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N.T. Pavlović / N.D. Pavlović

DESIGN OF COMPLIANT SPRING GUIDING MECHANISM

1. INTRODUCTION

Compliant mechanisms gain at least some of their mobility from the deflection of flexible members rather than movable joints only [2]. Compliant mechanisms can provide many benefits in the solution of design problems. They are desirable because they have less wear, weight, noise and backlash than their rigid-body counterparts. By reducing the number of required interconnections the reliability of a design can be improved. Although there are many advantages, the inclusion of compliance provides several challenges in mechanism analysis and design. Nonlinearities introduced by the large deflection of members further complicate the analysis of compliant mechanisms. The mechanisms with compliant joints can realize relatively small displacements, that is, their mobility is limited. Another limitation to their use is fatigue failure at the elastic joints.

There are many papers considering the structure and function of the compliant joints and compliant mechanisms. The paper [5] established basic nomenclature and classification for the components of compliant mechanisms. The paper [6] introduced a method to aid in the design of a class of compliant mechanisms wherein the flexible sections (flexural pivots) are small in length compared to the relatively rigid sections. The paper [7] presented a formal structural optimization technique called the homogenization method in order to design flexible structures (compliant mechanisms). The paper [9] introduced new ideas of technically realizable joints from nature and their integration into elastically movable structures for motion tasks in positioning and manipulating engineering.

Some papers have analyzed the influence of the geometry, as well as the material type of the compliant joints on the guiding accuracy of the compliant mechanisms [4,11]. The paper [8] introduced a method for determining the limit positions of compliant mechanisms for which an appropriate pseudo-rigid-body model may be created. The paper [10] deals with the influence of the geometry, as well as the material type of the compliant joints on a mobility of the single compliant joint and entire compliant mechanisms.

This paper takes into consideration some typical spring guiding systems with translating planar displacement of the link, and their compliant counterparts. The aim of the paper is to present some new designs of compliant mechanisms, as well as to suggest optimal design solution being able to realize translating planar displacement of the link.

2. COMPLIANT PARALLEL-GUIDING MECHANISMS

Figure 1 shows a rigid-link parallel-guiding mechanism. The mechanism is a simple four-bar linkage in which the opposing links have the same length, thus forming a parallelogram.

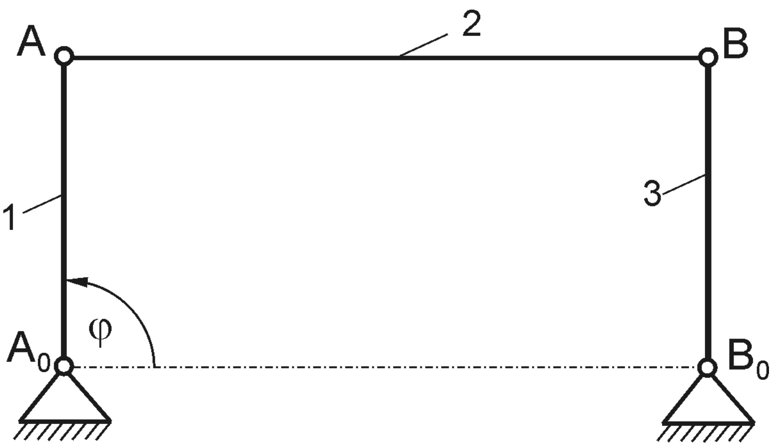

Fig. 1. Rigid-link parallel-guiding four-bar linkage

A spring guiding system based on the rigid-link parallel-guiding four-bar linkage [1] has been shown in the Fig.2 and Fig.3. The bottom end of the springs 1 and 3 has been fixed to the frame at the points A_0 and B_0 respectively (Fig.2), whilst the other end has been connected to the movable "coupler" at the points A and B. The force F acting at the coupler point A causes parallel guiding of the coupler AB.

The horizontal displacement Δx of the coupler AB and vertical deviation Δy (the difference between realized and straight line link translation) can be calculated using the equations [1]:

$$\Delta x = \frac{Fl^3}{24EI_z} \qquad\qquad \Delta y = \frac{3(\Delta x)^2}{5l} \qquad\qquad (1.a,\ b)$$

where:

E - Young's modulus,

I_z – area moment of inertia.

On the basis of the equations (1.a,b) it can be concluded that the horizontal displacement of the "coupler" AB (Δx) and vertical deviation (Δy) do not depend on the "coupler" length $c = \overline{AB}$.

