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PARAMETRIC SYNTHESIS OF LINKAGE MECHANISMS BY DYNAMIC SYSTEMS

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Summary: *The paper deals with the application of a new parametric synthesis method for the parametric synthesis of planar linkage mechanisms. The new synthesis method transforms the difficult problem of parametric mechanism synthesis to the computation of equlibrium of a special dynamic system that is equivalent to the formulation of parametric synthesis problem.*

1. Introduction

Mechanical synthesis is necessary method that is used during design of mechanism. This method gives the optional kinematical parameters of a designed mechanism. Solution of such difficult task usually requires large amount of iterations. The current applied methods are either very specific for simple mechanisms or they are based on iterative solution of kinematical description of mechanism motion in certain limited number of so called precision points or they are based on more general methods of optimization approaches, recently using evolutionary methods like genetics algorithm (Haug, 1984).

The general synthesis methods seem to be enough powerful and to find the solutions for all problems. They are based on performing mechanism synthesis rely on an attempt to redefine the dimensions of the system in such a way that a deviation from the desired behavior is minimized by the use of optimization methods. However, all current methods suffer from two related problems. The first problem is that the proposed dimensions of the mechanism being synthesized do not allow the mechanism assembly in all positions required for the desired motion. The second problem is that if a mechanism's synthesis iteration fails for certain parameter because of constraint and/or assembly violation the whole knowledge from this iteration is lost. The solution of the first problem has been proposed by the usage of time-varying dimensions during motion of mechanism's dimension iteration (Hansen, 2002). However, this scheme requires large amount of iterations. This insufficiency has been overcome by the approach based on nonlinear dynamic or nonlinear control (Valášek, Šika, 2004). This approach reformulates mechanism's synthesis as a nonlinear dynamic problem or nonlinear control problem. It showed that nonlinear dynamic or nonlinear control could be used for kinematical synthesis of a track of a guiding mechanism.

This paper deals with kinematical synthesis of planar linkage mechanism. The objective of this article is to show the features of the method on the synthesis with inexact solution and thus advance in the previous research of synthesis of mechanisms using parametric optimization by evolution of dynamical dissipative system (Grešl, Šika, Valášek, 2008).

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2. Synthesis of feasible mechanism

Realization of the method of geometrical synthesis is for its simplicity presented on synthesis of transmission of crank shaft mechanism. In such a case the kinematical system has only 3 dimensions to be synthesized. They are length of the crank r, length of the rod l and eccentricity e. The task of transmission synthesis is to find such dimensions of the mechanism that fulfill transmission demands. The demand for this example is that the vector of angles of crank shaft φ should correspond with vector of positions of piston h. It means that for $\varphi=\varphi_1$, $h=h_1$, for $\varphi=\varphi_2$, $h=h_2$, etc for constant dimensions of the mechanism. The picture of the crank shaft mechanism for kinematical synthesis of transmission is presented in Fig. 1.

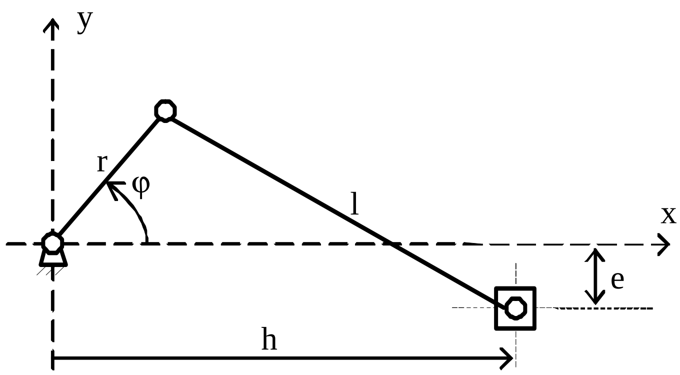

Fig. 1 Original mechanism

As it has been mentioned this method transforms problem of kinematical synthesis into problem of solution of associated dissipative dynamical system. The solution consists of synthesis of particular precision positions (angles) of the transmission and also lengths of the particular bodies. The positions (angles) of the transmission and lengths of the bodies assemble the overall dynamical system that enables separate parameters but requires their ultimate equality.

