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ANALYSIS OF DEFORMATIONS AND VON-MISSES STRESSES OF THE MAIN SHAFT SYSTEM IN A LOAD BREAK SWITCH

ANALIZA DEFORMACIJA I VON-MISESOVIH NAPONA SISTEMA GLAVNOG VRATILA SREDNJENAPONSKE SKLOPKE

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Abstract: This paper presents the study of deformations and Von-Misses stresses of the main shaft system during opening and closing operations of a rotary SF₆ load break switch (LBS). The shaft consists of three axially connected parts made of steel where components are on ground potential and of plastic material, which is on high potential. The insulating shaft carries three rotating knife-blade contacts for the three phases. Static deformation of the insulating shaft is calculated by applying a defined torque between the two ends of the shaft. The results allow deducing the dynamic deformation. Maximum values of Von-Misses stresses are located at the geared connection between the plastic and the steel shaft. The rotation of the shaft system is measured synchronously by two optical rotary encoders in the front and rear sides of the LBS. The results confirm the twisting of the shaft system and provide its elastic deformation values. Travel curves obtained on both side of the LBS show different courses with respect to overtravel and rebound. Discrepancies can be explained by the deformation of the main shaft due to the acting forces, whereas manufacturing tolerances resulting in loose have a certain contribution.

Keywords: load break switch, operating mechanism, travel measurements, Multi-body dynamic analysis, Von-Misses stresses, deformations, shock absorber

Sažetak: Rad prikazuje analizu deformacija i Von-Misses napona glavnog vratila tokom operacije isklopa i uklopa rotacione SF₆ sklopke (LBS - Load Break Switch). Vratilo se sastoji od tri aksijalno spojena dijela proizvedena od čelika koji su uzemljeni i plastičnog dijela koji je pod naponom. Izolaciono vratilo nosi tri rotirajuća kontakta u obliku noža za tri faze. Statičke deformacije izolacionog vratila su izračunate na način da je zadan momenat na kraju vratila. Rezultati su omogućili određivanje dinamičkih deformacija. Maksimalna vrijednost Von-Misses napona je locirana u ozubljenom dijelu vratila gdje se čelično i plasično vratilo spajaju. Rotacija sistema vratila je izmjerena sinhronizovano pomoću dva optička rotaciona enkodera sa prednje i zadnje strane LBS-a. Rezultati potvrđuju uvijanje sistema vratila i daju vrijednosti elastičnih deformacija. Hodogrami koji su snimljeni sa obje strane LBS-a pokazuju različite vrijednosti prelijetanja i odbijanja. Razlike u hodogramima se mogu objasniti deformacijom glavnog vratila usljed djelujućih sila, gdje tolerancije zadane crtežima imaju određeni utjecaj na diskrepanciju rezultata.

Ključne riječi: sklopka, pogonski mehanizam, mjerenje hodograma, dinamika sistema krutih tijela, Von-Misesovi naponi, deformacije, amortizer

INTRODUCTION

Switchgear industry involves both electrical and mechanical problems and phenomena and is one of the most challenging industries that humans created and evolved during the past 100 years.

This paper is analysing one subassembly of a rotary Load Break Switch (LBS). The main shaft assembly, which represents a system of three shafts, is analysed in this pa-

per. The first shaft is connected directly to the mechanism (drive shaft), the second shaft is connecting (transmission shaft) the drive shaft and the plastic operating shaft inside the enclosed chamber filled with SF₆ gas (Figure 1). Due to the rotation of the main shaft system, discrepancies were noticed during the measurements of the travel diagrams on the front side (mechanism side) and the rear side of the shaft. The focus of this paper is to explain qualitatively and quantitatively the phenomenon that is causing this discrepancy. The two factors that could cause the

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twisting of the plastic shaft, and the tolerances at the connections are analysed numerically and experimentally in this paper. Based on these results, different diagrams are formed to quantitatively point out the reason of the discrepancy. Throughout the paper different phenomena are recognized, pointed out and based on the analysis at the end of the paper with the conclusions, recommendations for improvements are given.