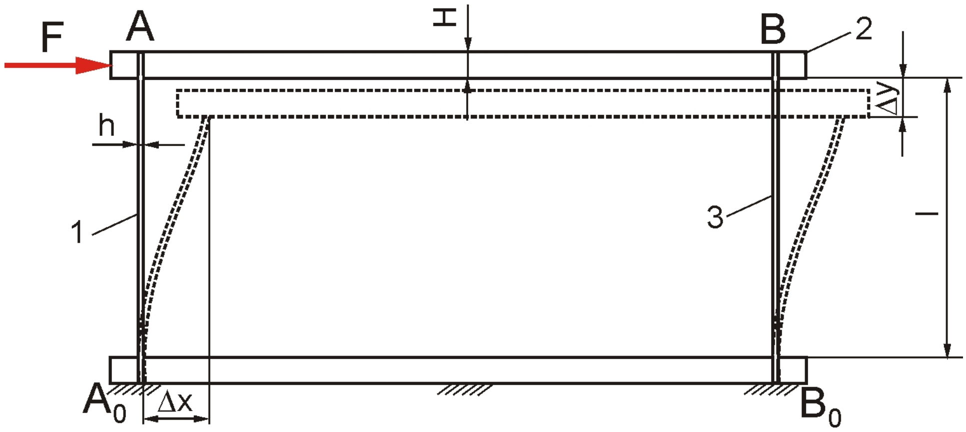

Fig. 2. Spring guiding system based on the rigid-link parallel-guiding four-bar linkage

Figure 3 shows a parallel-guiding plate-spring mechanism [2] as an example of practical use of compliant spring guiding mechanism. This mechanism is used in various fields of application for force-displacement measurement systems, accurate and reproducible motion in optical systems, and guiding parts over small displacement while subject to and without disturbance from dynamic loading forces.

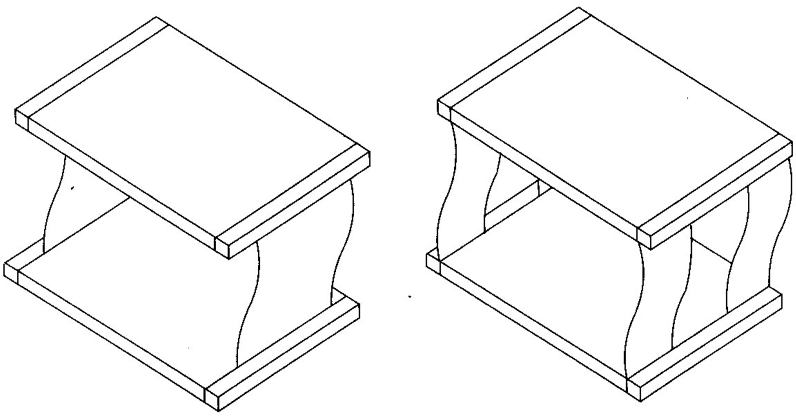

Fig. 3. Plate-spring mechanism [2]

We have performed the calculation of the "coupler" horizontal displacement and vertical deviation (Δy) for the spring guiding system with dimensions: $a = \overline{A_0A} = \overline{B_0B} = 40\ \text{mm}$, $c = \overline{AB} = \overline{A_0B_0} = 80\ \text{mm}$. Cross sectional area of the links is assumed to be rectangular, defined by the width $b = 5$ mm and height $h = 0.5$ mm (for the links A_0A and B_0B), that is, width $B = 16$ mm and height $H = 8$ mm (for the link AB).The springs are assumed to be made of spring steel, with Young's modulus $E = 210000$ N/mm^2. The results have been shown in the Table 1.

F [N]	Δx [mm]	Δy [mm]
5	1.219	0.022
10.257	2.5	0.094
20.514	5	0.375
41.028	10	1.5

Table 1. The horizontal displacement of the "coupler" (Δx) and vertical deviation (Δy) of the spring guiding system (Fig. 2) according to the equations (1)

We have also performed the calculation of the "coupler" horizontal displacement and deviation (Δy) for the above mentioned mechanism by using of ANSYS Software.

Two-dimensional Elastic Beam (Fig.4) has been used as a characteristic ANSYS element type in the calculation procedure. The element has three degrees of freedom at each node (I, J): translations in the nodal x- and y-directions and rotation about the nodal z-axis.

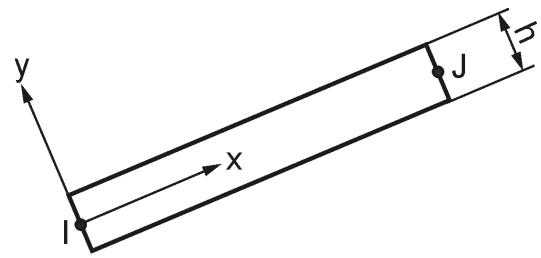

Fig. 4. Two-dimensional Elastic Beam

The results obtained by using of the ANSYS Software have been shown in the Table 2. These results are similar to the ones obtained by the equations (1).

F [N]	Δx [mm]	Δy_{max}[mm]
5	1.218	0.020
10.3	2.5	0.085
20.75	5	0.342
43.1	10	1.390

Table 2. The horizontal displacement of the "coupler" (Δx) and maximal vertical deviation (Δy_{max}) of the spring guiding system (Fig.2) by using of ANSYS Software

In order to obtain possibly better guiding accuracy (smaller deviation Δy), we have designed the compliant mechanisms with the beam and film joints [11] as the counterparts of the rigid-link parallel-guiding four-bar linkage.

Fig.5a shows a link of the compliant mechanism with beam joint. The dimensions of the rectangular cross section have been denoted with b and h for the elastic segment, that is, B and H for the rigid segment, while the length of the elastic segment has been denoted with l (Fig.5b).