The associated dynamical system consists of n subsystems for individual demanded positions and angles of transmission mechanism. The masses m_{Ai}, m_{Bi} are introduced in points A_i, B_i. The interactions between the subsystems are ensured by forces and linear spring nature. The nonzero force acts into relevant masses whenever the corresponding dimension differs between subsystems i and j (i,j = 1,2,….n). The stabilization of the whole system is ensured by damper elements between masses and inertial frame according to sky-hook idea (Karnopp, Crosby, Harwood, 1974) or (Valášek, et al, 1998). The idea of the transformation is presented in Fig. 2.

As it has been mentioned, one kinematical system has n subsystems, where n is the number of demanded synthesized configurations. To completely and clearly introduce the force system among the subsystems let's take only three subsystems 1, 2 and 3 represented by three demanded angles of crank shaft φ_1, φ_2 and φ_3 with corresponding positions of the piston B h_1, h_2 and h_3. The initial lengths r, l and e are equal to random numbers. The idea is illustrated in Fig. 3.

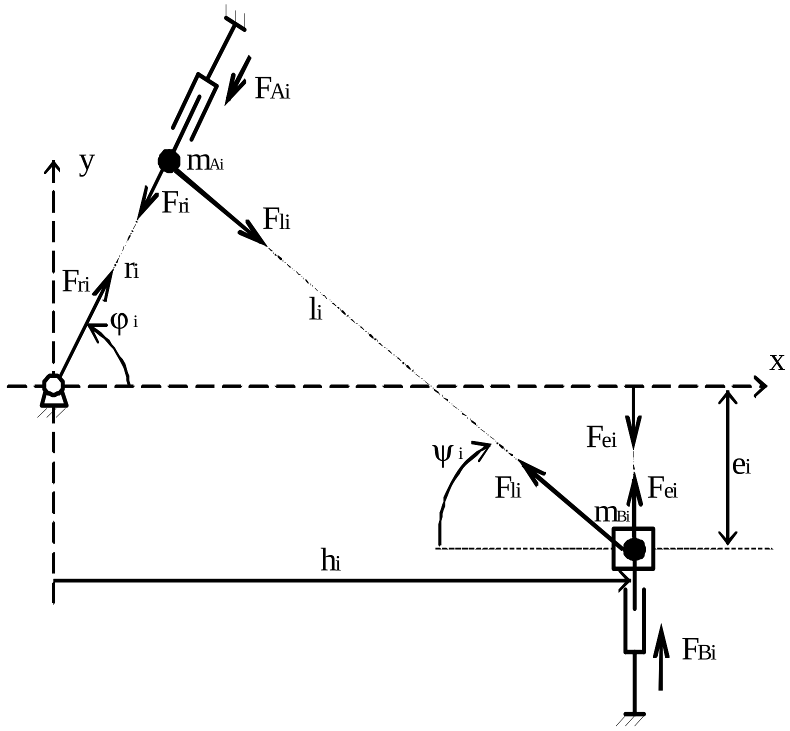

Fig. 2 Associated dissipative system

The synthesized dimensions of subsystem 1 seem to be the longest of the other corresponding dimensions. The dimensions of subsystem 2 seem to have the middle lengths. And the dimensions of the subsystem 3 are the smallest to the corresponding dimensions of the other subsystems. Yet the masses of the subsystem with the smallest dimensions are forced by forces that correspond to the difference between the dimensions of the other subsystems multiplied by some stiffness factor. The masses of the subsystem with the middle distances are forced form on one hand by positive forces that correspond to the difference between the dimensions of subsystem with smaller distances and on the second hand by negative forces that correspond to the difference between the dimensions of subsystem with greater distances. All the differences are again multiplied by some stiffness factor. Hence, it’s clear that the masses of subsystem with the middle distance are in sum forced by really small forces, because of subtractions of positive and negative forces. At the end, the masses of the subsystem with the longest dimensions are forced by forces that correspond to the difference between the dimensions of the other subsystems multiplied by some stiffness factor. The nonzero forces act among relevant masses whenever the corresponding dimensions differ among subsystems. The forces that damp the system according to sky-hook idea are introduced in the Fig. 3 as forces with dampers.