Based on the mentioned analysis of this paper, earlier research cannot be found on the same or similar topic. Few papers exist that cover the same problematic of this paper. In [1] stress analysis with multi-body dynamics software of a double motion mechanism is conducted and directions for improvement are given. The difference between the two papers is that the final moving contact is translating, whereas in this paper the contacts are positioned inside the plastic shaft and are rotating with the shaft. The analysis is different as well. The multi-body dynamics analysis is giving information about the joint force of the double motion mechanism in the aforementioned paper, whereas in this paper, the acting torque is increased to find the value that would cause the deformation captured on the travel diagram. [2] is explaining the principals and phenomenon of different switchgears, including the rotary LBS.

Papers [3]-[4] are related to reliability testing of a high voltage circuit breaker as a whole and the specific double motion mechanism. These experimental analyses are interesting in the sense that they give insight to weak points of a mechanism, different practical solutions, and stress contours of mechanism parts. In comparison to this paper, the difference is the working principal of the two devices and the involvement of a non-stiff rotating plastic shaft.

A study on dynamic behaviour of multi-body dynamics model on ring main unit is given in paper [5]. The ring main unit uses SF_6 gas for insulation and therefore and it consists of load break switch, fuse combination switch and circuit breaker. In the paper are shown results of a multi-body dynamics simulation and experimental displacement diagrams of the main contact. The experimentally obtained diagrams are similar to the one obtained in this paper, but the clarification why they are different from the theoretical ones was not given whereas this paper intends to clarify the phenomenon of this discrepancy.

In figure [6] an optimization of a linear puffer medium voltage circuit breaker operating mechanism and drive system is shown. A dynamical model of the analysed system is given with experimental results related to the mechanical analysis of the device. Based on this results values of the acting torque through the mechanism are given where comparing to the values in this paper they are lower because of the weaker spring. Different parameters influencing the load inside the mechanism, which could be of an importance to be analysed in a future research to quantify how these would influence the phenomenon analysed in this paper are given.

Paper [7] analyses the properties of the materials which were used for the manufacturing of the polycarbonate (PC) shaft in this research. Besides the glass fibre (GF) used to reinforce the PC material the influence of carbon fibre (CF) is analysed also. Different combinations of fibre percentage and fibres are given in the paper and how they are influencing the mechanical and electrical properties. Based on this paper, CF cannot be used because of the ability to increase the conductivity of the PC.

The paper consists of five sections. The first section describes the model that is analysed, and it forms the hypothesis of the analysed phenomenon. The second and third section are related to a numerical and experimental analysis of the model. The fourth section is giving an insight in a short tolerance analysis of the main shaft system. In the fifth section conclusions and recommendations are given.

1. MODEL AND PROBLEM DESCRIPTION

In this section the 3D model of the analysed parts and the discrepancy are shown. A hypothesis of the problem is also explained. The system that will be analysed is shown in Figure 1. The main shaft system consists of the 3 parts, the driving shaft, which is a part of the operating mechanism, the steel shaft or the transmission shaft, which is transmitting the energy from the mechanism to the enclosed chamber with high potential. The transmission shaft is then connected with a plastic (PCGF 20) shaft, which is located in the enclosed chamber filled with SF_6 gas. The plastic shaft carries the knife-blade contacts with leaf springs, which are providing the contact force. In Figure 1, the plastic shaft is denoted with 2, the transmission shaft is denoted with 3 and the driving shaft is denoted with 4. The front and rear side are marked with 5 and 1. The main shaft assembly is a combination of two materials, the driving and the transmission shaft are made out of steel, whereas the shaft with the contact is made of plastic as it was mentioned.

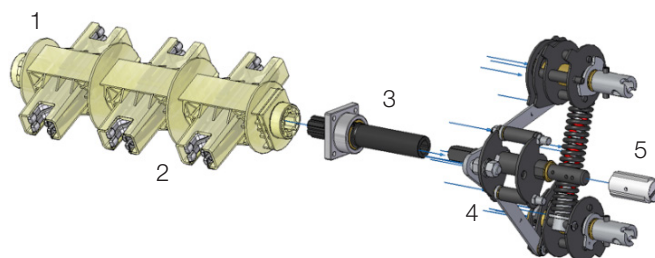


Figure 1: Physical model of the shaft main shaft system. 1- Rear side; 2- Plastic shaft (PCGF 20); 3- Steel shaft; 4- Drive shaft; 5- Front side

The discrepancy was initially observed by using the high-speed camera on the front and rear side of the module. Optical encoders were then installed on both sides to validate what was observed. Figure 2 represents synchronized measurements of the travel recordings. A large difference between the overtravel and rebound values of the 2 shafts are denoted. Two main reasons for the discrepancy

ancy are identified: the possible deformation of the plastic shaft and the loose tolerance on the shaft couplings.

