Mechanism and Machine Theory 44 (2009) 445–456

Contents lists available at ScienceDirect

Mechanism and Machine Theory

journal homepage: www.elsevier.com/locate/mechmt

Tolerance analysis of flexible kinematic mechanism using DLM method

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ARTICLE INFO

Article history:
Received 24 July 2007
Received in revised form 13 January 2008
Accepted 9 March 2008
Available online 12 May 2008

Keywords:
Tolerance Analysis
Assembly specification
flexible kinematic mechanism

ABSTRACT

Due to the fact that there are unwanted variations in manufacturing processes, all parts produced in industry are not manufactured with nominal or exact dimensions. Therefore for each part dimension, a tolerance limit is prescribed. Also for all assemblies, a limit of variation is prescribed for a specified parameter of the assembly which is referred to as the assembly specification, and it could be the position of a point, a gap or geometry tolerance of a feature in the assembly. As a matter of fact, the performance of the assembly is measured by accuracy of the assembly specification which is a function of associated part tolerances. If the assembly specification has limits of variation in two or more directions, the correlation between these variations also impresses the limit of variation. To determine the bivariate distribution of the assembly specification, in terms of part tolerances, the direct linearization method (DLM) is used. In this paper, the coupler point (C.P.) position of a four-bar mechanism during one cycle of motion is considered as the assembly specification. The mechanism consists of flexible parts and is subjected to external loading. The extra variations of each part dimensions due to its flexibilities will impose new variations for the assembly specification. The influence of loading on variation of the assembly specification is modeled by the finite elements method (FEM) using CALFEM toolbox of MATLAB software. First, the valid domains of DLM are recognized by means of Monte Carlo simulation and then, the percent contribution of each manufacturing variable in assembly specification is determined by DLM method. The simulation results confirm that after loading the mechanism, mass production rejects are remarkably increased. The paper proposes that by decreasing the tolerance limits of those manufacturing variables that have the highest contribution in the assembly specification, the number of rejects could be decreased significantly.

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1. Introduction

An important aim in designing kinematic linkages is creating an accurate path by means of a point on the coupler. This point and the corresponding path are called coupler point (C.P.) and coupler point Path, respectively. In each cycle of motion, manufacturing tolerances of the parts and also extra variations due to flexibilities and loading cause a deviation in the C.P. path from its designed or ideal state. These deviations can lead to undesirable performance of the mechanism. There are several methods which were proposed to determine the effect of part tolerances on C.P. path deviations or the performance of the mechanism. The direct linearization method (DLM) is firstly presented by Marler [1]. This method has been extended by Parkinson and Chase for static structures and kinematic mechanisms [2]. However, they assumed that all components are rigid. Markley presented a method to analyze assemblies with flexible parts [3]. He used the linear elastic assumption for contact of two parts which was proposed by Francavilla and Zienkiewicz [4]. He also presented a method to determine

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doi:10.1016/j.mechmachtheory.2008.03.010

the variance and mean value of a dimension under loading. The current work implemented FEM model of kinematic mechanisms and investigated variations of assembly specification during one cycle of motion under external loading.

In Section 2 of this paper, the kinematic model of a four-bar mechanism including tolerances of manufacturing variables is expressed. In Section 3, the direct linearization method is demonstrated and the equations of vector loops, sensitivity matrix and position error are obtained. In the following section, the DLM method is applied to find the bivariate distribution of the C.P. position error. The valid domains of the DLM are determined by means of Monte Carlo simulation in Section 5. The four-bar mechanism with external loading is modeled by FEM and the variations of the maximum normal-to-path error are estimated during one cycle of motion. The percent contribution of manufacturing variables in the assembly specification is then determined. The number of rejects is also estimated from standard normal tables and reported in Section 6. Finally, Section 7 presents the conclusions of the paper and offers some courses of action for the future work.

2. Four-bar mechanism model

In the current work, the C.P. position error of a four-bar mechanism is analyzed, (see Fig. 1). The reference path of C.P. is generated by assuming nominal dimensions for all components.

For each component of the mechanism, the manufacturing tolerances are specified on the basis of corresponding manufacturing processes [5]. Nominal dimensions and tolerances of each component are reported in Table 1.

Angular position of link 2 (θ_2) is considered as an input to the mechanism. Therefore, it is not a manufacturing dimension and zero tolerance is assigned. All manufacturing dimensions are assumed to be normally distributed with a mean equal to the nominal link length. Also, the acceptable limit of distribution is taken according to common standard of 3σ.

3. Direct linearization method (DLM)

The direct linearization method (DLM) can be used to determine the position error of a kinematic linkage [6]. In this method, the sensitivity matrix is derived using open and closed vector loops. The error can be predicted by applying statistical approaches. The nominal position of C.P. is found by solving the following closed and open vector loop equations. These two vector loops are demonstrated in Fig. 2.

$$\vec{h} = \vec{r}_1 + \vec{r}_2 + \vec{r}_3 + \vec{r}_4 = 0 \tag{1}$$

$$\vec{C}.P. = \vec{r}_2 + \vec{r}_P \tag{2}$$

Eqs. (1) and (2) can be represented in terms of their components in x and y directions as follows:

$$h_x = r_2 \sin(\theta_2) + r_3 \sin(\theta_3) - r_4 \sin(\theta_4) - r_1 \sin(\theta_1) \tag{3}$$

$$h_y = r_2 \sin(\theta_2) + r_3 \sin(\theta_3) + r_4 \sin(\theta_4) - r_1 \sin(\theta_1) \tag{4}$$

$$(\text{C.P.})_x = r_2 \cos(\theta_2) + r_p \cos(\theta_3 + \beta) \tag{5}$$

$$(\text{C.P.})_y = r_2 \sin(\theta_2) + r_p \sin(\theta_3 + \beta) \tag{6}$$

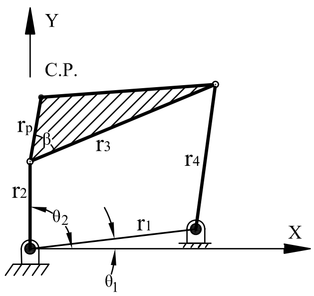

Fig. 1. Four-bar mechanism with driving crank.

Table 1
Nominal dimensions and tolerances of manufacturing variables

Manufacturing variables	r_1	r_2	r_3	r_4	r_p	β	θ_1 (deg)	θ_2(deg)
Nominal dimensions (mm)	400	250	600	470	104	80°	0°	0–360°
Tolerances (mm)	±0.2	±0.2	±0.6	±0.5	±0.2	±0.5°	±0.5°	0

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