of the B.1 spring due to cyclic loading is illustrated in Fig. 7.

Pearson correlation coefficient has follow a high value for the same type of springs (B.1 - W1; B.2 - W.2; B.3 – W.3) from both manufacturers before and after fatigue testing, Tab. 7, Fig. 8.

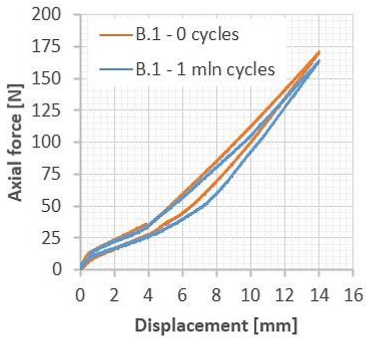

Fig. 7. Comparison of force-displacement relationship for B.1(0 loading cycles) – B.1 (1×10^6 loading cycles) springs

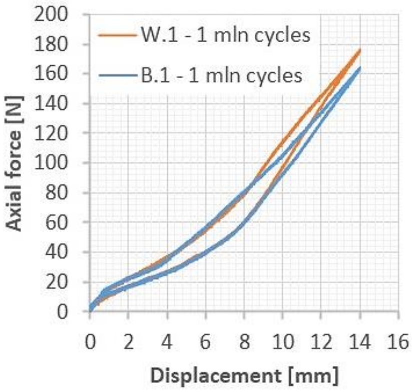

Fig. 8. Comparison of force-displacement relationship for W.1 and B.1 springs subjected to fatigue testing up to 1×10^6 loading cycles

Table 5. Pearson correlation coefficient of B.1, B2 and B.3 springs in the as-received state(0 loading cycles) and after fatigue test (1×10^6 loading cycles)

Spring code	Pearson correlation coefficient
B.1-1 (0 loading cycles)	0.98
B.1-1 (1×10^6 loading cycles)	
B.2-1 (0 loading cycles)	0.99
B.2-1 (1×10^6 loading cycles)	
B.3-1 (0 loading cycles)	0.97
B.3-1 (1×10^6 loading cycles)	

Table 6. Pearson correlation coefficient of W.1, W2 and W.3 springs in the as-received state (0 loading cycles) and after fatigue test (1×10^6 loading cycles)

Spring code	Pearson correlation coefficient
W.1-1 (0 loading cycles)	0.99
W.1-1 (1×10^6 loading cycles)	
W.2-1 (0 loading cycles)	0.99
W.2-1 (1×10^6 loading cycles)	
W.3-1 (0 loading cycles)	0.98
W.3-1 (1×10^6 loading cycles)	

Table 7. Pearson correlation coefficient for B.1, W.1, B.2,W.2, B.3, W.3 springs subjected to fatigue testing up to 1×10^6 loading cycles

Spring code	Pearson correlation coefficient
1×10^6 loading cycles	
B.1	0.99
W.1	
B.2	0.97
W.2	
B.3	0.97
W.3	

4. SUMMARY

1. The static relationship force-displacement of the tested objects was sectionally linear which indicates the springs constant follows different values resulted the component technical features.
2. The springs in the as-received state, which were collected in each group, had very similar responses on a static loading. Independently on the manufacturer. This proves the high repeatability of the production process and mechanical properties of the springs tested.
3. Independently on the manufacturer, comparison of the static test results of the same spring type subjected to fatigue up to 1×10^6 loading cycles showed a very strong correlation in responses.
4. Fatigue tests simulating the number of braking over a distance of 300-500,000 showed that the elastic responses of the springs intended to the drum braking system did not change significantly.
5. Considering that the manufacturers of brake drums and brake lining predict their durability at 150,000 and 50,000 kilometres, respectively, it can be concluded that springs are the most reliable element of this system.
6. The differences in the elastic response of the spring can be taken into account for the modelling.

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