As it can be seen in Figure 2, the first overtravel value on the rear is 27.00 mm, were on the front the value is much smaller and it is 8.6 mm. The first rebound value on the rear is 21.4 mm, whereas on the front this value is 4.5 mm.

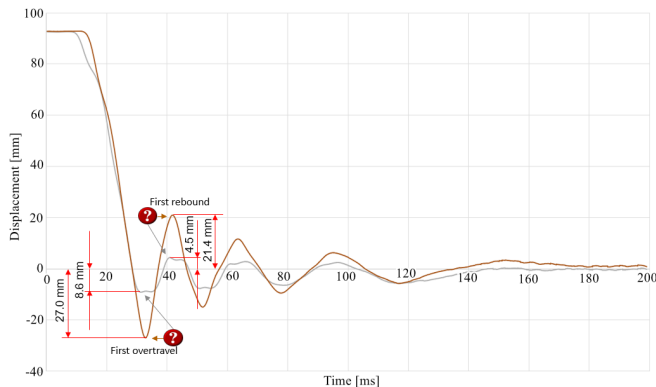


Figure 2: Synchronized measurements of the opening operation on the front and rear side of the LBS using front and rear optical encoders

The drive shaft is stopped by the two hooks (Figure 3) of the mechanism while the plastic shaft continues to “rotate”, the rotation of the plastic shaft is deforming, due to the inertial torques, its angular momentum and the loose tolerances between the plastic shaft and the steel shaft coupling. Because the plastic shaft is less stiff than the steel shafts, it was expected it will deform the most.

The hooks, which are preventing the overtravel and rebound, are shown in Figure 3 and are denoted with I and II. The shaft is only stopping the driving shaft (Figure 1 denoted with 4), which is the first shaft in the main shaft system of the LBS.

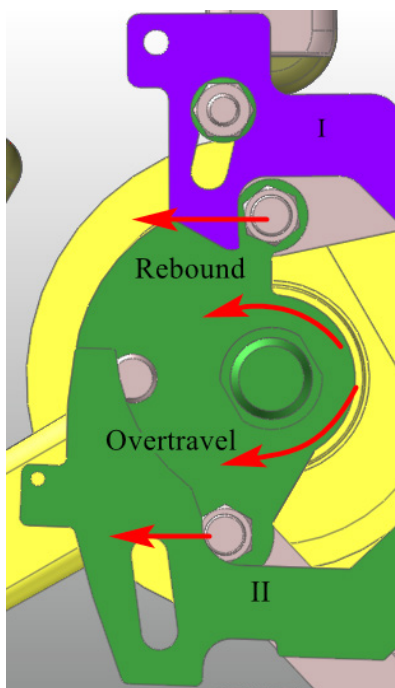


Figure 3: Hooks that are preventing overtravel and rebound: I - Overtravel, II - Rebound

2. NUMERICAL ANALYSIS

In this section results obtained by numerical FEM analysis is shown. The analysis is conducted using ANSYS Mechanical. The model is shown in Figure 4. The highlighted blue part (A) on the front is assumed to be fixed. A torque is applied at the end of the plastic shaft and is denoted as B in a red rectangle (rear). Based on the model the applied torque will be the same from the rear to the position of the fixed support. This assumption is approximating the inertial torque, which is acting on the whole length of shaft assembly.

