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Effect of Rocker Length on the Dynamic Behavior of a Coupler Link in Four Bar Planar Mechanism

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Abstract

It is well-known that the dynamic analysis of mechanisms operating at high speed cannot neglect the effects of link elastic flexibility. In fact this effect may affect the dynamic response of the output link motion, so that the mechanisms may fail to perform their assign tasks effectively. The dynamic analysis of high speed mechanism having rigid rocker length is carried out by using Finite Element Method (FEM) and the same is discussed in the present work. Moreover, the behavior of damped-flexible coupler under varying length of rigid rocker length is analyzed. Modeling and simulation of mechanism has been analyzed by using ANSYS and results are found to be in agreement with the experimental result mention in literature. It is observed that increase in length of rocker link gives rise in the strain value in middle of the coupler link and hence the length of rocker link should be kept as minimum as possible.
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Keywords: Flexible linkage; FEM; Effect of length of rocker link; Strain in coupler link; Simulation

1. Introduction

Generally, all considerations in the force analysis of mechanisms, whether static or dynamic, the links are assumed to be rigid. The complexity of the mathematical analysis of mechanisms with elastic links has been a deferent against giving up the rigidity assumptions. The area of research pertaining to the motion of mechanisms,

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with consideration of link elasticity and mass distribution has been called kineto-elastodynamics of mechanism. The requirement for machines to run at higher speed brought to the surface many problems, such as balancing and vibrations, which were not serious factor at lower speeds. Therefore the problem they were facing was now to run machines at higher speeds with lower power consumption. New method to understand the elastic behavior of mechanisms was needed. The researcher interest is the development of preferably simple mathematical model which is able to simulate the dynamic behavior of the mechanisms. Since this model make it possible to relate the design and operational parameters of mechanisms to their actual dynamic behavior.

A G Erdman et al. [1] carried out a survey on different types of analyses of mechanism. Imdad Imam et al. [2] have made a notable contribution by considering the elastic coupler link and also studied the effects of elastic deformation and mass distribution of the links. A M Vaidya et al. [3] studied the joint clearance on kinematics of mechanism and bearing stiffness along with links flexibility on modal analysis at higher frequency. A scheme for the position and vibration control of a four-bar linkage with all the links flexible has been designed and tested, both numerically and experimentally by A Trevisani [4]. D A Turcic et al. [5, 6] emphasised on the need for including vibration effects; these, until recently, had generally been neglected, primarily for reason of the complexity of the mathematical analysis of such systems. Finite element theory was used to facilitate the modelling of elastic mechanisms. Formulation of the problem is a linear ordinary second order differential equation with the variable coefficients. Spectrum analysis of the dynamic strain response is obtained theoretically representing the steady state solution by Fourier series. A favourable comparison between the theoretical and experimental results have also analysed by V Masurekar at el. [7].

2. Finite element model

In the analysis of the planar four bar elastic mechanisms at a given position, it is assumed that it is a structure composed of discrete members. In this respect, each of the constituent members is regarded as a beam and so the beam theories of bending apply. The effect of shear deformation and rotary inertia are neglected.

3. Elastic beam element in plane

A general beam element, representing a link of a mechanism, is shown in Fig. 1, without the other links. There are two reference frames; (i) The fixed (OXY) frame and (ii) The rotated (Oxy) frame and both having a common origin O. The x axis of the rotated frame is always parallel to the rigid body position of the beam element axis throughout its motion. The beam element is shown by dotted lines is elastically deformed position and its rigid body position by full lines. The elastic deformations of the beam element (link) are completely described by the six generalized nodal displacement coordinates by u_1 to u_6, shown in Fig. 1. These displacements shown in their positive directions with reference to the rigid body position of the beam (link), in Fig. 1 also locate the deformed position P' and Q' of the end points P and Q.

By consider the link as a beam element in plane motion. We get the accelerations vectors as follows:

$$\{\ddot{u}_a\} = \{\ddot{u}_r\} + \{\ddot{u}\} + \{a_n\} + \{a_c\} + \{a_t\} \tag{1}$$

where, the vectors from left to right represent the absolute, rigid body, generalized relative to the rigid body position of link, normal, coriolis and tangential accelerations respectively.

