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STRESS DISTRIBUTION ALONG THE CROSS SECTION OF THE NARROWEST PART OF THE DIAPHRAGM SPRING FINGERS

SASKO MILEV AND DARKO TASEVSKI AND BLAGOJA NESTOROVSKI

Abstract. Torque from the engine of the vehicles to the transmission of the cars is transferred using friction clutches. A part of these clutches is a diaphragm spring. In this paper, using the Finite Element Method the distribution of the stress along the cross section of the narrowest part of the diaphragm spring fingers and the intensity of the changes of the stress differences in individual parts of this section are studied.

1. Introduction

A diaphragm spring is a part of the friction clutch, which is used in the cars for transferring torque from the engine of the vehicle to the transmission. As one of the main parts of the clutch, it creates pressure force of the clutch and enables engagement and disengagement of the clutch. A diaphragm spring, when performing its function, is dynamically loaded. It is a single thin sheet of metal which yields when pressure is applied to it. (Fig.1)

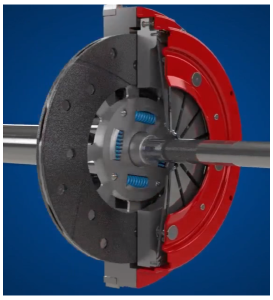

Figure 1. *Friction clutch with diaphragm spring*

When pressure is removed, the metal spring backs to its original shape. The central part of the spring is split into numerous fingers that act as release levers (unblock). When the clutch is in disengagement, the fingers are moved forward by the release bearing, separation of the driving disc and pressure plate from flywheel occurs and this prevents the rotation of the driving disc. And as it is a piece with a high stress concentration in driving conditions, this is often the cause of cracks and destruction (breaking) of the springs. The diaphragm spring clutch (Fig.2) has more compact designs than other clutches; it is easy to balance rotationally and is less subjected to defects due to the centrifugal force at high rotational speeds. It needs no release levers, minimum effort is sufficient to disengage the clutch, it provides a minimum number of moving components so minimum internal friction is experienced. It gives uniformly distributed pressure on the pressure plate. In the Diaphragm spring the load deflection curve is not linear. To get a higher coefficient of friction, the size and diameter of the diaphragm spring is increased.

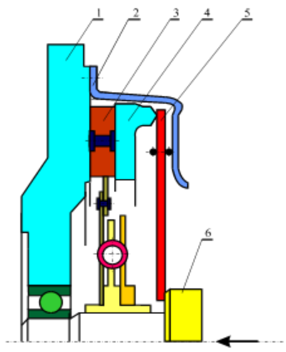

Figure 2. C*lutch parts: 1. Flywheel, 2. Clutch cover, 3. Disk, 4. Pressure plate, 5. Diaphragm spring, 6. Release bearing*

The purpose of this research is, using the Finite Element Method, to determine stress distribution along the cross section of the narrowest part of the diaphragm spring fingers.

2. Materials and Methods

The material used for the FEM model of the spring is obtained from the technical specifications of the examined diaphragm spring and that is structural steel designated as 51CrV4, with the modulus of elasticity of E=210.000 N/mm^2, density of R_{ho} = 7.85g/cm^3 and Poisson ratio Nu = 0.3. The yielding point considering the material enhancement for this spring is R_{eh} = 1300 [N/mm^2]. (Figure 3)

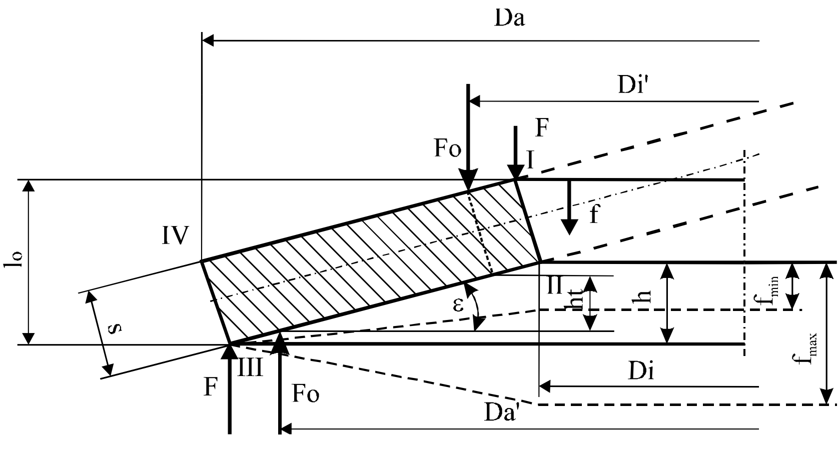

Figure 3. *Diaphragm spring parameters*

Internal diameter of a diaphragm spring Di = 313 [mm],
Outer diameter of a diaphragm spring Da = 395 [mm],
Spring thickness s = 5.2 [mm],
Module of elasticity of the steel E = 210.000 [N /mm^2],
Poisson number of the spring steel μ = 0.3,
Internal diameter of the diaphragm spring, supporting points Di’ = 336 [mm], Outer diameter of the diaphragm spring, supporting points Da’ = 392 [mm],
Path of the clutch while disengaging ℓ = 12 [mm],
Release bearing diameter d = 120 [mm].
Finite element (FE) analysis of the diaphragm spring was used for this research.
Numerous simulations were run for bringing closer the spring behavior. Despite the sheet metal form that, in general, would be modeled with shell elements, for the diaphragm spring volumetric solid hexagon elements were used. Analyzing the results that were concluded by using shell

elements, the stress distribution from the upper face was equalized with the stress from the bottom face. When using surface or shell elements, both are equalized and non-real stress distribution occurs. Despite the sheet metal form of the spring, for this kind of parts subjected to similar forces volumetric elements on several layers must be used.