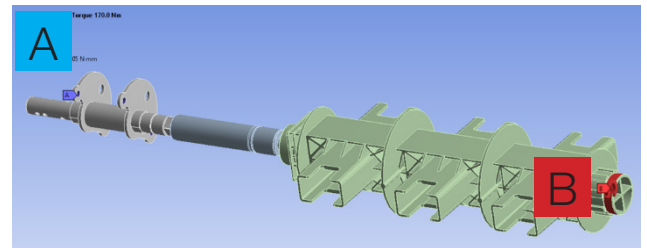


Figure 4: ANSYS simulation model with the boundary condition BC positions indicated

Figure 5 and Figure 6 show the deformation and Von-Mises stresses obtained from the analysis. The positions on which the values of the deformation values are extracted are the position where the contact is. The highest stress concentration is occurring on the geared connection between the transmission and the plastic shaft. The value of the acting torque is varied so that it is causing stresses from very small values to a value at which the shaft would fail with a safety factor of 1.50.

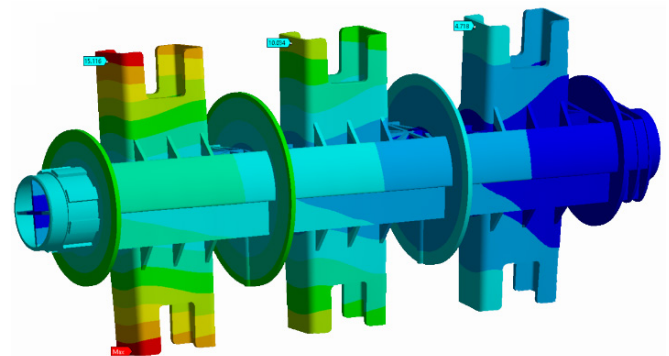


Figure 5: Numerical results – Deformation of the plastic shaft; Red – maximum values of deformation; Blue – minimum values of deformation

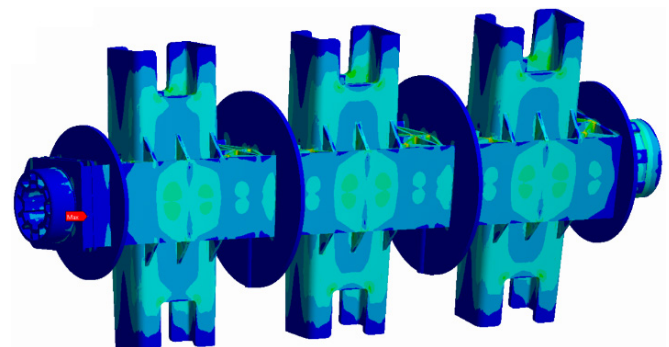


Figure 6: Numerical results – Von-Mises stress of the plastic shaft; Red – maximum values of the stress; Blue – minimum values of the stress

The results of the shaft system deformation and stresses are gathered in Table I.

Table I: Numerical results obtained by ANSYS

Torque value [Nm]	Max Von-Mises stress [MPa]	Deformation of the last contact chamber [mm]	Deformation of the middle contact chamber [mm]	Deformation of the first contact chamber [mm]
170	92.1	15.116	10.034	4.601
160	86.68	14.282	9.442	4.346
140	75.85	12.524	8.266	3.865
120	65.01	10.668	7.095	3.335
100	54.17	8.916	5.865	2.757
80	43.34	7.113	4.731	2.204
60	32.5	5.345	3.523	1.644
40	21.67	3.586	2.365	1.104
20	10.83	1.767	1.171	0.545

Based on the results in Table I, a diagram is showing how the deformations at each contact chamber are changing due to different values of the acting torque. The largest deformations are on the rear contact, 15.1 mm at a torque of 170 Nm. At the same torque value, the deformation of the middle contact resulted is 10.0 mm, while the deformation of the front contact is only 4.6 mm.

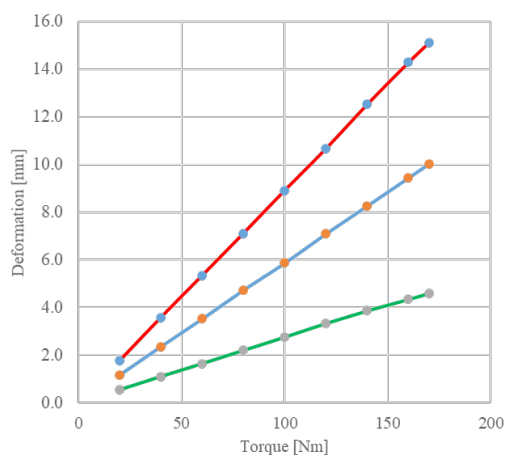


Figure 7: Main shaft contact chamber deformation vs Torque – Numerical results; — - Rear contact chamber; — - Middle contact chamber; — - Middle contact chamber;