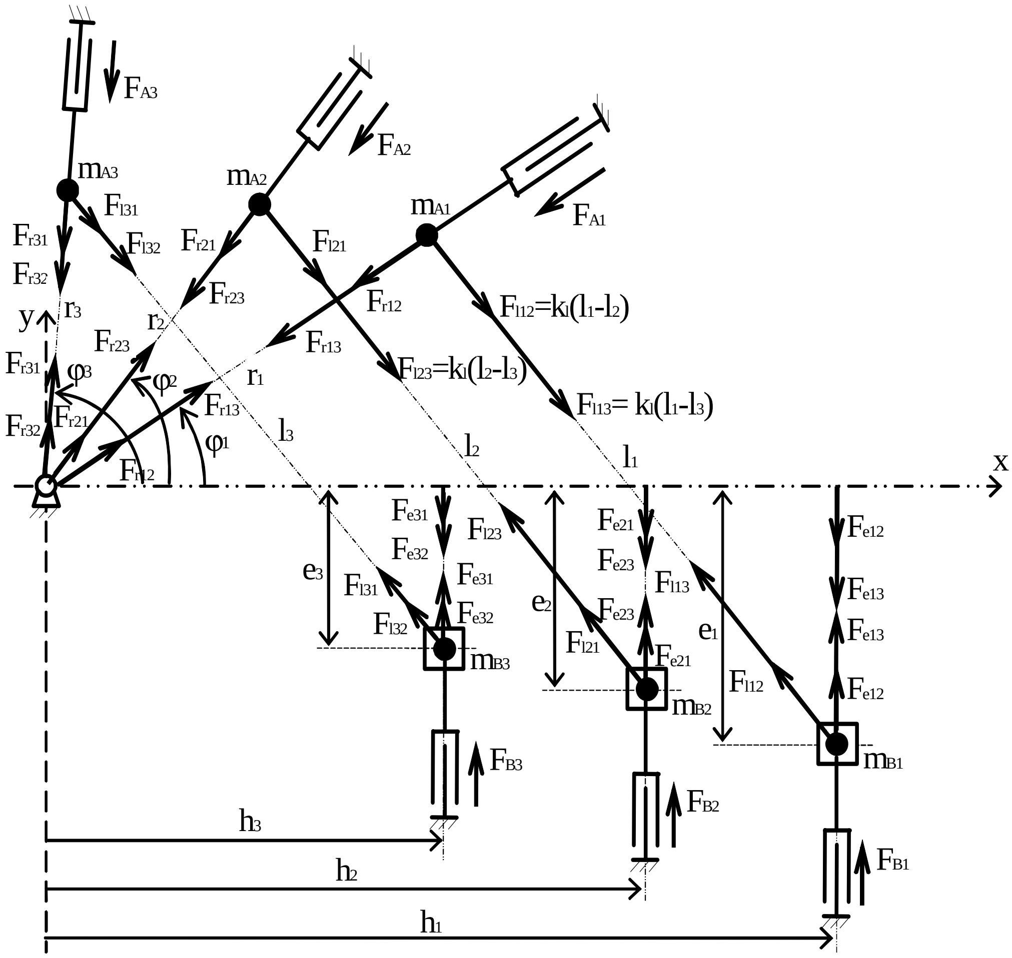

Fig. 3 Associated dissipative system with three subsystems

In order to describe general transmission synthesis of this mechanism, let's take again i=1,2,…,n demanded transmissions and thus n demanded subsystems.

Forces that act in the dynamical subsystem i are in such a general case as follows

$$F_{li} = \sum_{j=1}^{n} k_l(l_i - l_j); F_{ri} = \sum_{j=1}^{n} k_r(r_i - r_j); F_{ei} = \sum_{j=1}^{n} k_e(e_i - e_j) \quad . \qquad (7)$$
$$F_{sAi} = b_{sA}\dot{s}_{Ai}; F_{yBi} = b_{yB}\dot{y}_{Bi}$$

