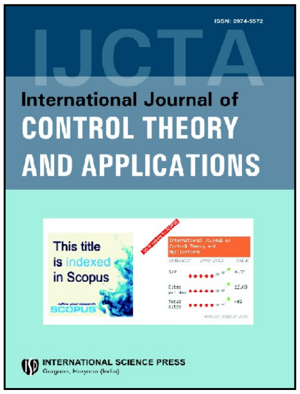

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Performance Checks of the Synthesized Function Generator

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Abstract: Once the function generator is generated it is to be analyzed for the feasibility of its functioning as a mechanism. The present paper explains the same. As an aid in finalizing initial design of kinematic chains, based on the various functions f(x) the function generator mechanism, the checks are applied. The code can be utilized as a tool in preliminary design stage for analyzing and modifying the behavior of proposed mechanism. The code is exemplified with sample function of y=1/x.

Keywords: Kinematic chains, Mechanisms, Enumeration, Analysis, Mobility.

1. INTRODUCTION

Initially Bauchabaum and Freudenstein [1] used 3 different methods to represent kinematic structure namely, functional, schematic and graphical representation. The conventional drawn cross section of the mechanism, shafts, gears and many more mechanical parts is known as Functional representation with which the visualization of the component is easy. Diagram consisting of polygons and lines representing links of different degrees, connected by small circles representing pairs/joints is known as schematic representation. The method is very handy to identify chains with small number of links. However, in linkages with large number of links of different degrees, schematic diagram becomes complicated and dissimilar diagrams may represent isomorphic kinematic chains. The method is also not useful when structural analysis and synthesis are carried out with the help of computer. In graphical representation a link is represented by a vertex, a joint by an edge and the edges are labeled according to the type of joint. It is the powerful tool as it is well suited to computer implementation , by using vertex– vertex (link–link) adjacency matrices to represent the graph.

Freudenstein and, Dobrjanskyj [2] worked on the type synthesis of mechanisms. The same type synthesis was also the research area of Woo[3]Freudenstein [4] also had the research on the theory of enumeration. Dobrjanskyj and Freudenstein [5], focused on the applications of graph theory.

The 1960s year was a “year of graph theory “as number of researchers working on the different aspects of the kinematic chains. Mruthyunjaya [6,7] focused on both structural synthesis and analysis of plane linkages. The best linkage approximation to a function occurs when the absolute value of the maximum structural error between the precision points & at both ends of the range are equalized [8] was determined by Chebyshev. There are three customary tasks for kinematic synthesis: motion, path and function generation. In function generation [9] the design requirement is that of causing an output member to rotate, oscillate or reciprocate according to a specified function of time or input member. An application of function generation would be an engine where the mixing ratios of fuel to oxidant might vary as the function y = f (x). Here f might control the fuel valve while Y would control the oxidant valve. Flow characteristics of the valves and the required ratios at various fuel rates would dictate the nature of the functional relationship to be generated.

Another example [9] is a linkage to correlate steering positions of the front wheels of an all terrain vehicle with the relative speed which each individually driven wheel should rotate to avoid scuffing. Here the input crank is connected to the steering arm, while the output adjusts a potentiometer controlling the relative speed of the two front drive wheels.

A frequent requirement in design is that of causing an output member to rotate, oscillate, or reciprocate according to a specified function of time or function of the input motion. This is called function generation. That is correlation of an input motion with an output motion in a linkage. A simple example is that of synthesizing a four-bar linkage to generate the function the function[8,9] y=f(x). In this case, x would represent the motion (crank angle) of the input crank, and the linkage would be designed so that the motion (angle) of the output rocker would approximate the function y. The problem of structural synthesis is then reduced to one of enumeration of the corresponding graphs. In the graph theory based structural synthesis, first, for each link- assortment contracted graphs representing polygonal-link patterns are generated and then for each such contracted graph vertices representing available number of binary links are added in all possible ways to obtain graphs representing final desired chains. Analytical tools from graph theory are used to aid these steps.