3. EXPERIMENTAL ANALYSIS

Experimental analysis is conducted on a new LBS, which is shown in Figure 8.a). The measurements are conducted similarly to the numerical analysis, where a torque is applied at the end of the shaft while the driving shaft is fixed at the same and positioned as it was carried out in the numerical analysis. A torque wrench, as shown in Figure 8.b), was used to vary the torque applied from 80 Nm until 200 Nm. The deformations, as well as the tolerance stuck up, are measured using a digital calliper shown in Figure 8.c).

The clearance between the different couplings of the shaft assembly showed a total tolerance stuck up of 9.4 mm. The deformation measurement of each contact chamber at various torque values applied is given in Table II.

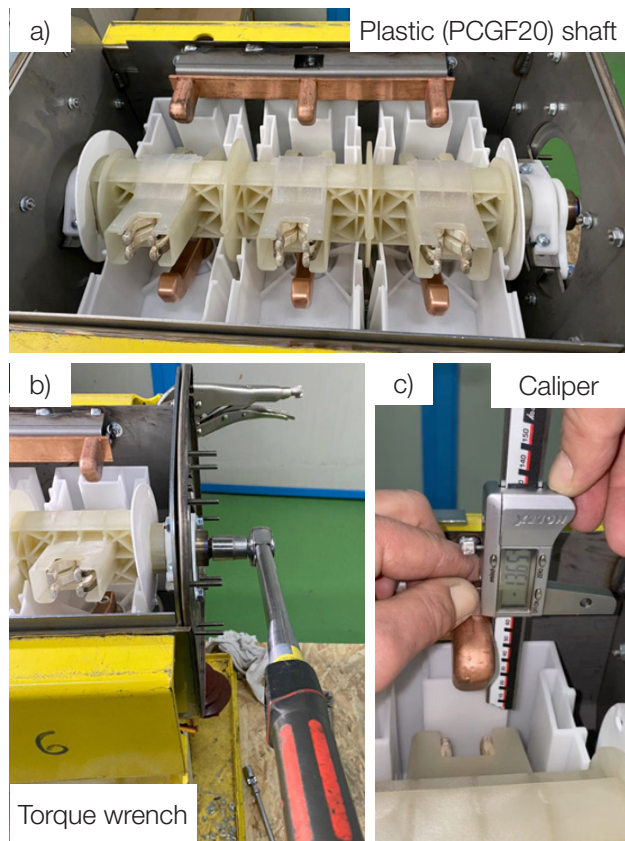


Figure 8: Experimental analysis set up

Table II: Experimental results

Torque value [Nm]	Deformation of the first contact chamber [mm]	Deformation of the middle contact chamber [mm]	Deformation of the last contact chamber [mm]
80	5.47	9.25	13.19
100	6.62	11.23	15.22
120	7.14	12.39	16.47
140	8.75	14.34	18.74
160	9.76	16.09	22.58
180	10.43	18.02	25.84
200	12.91	20.09	28.26

Based on the results shown in Table II, diagram that is showing the deformation of each chamber of the plastic shaft in reference to the applied torque at the rear of the shaft is shown in Figure 9.

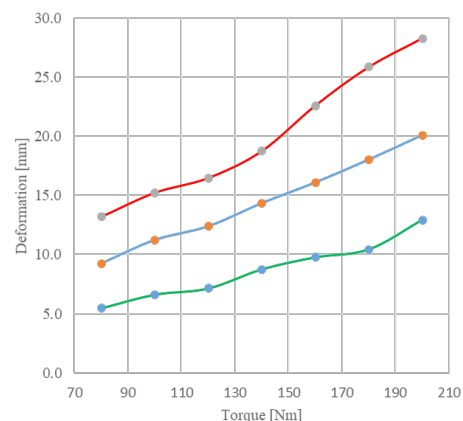


Figure 9: Main shaft contact chamber deformation vs Torque – Experimental results; — - Rear contact chamber; — - Middle contact chamber; — - Middle contact chamber;

4. TOLERANCE ANALYSIS

The tolerance analysis based on the manufacturing drawing was conducted. The connecting “pin/hole” positions at the shaft assembly were decreased and increased by the minimum values of the “pins” and maximum values of the “holes” in 3D software. These positions are shown in Figure 10.