The final dynamical equations for mass particles in the points A and B for dynamical subsystem i are as follows

$$m_A\ddot{s}_{Ai} = \sum_{j=1}^{n}\left[k_l(l_i - l_j)cos(\varphi_i - \psi_i) - k_r(r_i - r_j)\right] - b_{sA}\dot{s}_{Ai}$$
$$m_B\ddot{y}_{Bi} = -\sum_{j=1}^{n}\left[k_l(l_i - l_j)sin(\psi_i) + k_e(e_i - e_j)\right] - b_{yB}\dot{y}_{Bi} \quad , \qquad (8)$$

where integrated coordinates are $\ddot{s}_{Ai}$ and $\ddot{y}_{Bi}$. This means that the mass A is allowed to move only in the direction of the crank shaft (corresponds to dimension r) and the mass B is allowed to move only in the direction y of the coordinate system.

The simulation of associated dynamical system has been realized within Matlab-Simulink. The system coordinates (s_{Ai}, y_{Bi}, l_i, i=1,2,…,9) for all subsystems come to stay on equilibrium values (Fig. 4). These equilibrium values can be interpreted as searched parameters of mechanism (s_A=r, y_B=e and l).

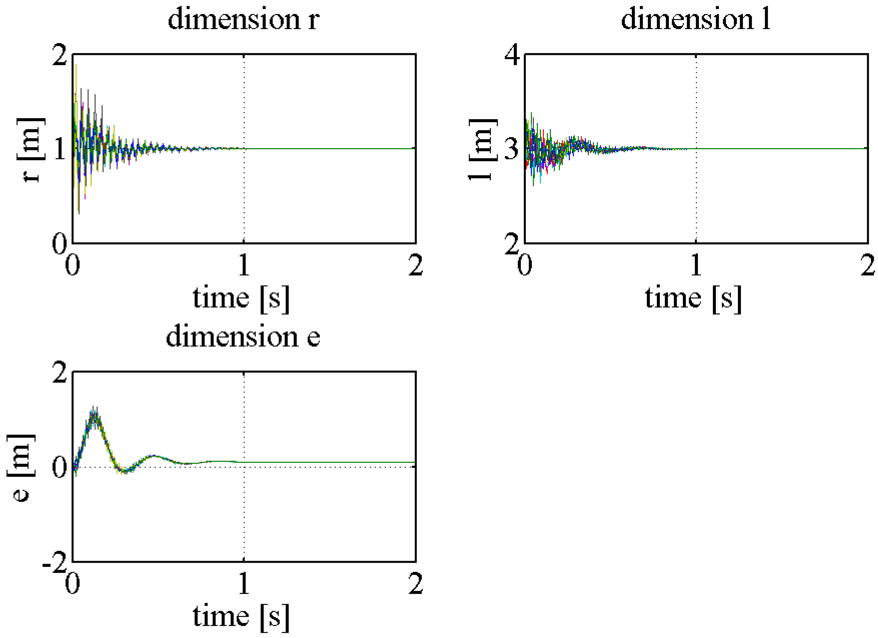

Fig. 4 Dynamical response of dimensions r, l and e

Evolution of whole structure of the crank mechanism is presented in fig. 5. Simulation started from random dimensions of the crank shaft mechanism marked as initial structure and finished in the final structure. In the final structure image, it's possible to see that the corresponding dimensions are equal.

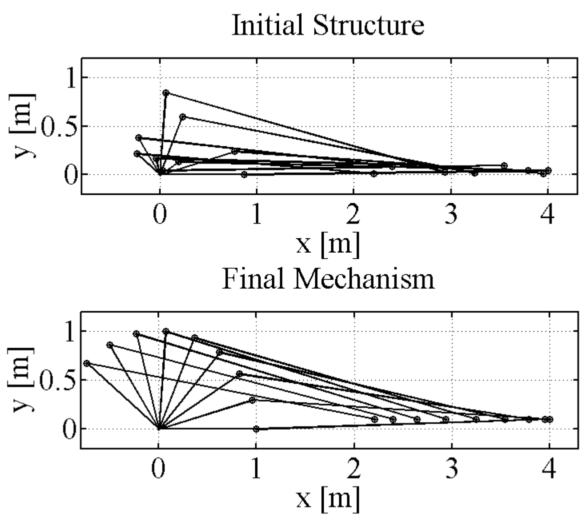

Fig. 5 Evolution of structure of the crank shaft mechanism during its synthesis

The last figure, Fig. 6 introduces how the solution fulfills the task. It looks that the mechanism with the synthesized dimensions really goes through the demanded precision points and thus it satisfies the task.