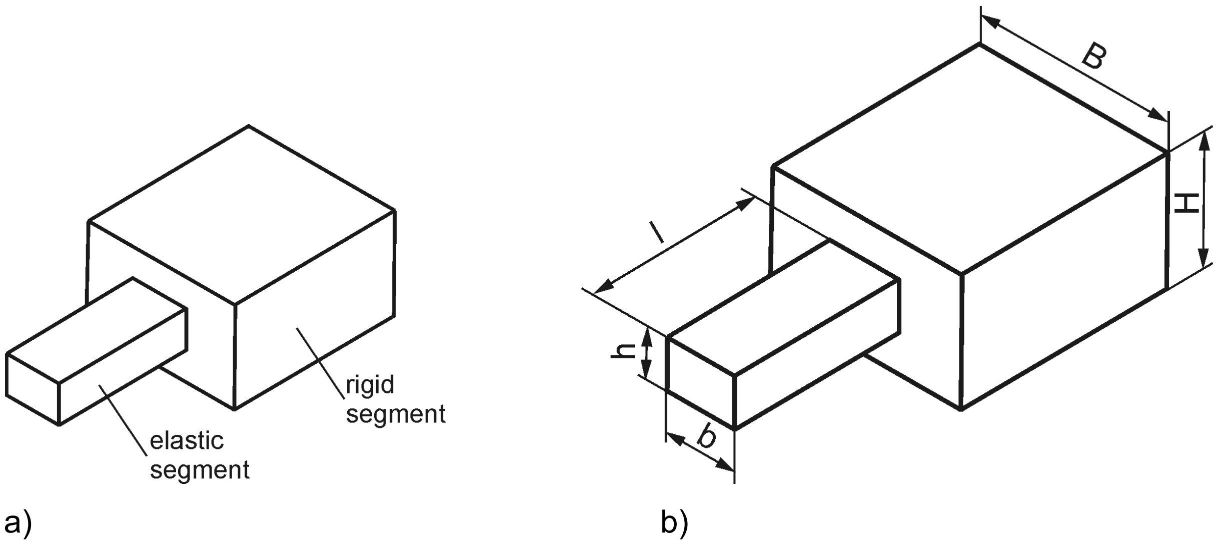

Fig. 5. A link of the compliant mechanism with beam joint

Fig. 6 shows a designed compliant mechanism with the beam joints as a counterpart of the rigid-link parallel-guiding four-bar linkage.

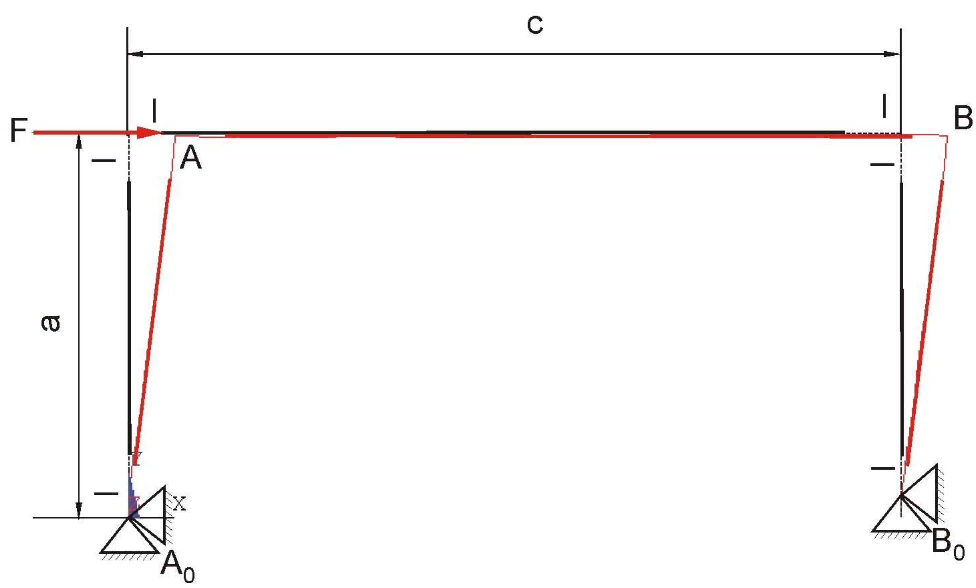

Fig. 6. A compliant mechanism with the beam joints
as a counterpart of the rigid-link parallel-guiding four-bar linkage

We have performed the displacement calculation for this compliant mechanism with above-mentioned parameters (a = 40 mm, c = 80 mm, b = 5 mm, h = 0.5 mm, B = 16 mm, H = 8 mm, E = 210000 N/mm^2) by using of ANSYS Software. The results of the "coupler" horizontal displacement and vertical deviation (Δy) have been shown in the Table 3 and in the Fig.7 for different values of the length of the elastic segments (l).

l = 2.5 mm		**l = 5 mm**		**l = 10 mm**	
Δx [mm]	**Δy_{max} [mm]**	**Δx [mm]**	**Δy_{max} [mm]**	**Δx [mm]**	**Δy_{max} [mm]**
2.5	0.120	2.5	0.162	2.5	0.247
5	0.400	5	0.489	5	0.668
10	1.463	10	1.657	10	2.033

Table 3. The horizontal displacement of the "coupler" AB (Δx) and maximal vertical deviation (Δy_{max}) of the compliant mechanism (Fig.6)