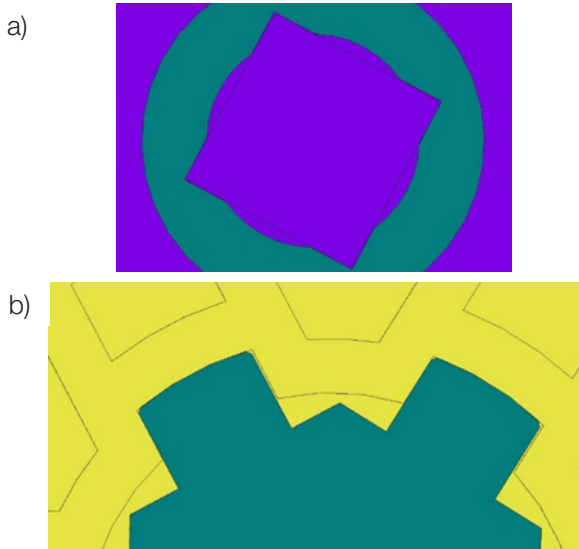


Figure 10: Connection between the shafts; a) Connection between the drive and the transmission shaft; b) Connection between the transmission and the plastic shaft

The plastic and the transmission shafts are rotated for the maximum value to get in contact the “pins” at both connection points as shown in Figure 11. This motion resulted in a total contact deformation of 1.04 mm.

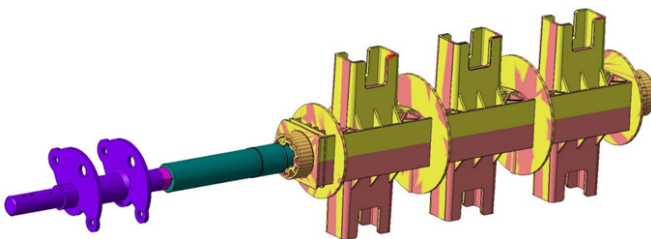


Figure 11: Tolerance analysis in a 3D software with the maximum and minimum values of tolerances applied based on the manufacturing drawings

Besides the tolerance analysis in the software, a tolerance analysis was conducted by adding together the values from the drawings. The allowable motion resulted from the analysis is 0.972 mm. This was performed as a validation of the tolerance analysis conducted in the 3D CAD software because the discrepancy between the theoretical and the experimental value is enormous. The results and values of this analysis are shown in Table III.

Free rotation due to tolerance stack measured on a new LBS is 9.4 mm. This value is much higher than the theoretical one, which is only 1 mm.

Table III: Tolerance summary according to the drawings

Plastic (PCGF20) shaft [mm]	+0.05	
Steel shaft [mm]	-0.05	+0.086
Drive shaft [mm]	-0.027	
Theoretical clearance calculated on the points where the rotary contacts separate [mm]	0.972	

5. DISCUSSION AND CONCLUSION

Travel curves measured synchronously by two optical rotary encoders show discrepancy between the main shaft rotation on the front and the rear sides of LBS.

Static deformation of the plastic (PCGF20) shaft is calculated by applying a defined torque between the two ends of the shaft. Deformations are calculated on the points where the rotating knife-blade contacts separate from the fixed contacts. The results shown in Figure 2 confirm the twisting of the shaft system and provide its elastic deformation values.

The open-end travel limiter and the hook of the latch stop the drive shaft as shown on Figure 3. This system does not break the plastic shaft with rotating contacts, which continues moving thus transmitting its rotational kinetic energy to the twisting of the shafts, resulting in mechanical oscillations with large overtravel and rebound values. The mechanical oscillations are damped only by friction, where the shaft goes to a resting state after few oscillations.

The process explained above causes square-shaped oscillations on the front, while sinusoid-like oscillations occur on the rear side of the LBS due to the deformation of the plastic shaft.