The authors [2,5]also showed that the number of groups of basic kinematic chains with given number of links, n and given degree of freedom, F, would depend on (n-F).

The analysis of the synthesized mechanism is necessary aspect. Structural synthesis and analysis of planar linkages and other mechanisms has been the subject of several investigations by Mruthyunjaya [6,7] . According to [10,11], the review of different approaches for the enumeration and identification of kinematic chains has been discussed.

So far the program for synthesis of function generator was not found in the literature and also the checks to be applied for the verification of the functioning of function generator is not observed.

The author in this work focuses on the synthesis and analysis of the mechanism which is explained using one sample function 1/x. The first step of analysis would be definitely visual inspection but human error affecting the results cannot be ruled out. There is thus a need for developing a fully computerized method for structural analysis which eliminates this error and further enables the designer to obtain results quickly and easily. The aim of this paper is to fulfill such a need. The author focuses on the various checks during analysis of generated function also.

The generated four bar function generator is checked for its feasibility with various parameters like

Assemblability of linkages; Revolvability of Angles and minimized structural error and are discussed.

2. ASSEMBLABILITY OF LINKAGES

For a given kinematic chain of n links ($n \geq 3$)can form an N-bar linkage, if and only if

$$l_n \leq l_1 + l_2 + \ldots\ldots\ldots + l_{n-1} \tag{1}$$

The above statement may be regarded as the “theorem of assemblability” of linkages. It is the foundation of general mobility criteria, the mobility criteria of N-bar linkages are based on the assemblability of (N-1) bar linkages.

3. MOBILITY CALCULATION

The mobility M or number of degrees of freedom (DoF) of a kinematic chain is the number of independent parameters required to specify completely the configuration of the kinematic chain in the space, with respect to one link chosen as the reference [12-14]. Mobility is used to verify the existence of a mechanism, to indicate the number of independent parameters in robot modeling and to determine the number of actuators needed to drive the mechanism [12].

3.1. Mobility criteria of single loop N-bar linkages

It is also assumed that $l_{1,} l_{2,} l_3 \ldots\ldots\ldots\ldots\ldots l_n$ are the link lengths of an N-bar linkage and $l_1 \leq l_2 \leq \ldots\ldots\ldots\ldots \leq l_{n-1} \leq l_n$

3.2. Revolvability of Angles

In an N-bar linkage an angle is revolvable if and only if it can reach both 0° and 180 °.

In an N-bar linkage the angles at both ends of link i are revolvable angles, if the sum of the lengths of link i and to longest link is not greater than the sum of other link lengths i.e.

$$l_n + l_i \leq \sum_{j=1}^{n} l_j - (l_n + l_i) \tag{2}$$

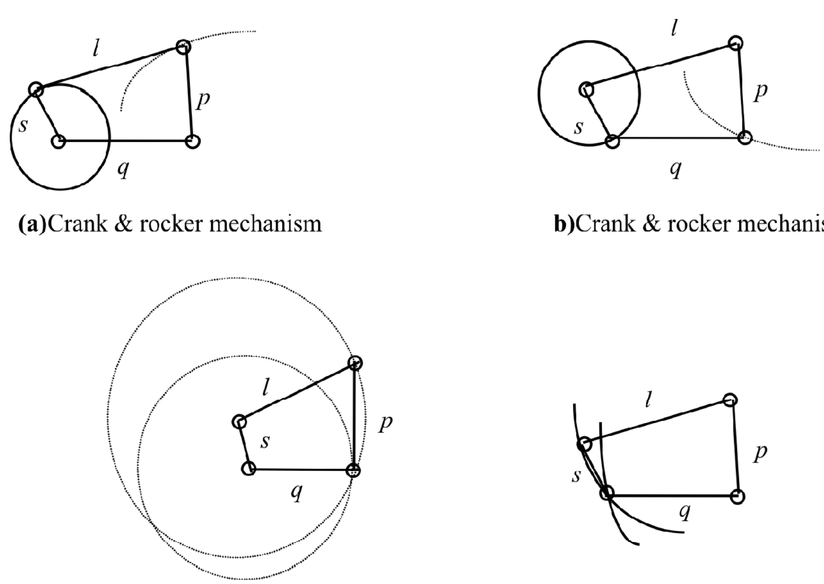
If the shorter link l_i of two adjacent links does not satisfy the above condition, the two links cannot revolve relative to each other and therefore, the angle between them is non-revolvable angle.