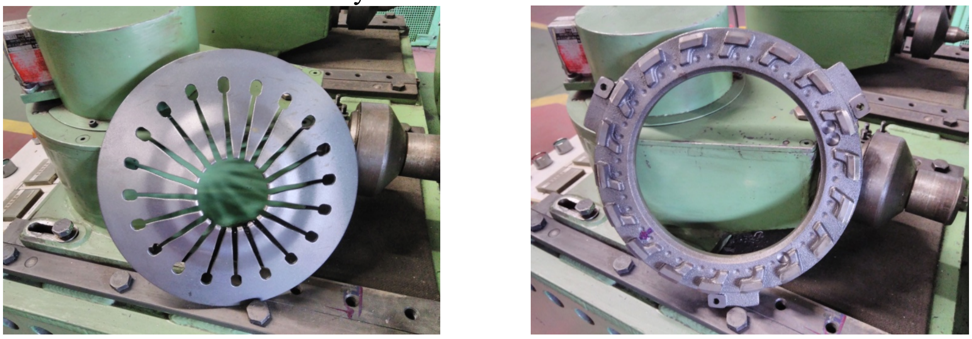

Figure 4. *Basic model of diaphragm spring and Pressure plate from friction clutch*

Using the Finite Element Method, the stresses along the cross-sections of the lower, narrowest part of the spring arms were determined. This part is divided into 11 finite elements, the obtained values for the stresses for each of the finite elements are given in Table 1, and for these values a diagram of the distribution of stress was constructed.

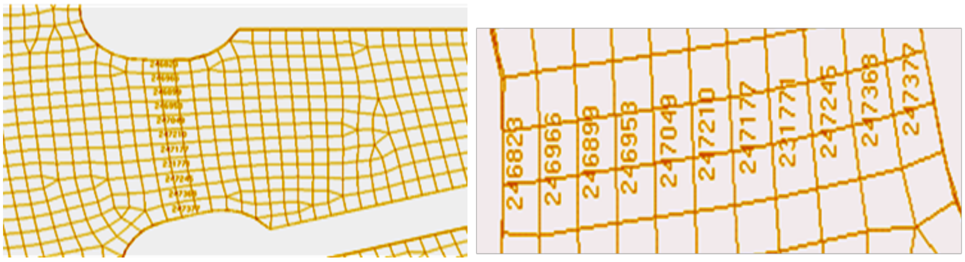

Figure 5. Mesh of measuring points

3. Results and discussion

Table 1. *Stresses calculated with FEM, in lower, the narrowest part of the diaphragm spring fingers*

Finite element	6823	6966	6899	6953	7049	7210	7177	1771	7245	7368	7377
	1	2	3	4	5	6	7	8	9	10	11
Spring deflection [mm]	Stress N/mm^2	Stress N/mm^2	Stress N/mm^2	Stress N/mm^2	Stress N/mm^2	Stress N/mm^2	Stress N/mm^2	Stress N/mm^2	Stress N/mm^2	Stress N/mm^2	Stress N/mm^2
1,4	74	71	66	60	55	53	54	58,5	65	71	73
2,8	128	125	117	106	97	93	95	103	115	127	132
4,2	166	158	148	132	120	115	116	125	140	155	164
5,6	175	161	149	131	118	112	114	124	141	158	173
7,1	190	169	153	132	118	112	113	124	144	165	186
8,5	221	204	186	161	146	141	142	154	178	203	220
10	263	262	248	222	207	201	203	216	241	263	262
11,4	316	341	336	311	296	291	293	306	330	343	316
12,8	383	435	446	425	412	408	411	422	444	444	388
14,1	469	544	569	552	541	537	538	547	563	546	468