Figure 12 shows the comparison between the numerical and experimental results of the shaft deformation versus torque applied. The comparison is performed in the way

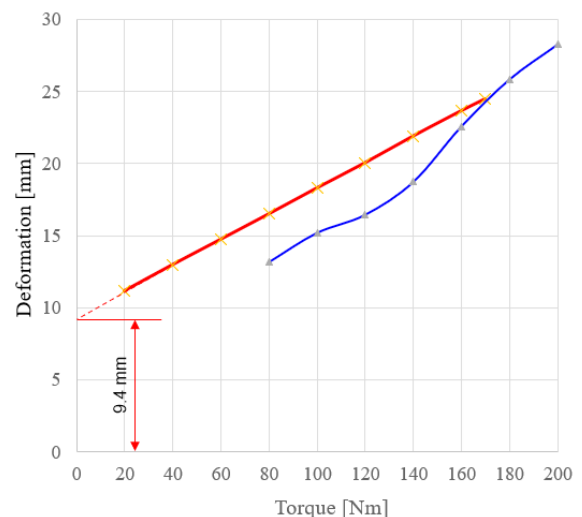


Figure 12: Comparison of numerical and experimental results of Torque vs Deformations. — - Rear contact numerical; — - Rear contact experimental;

that the values of the tolerance clearance (experimental values), which is allowing the plastic shaft to rotate freely, is added to the deformation results of the rear contact obtained by FEM analysis. The reason for this is to be able to compare the numerical and experimental results. The other way this could be carried out is by removing the tolerance clearance from the experimental results to obtain just the values of deformation. Good alignment is observed especially at higher torque values where the results are almost overlapping.

Maximum values of Von-Misses stresses are located at the geared connection between the plastic (PC GF20) and the steel shaft. This correlates with the results of mechanical tests where the geared connection is failing at this position after loading the shaft with high values of torque.

The comparison between the numerical and experimental results of the obtained travel curves, deformations of the shaft and the tolerances proves that the combination of the shaft deformation and the tolerance clearance is causing the discrepancy between the front and rear measurements of the travel traces.

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BIOGRAPHY

Džanko Hajradinović received PhD degree in Mechanical Engineering from University of Sarajevo, Sarajevo, Bosnia and Herzegovina, in 2021. Now he works at Mechanical Engineering Faculty Sarajevo as Assistant professor and as a consultant for G&W Electric. His current research interests include: analysis of dynamical systems, nonlinear vibrations, stress and deformation analysis of nonmechanical components manufactured from different materials, medium voltage switchgear.

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Nenad Uzelac graduated from School of Electrical Engineering in Belgrade, Serbia in 1995 with a major in Electrical Power Engineering, and completed Master of Science in Product Development at Northwestern University, Evanston, USA in 2004. In 1999 Mr. Uzelac joined G&W Electric company, as an R&D engineer, and since 2015 he is leading G&W Global Research, tasked to develop new technologies. His research interests include AC and DC switching, materials and insulation technologies, sensors, condition monitoring and diagnostics, sensors and arc fault studies. Mr. Uzelac is currently the chair of CIGRE "Transmission and Distribution Equipment" study committee, as well as an active member of IEEE switchgear committee.

Maroua Touhami was born in Casablanca, Morocco in 1996. She graduated with her Bachelor and Master's degree in Mechanical/ Aerospace engineering from Université Internationale de Rabat and Mississippi State University in 2018, 2020 respectively. She is currently a research engineer at G&W Electric Bolingbrook. Her current professional interests include multiphysics simulation for MV and HV circuit breakers and high-speed imaging for various switchgear applications.

Mirsad Kapetanović was born in Visoko, Bosnia and Herzegovina, in 1953. He received his M.Sc. degree for research work on electrical endurance of HV circuit breakers from the Sarajevo University, Bosnia and Herzegovina, in 1993 and the Ph.D. degree for discovery of the algebra of fractal vector (Bosnian algebra) from the same University, in 1997 (Ph.D. thesis: "Algebra of Fractal Vector as a Contribution to Electric Field Calculation"). He has worked for the Energoinvest Electric Power Institute (IRCE) in Sarajevo since 1977. As of 1982, he has been

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