3.3. Four-bar linkages

If $l_4 + l_1 \leq l_2 + l_3$ the angles at both ends of the shortest link are revolvable angles and the shortest link may revolve relative to any adjacent link.

If $l_4 + l_1 > l_2 + l_3$, there will be no revolvable angle and no link may revolve relative to an adjacent link.

Hence the statement of Grashof's law is[13] *for a planar four-bar linkage, the sum of the shortest(s) and longest(l) link lengths cannot be greater than the sum of the remaining two link lengths(p,q)*, if there is to be continuous relative rotation between two members. That is $s + l \leq p + q$



(a) Crank & rocker mechanism **b)**Crank & rocker mechanism

(c) Double crank mechanism **(d)**Double rocker mechanism

Figure 5.1: Grashof's law's results

4. STRUCTURAL ANALYSIS OF THE FOUR BAR MECHANISM

In the synthesised four bar function generator the following analysis is done.

3.1 Checking the satisfaction of Grashof's criterion.

3.2 To identify the kind of linkage ie.Doublecrank, crankrocker, doublerocker mechanism , so that as per the application selection of type of mechanism will be easier.

3.3 The linkage [14] is also made to run through its range and its performance in terms of structural error, percentage structural error , mechanical advantage is found for different positions of input link.

3.4 A planar four bar linkage has only one degree of freedom . When additional degrees of freedom are needed, more links may be added and so an N- bar linkage is formed. With multiple degrees of freedom, an N- bar linkage requires multiple inputs. If these inputs are independent , the linkage may function as a programmable linkage or a robot. If these inputs are coupled , the linkage may become a single degree of freedom mechanism, which still provides more versatility and design opportunity over a four bar linkage. Hence it is important to know the rivolvability or mobility of the linkage.

3.5 The extreme velocity ratio is found at which the mechanism is jammed.

3.6 The program also calculates the rms (root mean square) value of error.

Table 1
Synthesis of the four bar function generation mechanism for the function y=1/x

Inputs to the program	*Output in termsof lengths*
O_aO_b= 100mm	By analytical method
O_aA =75mm	AB=178.58 mm;
ϕ_s = 50°	O_bB = 153.56mm
$\Delta\phi$ = 90°	By graphical method
$\Delta\Psi$ = 60°	AB=179.86 mm;
	O_bB = 152.47mm

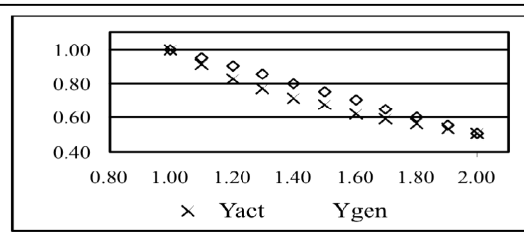


Figure 3: Comparison of actual & generated, Y=1/X function

Table 2
Output in the form of tabulated results

Xact	*Yact*	*Fi(f)*	*Si (Y)*	*Ygen*	*Error*	*Percentage Error*	*Mechanical advantage*
1.00	1.00	50.00	31.40	1.00	0.00	0.00	1.21
1.10	0.91	59.00	37.40	0.95	0.04	4.50	1.24
1.20	0.83	68.00	43.40	0.90	0.07	8.00	1.27
1.30	0.77	77.00	49.40	0.85	0.08	10.50	1.31
1.40	0.71	86.00	55.40	0.80	0.09	12.00	1.36
1.50	0.67	95.00	61.40	0.75	0.08	12.50	1.41
1.60	0.62	104.00	67.40	0.70	0.07	12.00	1.47
1.70	0.59	113.00	73.40	0.65	0.06	10.50	1.54
1.80	0.56	122.00	79.40	0.60	0.04	8.00	1.62
1.90	0.53	131.00	85.40	0.55	0.02	4.50	1.70
2.00	0.50	140.00	91.40	0.50	0.00	0.00	1.79