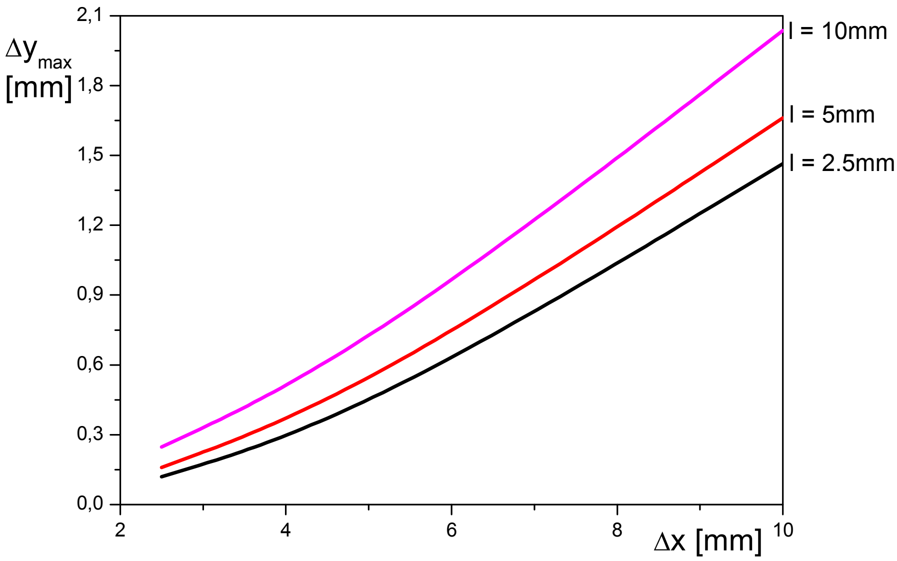

Fig 7. The horizontal displacement of the "coupler" AB (Δx) and maximal vertical deviation (Δy_{max}) of the compliant mechanism (Fig.6)

The guiding accuracy decreases with increasing of the "length" of the compliant beam joint. It means that the compliant mechanism with concentrated compliance (small "length" of the beam joint) provides better guiding accuracy.

Fig.8a shows a link of the compliant mechanism with film joint. The widths of the elastic segment and rigid segment have been denoted with w_E and w_R respectively, while the „length“ of the elastic segment has been denoted with l (Fig. 8b).

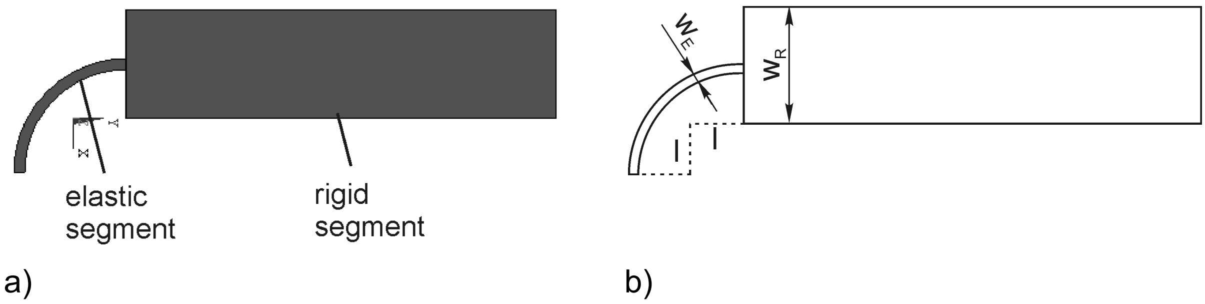

Fig. 8. A link of the compliant mechanism with film joint

Fig. 9 shows a designed compliant mechanism with the film joints as a counterpart of the rigid-link parallel-guiding four-bar linkage.

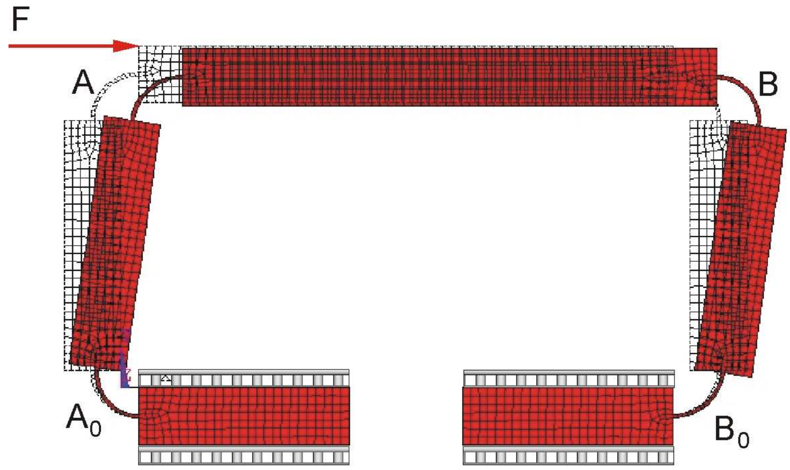

Fig. 9. A compliant mechanism with the film joints
as a counterpart of the rigid-link parallel-guiding four-bar linkage

We have performed the displacement calculation for this compliant mechanism (a = 40 mm, c = 80 mm, w_E = 0.5 mm, w_R = 8 mm, l = 2.5 mm, material thickness δ = 5 mm, E = 210000 N/mm^2) by using of ANSYS Software.