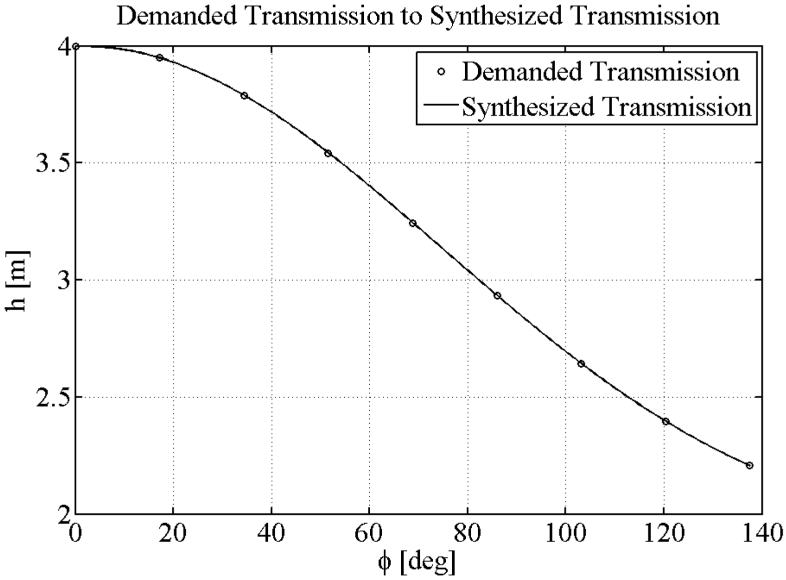

Fig. 6 Demanded and synthesized transmission of the crankshaft mechanism

3. Synthesis of unfeasible mechanism

To do the advanced test of functionality of the method, let's take synthesis of mechanism with task that has inexact solution (unfeasible mechanism). For this purpose, it's possible to take the same crank shaft mechanism with the other (random generated) demanded precision points. Such a solution is presented on Fig. 7.

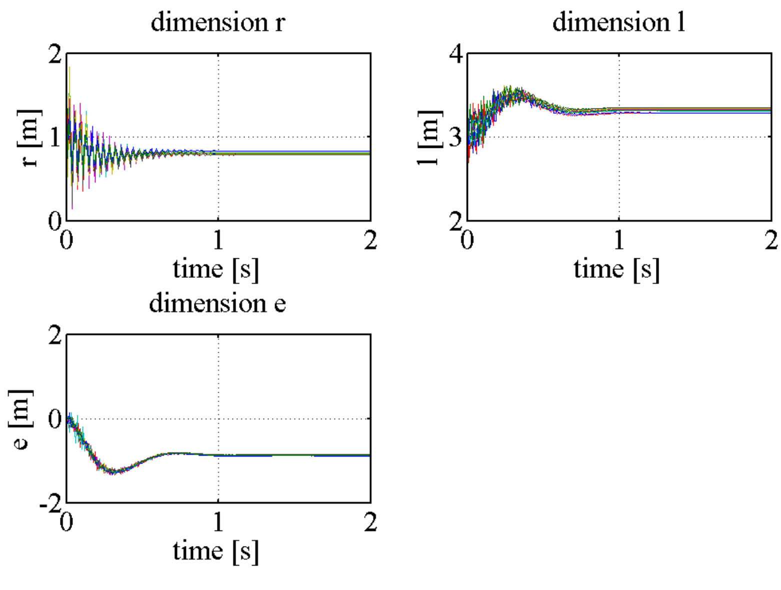

Fig. 7 Dynamical response of dimensions r, l and e of unfeasible mechanism

Its possible to see that that the system coordinates (s_{Ai}, y_{Bi}, l_i, i=1,2,…,9) for all subsystems didn't come to stay on equilibrium values. There are still small differences among the corresponding values at the end of the simulation. The parameter with the most distinctive differences is the parameter l.