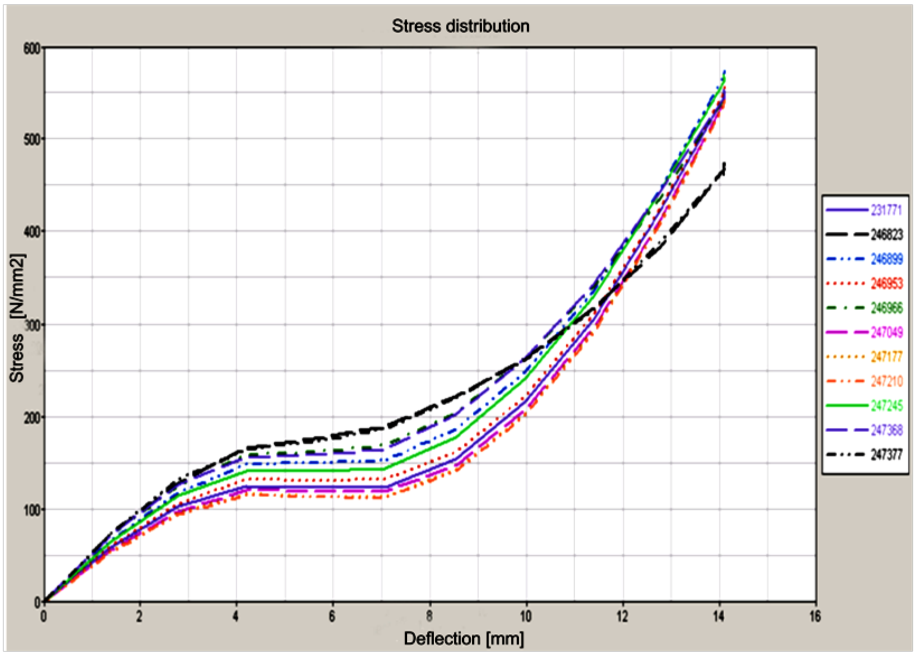

Figure 6. *Stresses calculated with FEM, in lower, the narrowest part of the diaphragm spring fingers-horizontal axis-deflection[mm], vertical axis-stress[N/mm^2]*

From the data in Table 1 and the diagram (Fig. 6) it is clear that the points distributed symmetrically in terms of the longitudinal axis of the spring finger have nearly the same stresses. With the increase in the deflection of the diaphragm spring, we have a gradual increase in the stresses in this part of the diaphragm spring. At the same time, the end finite elements marked with 1 and 11 are the most loaded. The finite element in the middle, marked with 6, is the least loaded. The maximum stresses, which occur at a deflection of the spring greater than 12 millimeters, are less than 600 [N/mm^2]; they are much lower in relation to the stresses to which parts of the plate spring are exposed (up to 1,400 [N/mm^2]).

Until spring deflection is smaller than 10[mm], finite elements 1 and 11 are more loaded than finite elements 2 and 10. When deflection is above 10[mm], the situation is reversed, finite elements 2 and 10 are more loaded than finite elements 1 and 11, and they are the most loaded from all observed finite elements. During the whole time when the deflection rises from 0 to 14.1 [mm], the stress in the middle part rises too, but these finite elements are still not the most loaded parts. When spring deflection is 12.8[mm] or more, the most loaded finite elements are 3 and 9 and they stay most loaded until the end of observing, to deflection 14.1[mm]. In the next Figure 7, the values for the spring deflection are plotted on a horizontal axis, the stress differences between adjacent finite elements are plotted on the vertical axis. Example: line diff. 1 to 2 means the difference in the stresses between finite element 1 and finite element 2, etc.

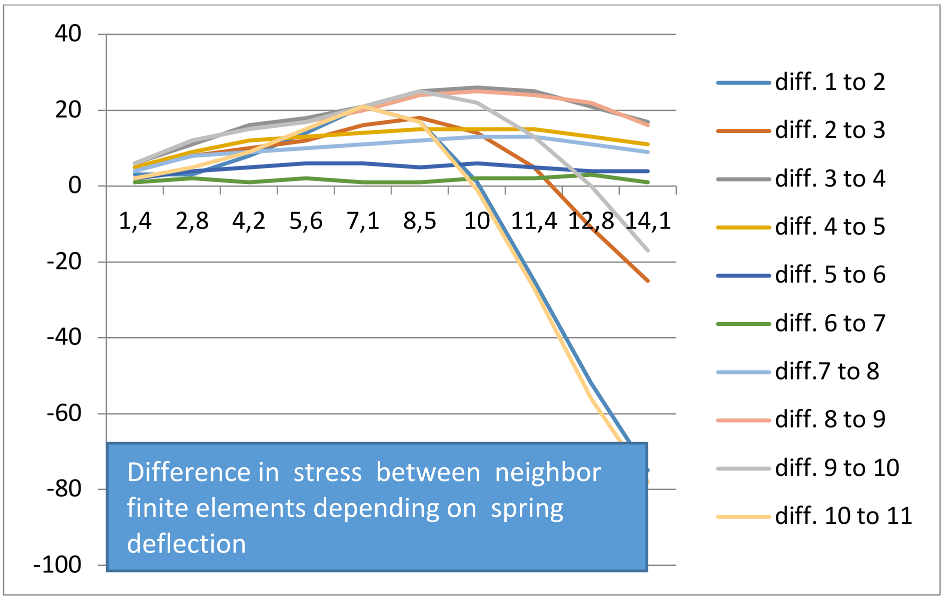

Figure 7. *Difference in stress between adjacent finite elements depending on the diaphragm spring deflection*

From the diagram (Figure 7) it is clear that during the spring deflection, the biggest change in the difference in the stresses occurs between outer finite elements 1 and 2 and between 10 and 11. One of the ways to achieve a balance of stresses in this part of the diaphragm spring is to improve the manufacturing process, to reduce as much as possible the residual stresses that appear during this process.

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