Calculated RMS value is 0.0179

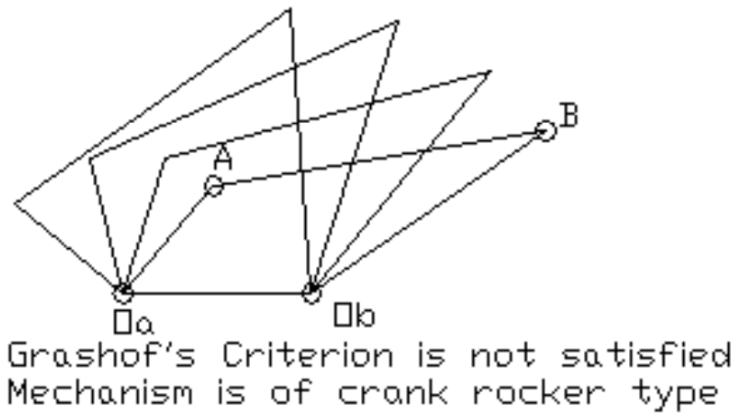


Figure 4: Synthesized four bar function generator

Comments: The Table 1, 2 and Figures 3 and 4 are output of the function y=1/x, with the initial positions 50^0 and 31.4^0 of the crank and rocker respectively will result into a mechanism which will not satisfy the Grashof's criterion and it is of crank rocker type of mechanism. The comparison between Generated function and the actual output function is shown in fig. 3. The error at different crank positions are given in Table 2.

5. COMPUTER ALGORITHM

A unified computer program was developed for the kinematic synthesis of four bar function generator which was explained for the function Y=1/X also the linkage is analysed with the animation . Given the input of Df & Dy , input link (crank) length , fixed link length & f_S the program in 'C' language first calculates & prints the link lengths followed by the animation of mechanism for the results of analysis presented in section 4 The algorithm of the program is as follows :

Step 1 : Read the input variables

Step 2 : All initial calculations are done including precision points.

Step 3 : Find the coordinates of the 4-bar linkage by the analytical or graphical method or by both the methods.

Step 4 : Find the link lengths of the mechanism.

Step 5 : Find the error followed by the RMS value

Step 6 : Find the minimum value of RMS & select that mechanism .

Step 7 : Find the mechanical advantage, percentage of error & Tabulate the values

Step 8 : For few cases where crank doesn't make complete revolution locate the jammed position or last point upto which the mechanism moves.

Step 9 : Plot the mechanism / linkage

Step 10: Check the satisfaction of Grashof's Criterion

Step 11: Identification of the types of the mechanism i.e. double crank, double rocker, crank rocker, output link is the shortest by animating the mechanism for the given input angles.

6. CONCLUSION

With the use of software the mechanism can be plotted and simulated or animated for the input range, which will assist the designer for faster selection of four link mechanism as it is the basic mechanism required for all the applications. The present work emphasizes on the basic aspects of feasibility checks which are must for the analysis of synthesized four bar function generator mechanism. The analyzed results will guide the designer to decide about the constraints to be applied on mechanism. The computer program can be used in iterative manner by varying the initial input parameters , viz frame length (OaOb) , crank length (OaA) , initial position of crank (f_s), range of variation of input crank (Df) & output link (Dy) . The error table is calculated for each run and RMS error is estimated thereof. By setting the cutoff value of permissible error, the optimised output may be obtained. The function y=1/x is used as a sample function for the discussion of various outputs of the code.

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