Evolution of the whole structure of the crank mechanism is presented in Fig. 8. Simulation has started from random dimensions of the crank shaft mechanism marked as initial structure and finished in the final structure. In the final structure image, it's possible to see that the corresponding dimensions especially the eccentricities e differ one another.

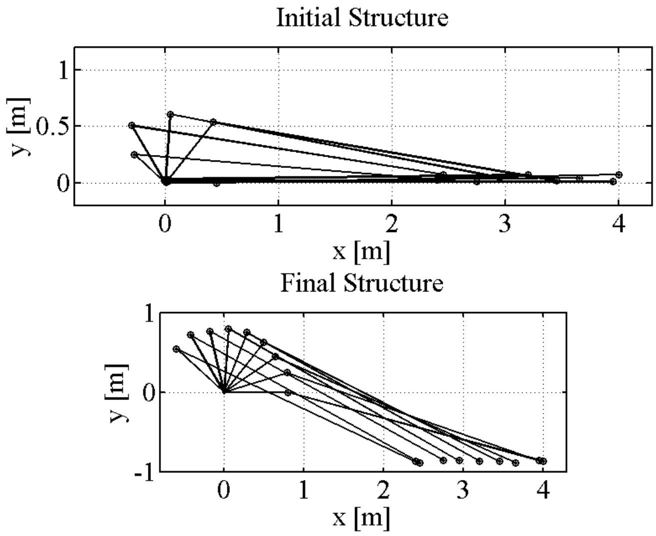

Fig. 8 Evolution of structure of the crank shaft mechanism during its synthesis of unfeasible mechanism

To see how exploit the solution of unfeasible mechanism, let's mean the synthesized parameters r_i, l_i, e_i and do the kinematical analysis. Such an analysis is presented on Fig. 9. There are two curves with demanded and synthesized transmission respectively. One could see that the synthesized transmission differs from the demanded transmission only a little. It means that is spite of inexistence of the solution of the problem, the method searches the maximal value of the target function and tends to the closest solution.

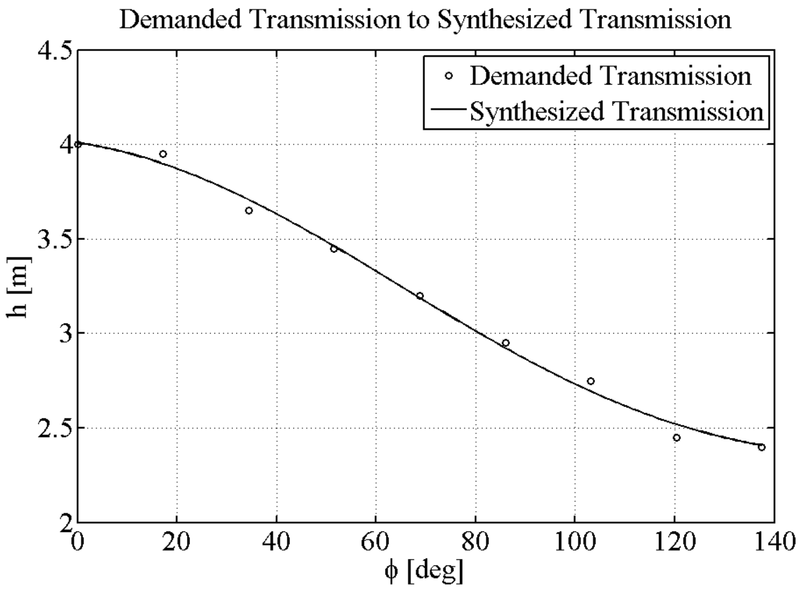

Fig. 9 Demanded and synthesized transmission of the crankshaft mechanism of unfeasible mechanism

4. Conclusion

The paper describes the new approach towards traditional kinematical (more precisely the initial geometrical) synthesis of mechanisms. The presented method transforms kinematical synthesis into the problem of the associated dynamical dissipative system. The synthesis process is realized as the time evolution of such system. The method has been tested on simple planar linkage mechanism – crank shaft mechanism. On one hand a problem of known exact solution has been done and on the other hand problem of unfeasible mechanism with inexact solution has been presented. The method is very robust and currently solved benchmarks indicate its ability for global optimization in context of mechanism synthesis.

5. Acknowledgement

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