Two-dimensional-eight-node Structural Solid (Fig. 10) has been used as a characteristic ANSYS element type in the calculation procedure. The element is defined by eight nodes (I, M, J, N, K, O, L, P) having two degrees of freedom at each node: translation in the nodal x- and y-directions.

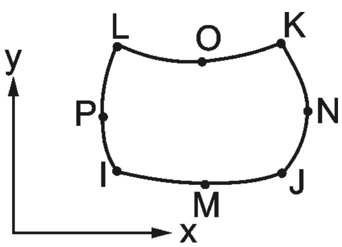

Fig. 10. Two-dimensional-eight-node
Structural Solid

The results of the "coupler" horizontal displacement and vertical deviation (Δy) have been shown in the Table 4.

l = 2.5 mm	
Δx [mm]	**Δy_{max} [mm]**
2.5	0.081
5	0.298
10	1.147

Table 4. The horizontal displacement of the "coupler" AB (Δx) and maximal vertical deviation (Δy_{max}) of the compliant mechanism (Fig.9)

If we compare the results presenting in the Tables 1, 2, 3 and 4, we can conclude that introducing of compliant film joints improves guiding accuracy of the compliant spring guiding system. However, introducing of compliant beam joints does not improve guiding accuracy of the compliant spring guiding system.

3. S-SHAPED SPRING GUIDING SYSTEM

Figure 11 shows a S-shaped spring guiding system [1] as an example of the compound spring guiding system, that has been designed in order to provide greater horizontal displacement of the "coupler".

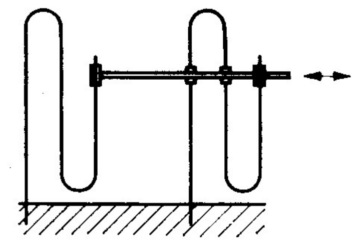

Fig. 11. S-shaped spring guiding system [1]

The S-shaped spring guiding system can be analyzed as a compliant mechanism with distributed compliance (Fig.12).

We have performed the calculation of the "coupler" horizontal displacement and vertical deviation (Δy) for this compliant mechanism by using of ANSYS Software. The link lengths of the mechanism are: $a = 40$ mm, $c = d = 80$ mm, $e = f = 20$ mm. Cross sectional area is assumed to be rectangular, defined by the width $b = 5$ mm and height $h = 0.5$ mm (for the links A_0A and B_0B), that is, width $B = 16$ mm and height $H = 8$ mm (for the link AB). The results for different values of radius (r) have been shown in the Table 5 and in the Fig.13.

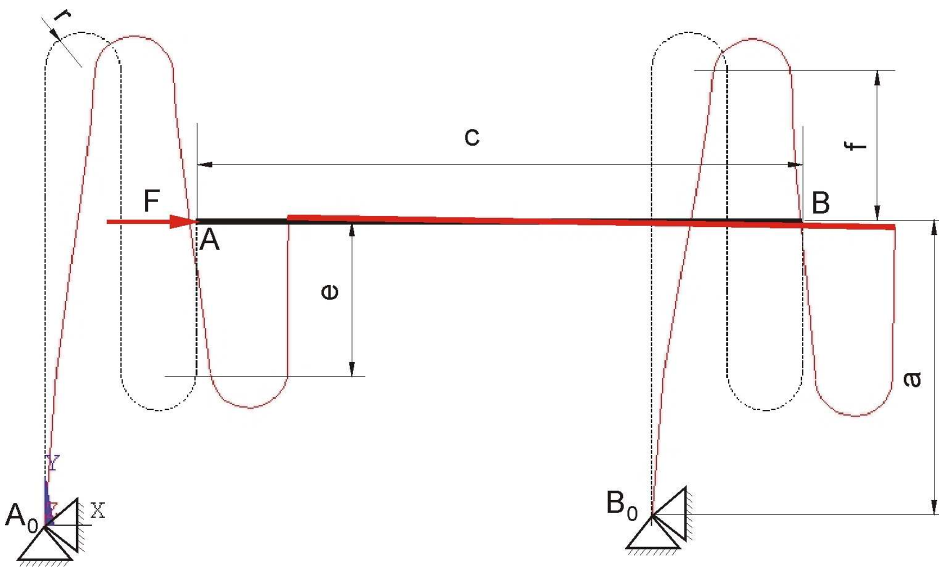

Fig. 12. The S-shaped spring guiding system as a compliant mechanism

r = 2.5 mm		r = 5 mm		r = 10 mm	
Δx [mm]	**Δy_{max} [mm]**	**Δx [mm]**	**Δy_{max} [mm]**	**Δx [mm]**	**Δy_{max} [mm]**
2.5	0.054	2.5	0.133	2.5	0.464
5	0.177	5	0.261	5	0.943
10	0.637	10	0.891	10	1.929

Table 5. The horizontal displacement (Δx) and maximal vertical deviation (Δy_{max}) of the S-shaped spring guiding system (a = 40 mm, c = d = 80 mm, e = f = 20 mm)

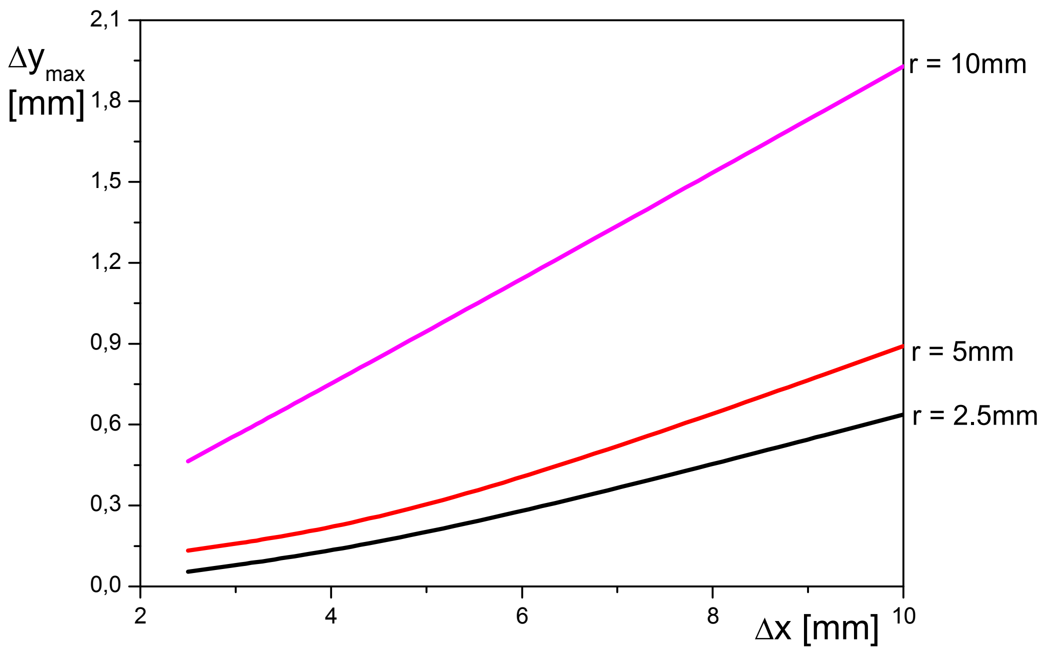

Fig. 13. The horizontal displacement (Δx) and maximal vertical deviation (Δy_{max}) of the S-shaped spring guiding system (a = 40 mm, c = d = 80 mm, e = f = 20 mm)

The best guiding accuracy has been provided by the S-shaped spring guiding system with the smallest value of curve radius (r = 2.5 mm).

Further, we have analyzed the influence of the dimension e = f on the guiding accuracy. The results have been shown in the Table 6 and in the Fig.14.

e=f = 5 mm		**e=f = 10 mm**		**e=f = 15 mm**		**e=f = 20 mm**		**e=f = 25 mm**		**e=f = 30 mm**	
Δx [mm]	**Δy_{max} [mm]**	**Δx [mm]**	**Δy_{max} [mm]**	**Δx [mm]**	**Δy_{max} [mm]**	**Δx [mm]**	**Δy_{max} [mm]**	**Δx [mm]**	**Δy_{max} [mm]**	**Δx [mm]**	**Δy_{max} [mm]**
2.5	0.398	2.5	0.304	2.5	0.173	2.5	0.054	2.5	0.072	2.5	0.116
5	0.974	5	0.762	5	0.460	5	0.177	5	0.146	5	0.257
10	2.662	10	2.125	10	1.366	10	0.637	10	0.305	10	0.617

Table 6. The horizontal displacement (Δx) and maximal vertical deviation (Δy_{max}) of the S-shaped spring guiding system (a = 40 mm, c = d = 80 mm, r = 2.5 mm)

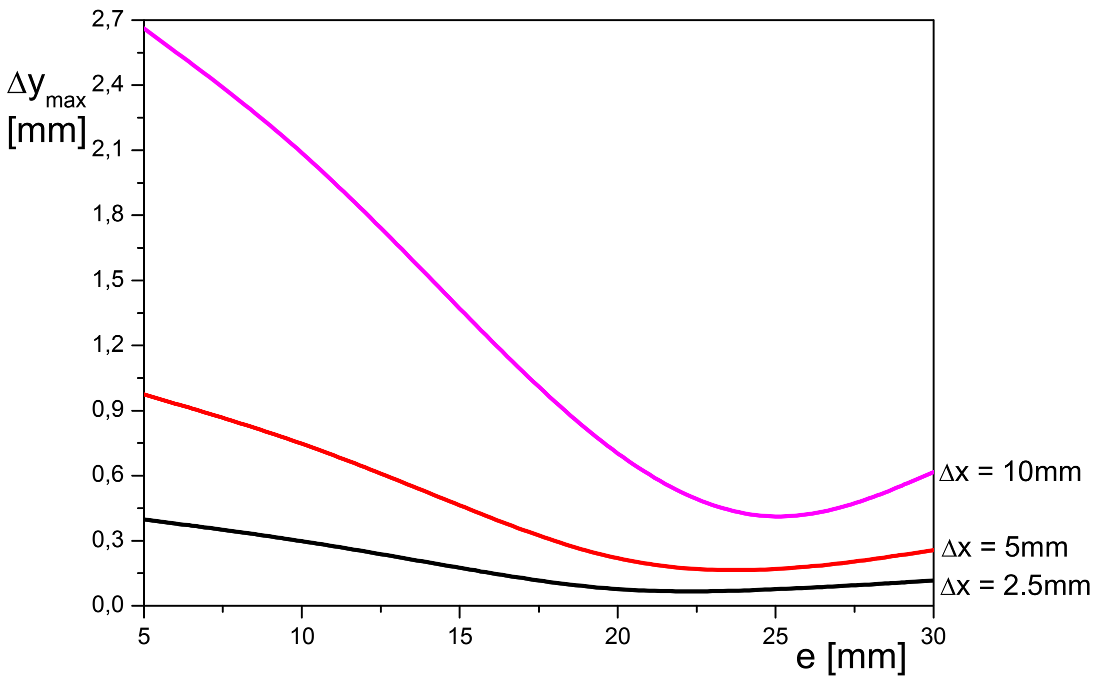

Fig.14. The horizontal displacement (Δx) and maximal vertical deviation (Δy_{max}) of the S-shaped spring guiding system (a = 40 mm, c = d = 80 mm, r = 2.5 mm)

The best guiding accuracy has been provided by the S-shaped spring guiding system with the values of e = f = 22 mm (Table 7).

e = f = 22 mm	
Δx [mm]	**Δy_{max} [mm]**
2.5	0.033
5	0.082
10	0.383

Table 7. The horizontal displacement (Δx) and maximal vertical deviation (Δy_{max}) of the S-shaped spring system (a = 40 mm, c = d = 80mm, r = 2.5mm, e = f = 22mm)

4. MOBILITY OF THE COMPLIANT GUIDING MECHANISMS

The compliant guiding mechanisms are moveable due to flexibility of their elastic segments. However, their mobility is limited. We have researched the limits of their mobility by using of ANSYS Software. The links are assumed to be made of spring steel (Young's modulus E = 210000 N/mm^2, bending strength σ_{bs} = 900 N/mm^2). Maximal permissible bending stress $\boldsymbol{\sigma_{max} < \sigma_{bs}}$ determines constraint positions of the links, that is, the limits of their displacement (mobility) and maximal permissible acting force.
The results have been shown in the Table 8. Maximal realizable displacement of the guiding "coupler" AB has been denoted with Δx_{max}. Maximal acting force causing appearing of maximal permissible bending stress has been denoted with F_{max}.

Compliant guiding mechanism	**F_{max} [N]**	**Δx_{max} [mm]**	**Δy_{max} [mm]**
Fig. 2	18.7	4.515	0.278
Fig. 6	14.2	1.556	0.056
Fig. 9	13.8	6.165	0.445
Fig. 12	10.5	18.84	1.53

Table 8. The mobility of the compliant guiding mechanisms

The mobility and guiding acurracy of the compliant guiding mechanisms have been compared in the Fig. 15. The mobility of the compliant mechanism with beam joints (Fig. 6) has been denoted with Δx^1_{max}, the mobility of the spring guiding system based on the rigid-link parallel-guiding four-bar linkage (Fig. 2) has been denoted with Δx^2_{max}, the mobility of the compliant mechanism with film joints (Fig. 9) has been denoted with Δx^3_{max}, and the mobility of the the S-shaped spring guiding system (Fig. 12) has been denoted with Δx^4_{max}.

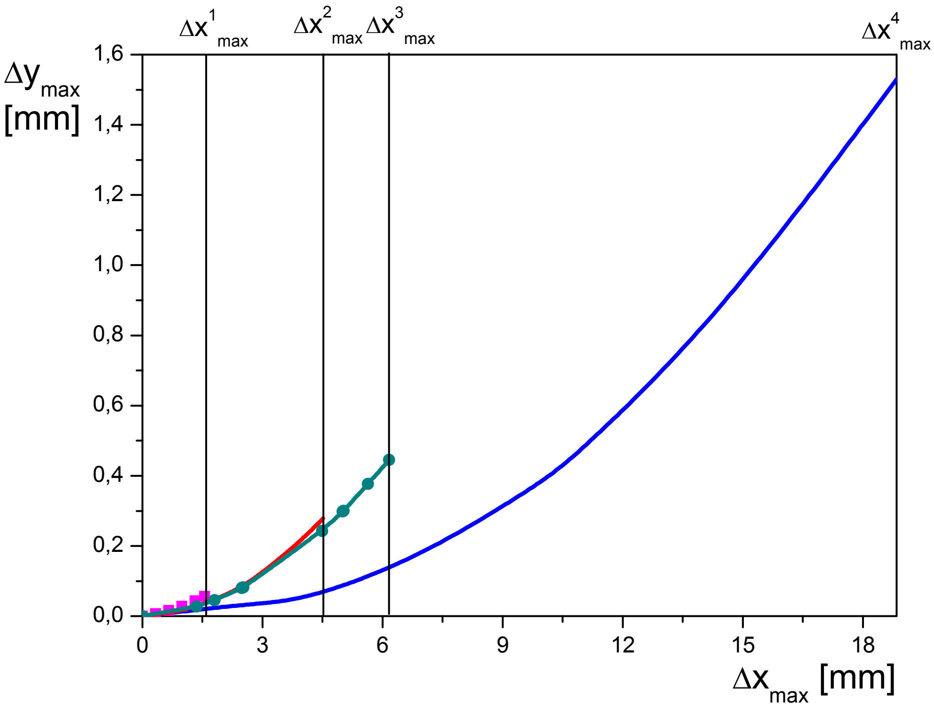

Fig. 15. The mobility and guiding acurracy of the compliant guiding mechanisms

5. CONCLUSION

Using of compliant mechanisms is desirable because they have less wear, weight, noise and backlash than their rigid-body counterparts. In this paper some designs of compliant mechanisms being able to realize translating planar displacement of the link have been analyzed.

Three types of the compliant counterparts of the rigid-link parallel-guiding four-bar linkage have been analyzed: the spring guiding system with distributed compliance, where the input crank and the follower have been the elastic segments of the mechanism, and the compliant mechanisms with beam and film joints, where the compliance has been concentrated in the joints.

On the basis of the displacement calculation results obtained by using of ANSYS Software, it can be concluded that introducing of compliant film joints improves guiding accuracy of the compliant spring guiding system; however, introducing of compliant beam joints does not improve guiding accuracy of the compliant spring guiding system.

It has been also concluded that the compliant mechanisms with more concentrated compliance (smaller "length" of the compliant joints) provide better guiding accuracy.

Further, we have analyzed a S-shaped spring guiding system as a compliant mechanism and determined its optimal dimensions in order to obtain the guiding accuracy as good as it is possible. For the mechanism made of the spring steel, with the dimensions a = 40 mm, c = d = 80 mm, r = 2.5 mm, e = f = 22 mm, the maximal deviation (the difference between realized and straight line link translation) Δy_{max} = 82 μm has been obtained on the planar displacement of Δx = 5 mm. This guiding accuracy is better than the ones of the above mentioned compliant mechanisms developed as the counterparts of the rigid-link parallel-guiding four-bar linkage.

Also, we have analyzed the mobility of the developed compliant mechanisms, that is, the limit positions of the translating link determined by permissible maximal bending stress. The S-shaped spring guiding system provides considerably greater mobility and the acting force providing the mobility is smaller in comparison with the compliant mechanisms developed as the counterparts of the rigid-link parallel-guiding linkage.

Therefore we suggest the S-shaped compliant spring guiding mechanism of the obtained optimal dimensions to be used for realization of the translating planar displacement of the link.

References:

[1] Hildebrandt, S., Feinmechanische Bauelemente, VEB Verlag, Berlin, 1988.

[2] Howell,L.L., Compliant Mechanisms, John Wiley &Sons, Inc., New York, 2001.

[3] Christen,G., Pfefferkorn,H., Zum Bewegungsverhalten nachgiebiger Mechanismen, Wissenschaftlliche Zeitschrift der Technishen Universität Dresden, 50 (2001) Heft 3, 53 - 58.

[4] Pavlović,N.T., Pavlović,N.D., Rectilinear Guiding Accuracy of Roberts-Чебышев Compliant Four-Bar Linkage With Silicone Joints, Механика на машините, година XII, книга 4, ISSN 0861-9727, TU Varna, (2004), 53-56.

[5] Midha, A., Norton, T.W., Howell, L.L., On the Nomenclature, Classification and Abstractions of Compliant Mechanisms, ASME Journal of Mechanical Design, Vol. 116, No. 1, 1994, 270-279.

[6] Howell, L.L., Midha, A., A Method for the Design of Compliant Mechanisms with Small-Length Flexural Pivots, ASME Journal of Mechanical Design, Vol. 116, No.1, 1994, 280-290.

[7] Ananthasuresh, G.K., Kota,S., Designing compliant mechanisms, Mechanical Engineering, Vol. 117, No.11, November 1995, 93- 96.

[8] Midha,A., Howell,L.L., Norton,W., Limit positions of compliant mechanism using the pseudo-rigid-body model concept, Mechanism and Machine Theory, Vol.35, No.1, 2000, 99-115.

[9] F.Böttcher, G.Christen, H.Pfefferkorn, Structure And Function of Joints And Compliant Mechanism, Motion Systems 2001, Collected Short Papers of the Innovationskolleg “Bewegungssysteme” Friedrich-Schiller Universität Jena, Technische Universität Jena, Technische Universität Ilmenau, Shaker Verlag, Aachen 2001, 30-35.

[10] Pavlović,N.T., Pavlović,N.D., Mobility of the compliant joints and compliant mechanisms, Theoretical and Applied Mechanics, Belgrade, 2005, Vol. 32 (4), ISBN 0350-2708, 341-357.

[11] N.T.Pavlović, N.D.Pavlović, Stress Analysis And Guiding Accuracy of The Compliant Four-bar Linkages for Rectilinear Guiding, 47. Internationales Wissentschaftliches Kolloquium, Tagungsband, TU Ilmenau, (2002), 345-346.

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