In Equation (1), the product terms in vectors $\{a_n\}$, $\{a_c\}$ and $\{a_t\}$ are considered small compared to corresponding terms $\{\ddot{u}_r\} + \{\ddot{u}\}$. When it is neglected, equation may be modified as,

$$\{\ddot{u}_a\} = \{\ddot{u}_r\} + \{\ddot{u}\} \tag{2}$$

Similarly, it may be shown that

$$\{\dot{u}_a\} = \{\dot{u}_r\} + \{\dot{u}\} \tag{3}$$

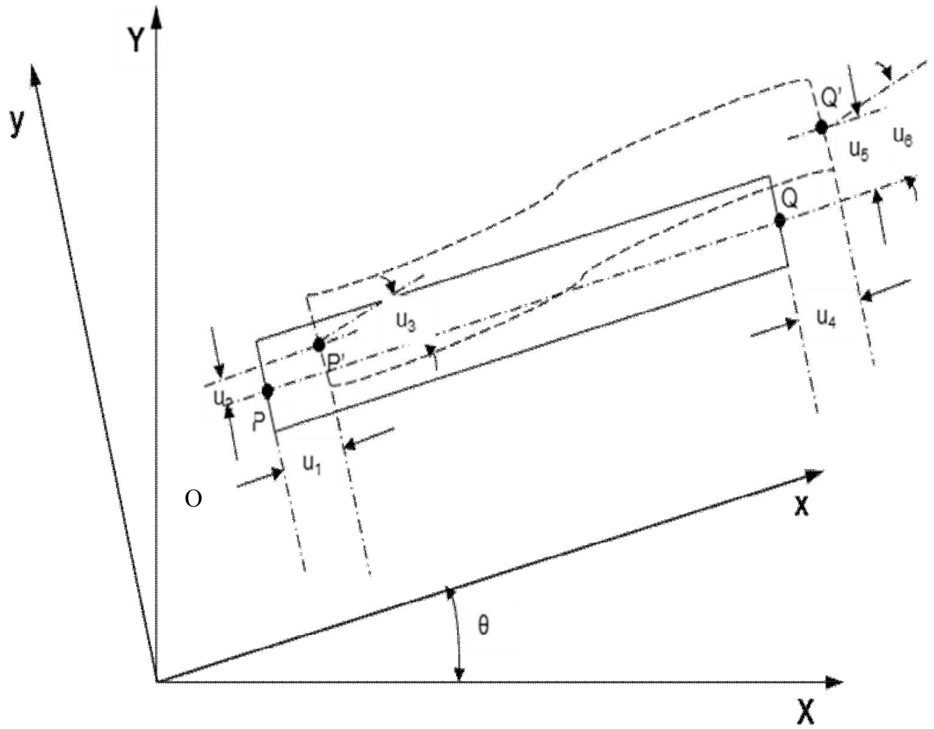


Fig. 1. Rigid and elastic body with coordinate systems

The unknown displacement field within an element will be interpolated by a linear distribution. This approximation becomes increasingly accurate as more elements are considered in the model. To implement this linear interpolation, linear shape functions will be introduced.

4. Mass and stiffness matrices of element

The equation of motion of the elastic beam element in Fig. 1 may be described by Lagrangean's equation.

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{u}_i}\right) - \left(\frac{\partial T}{\partial \dot{u}_i}\right) + \left(\frac{\partial U}{\partial \dot{u}_i}\right) = \overline{Q}_i, \quad i = 1, 2, ..., 6 \tag{4}$$

where, $\overline{Q}_i$ are the generalized forces acting in the direction of generalized coordinators and have no potential.

Considering the strains associated with the displacement functions and neglecting those due to temperature variations and any strain initially present, the strain energy in matrix form is

$$U = \frac{1}{2}\{u\}^T\left[\overline{k}\right]\{u\} \tag{5}$$

The mass and stiffness matrices [5] can be written as,

$$[\overline{m}] = \rho AL \begin{bmatrix} \frac{1}{3} & & & & & \\ 0 & \frac{13}{35} & & & & \\ 0 & \frac{11L}{210} & \frac{L^2}{105} & & & \\ \frac{1}{6} & 0 & 0 & \frac{1}{3} & & \\ 0 & \frac{9}{70} & \frac{13L}{420} & 0 & \frac{13}{35} & \\ 0 & \frac{-13L}{420} & \frac{-L^2}{140} & 0 & \frac{-11L}{210} & \frac{L^2}{105} \end{bmatrix} \qquad [\overline{k}] = \begin{bmatrix} \frac{EA}{L} & & & & & \\ 0 & \frac{12EI}{L^3} & & & & \\ 0 & \frac{6EI}{L^2} & \frac{4EI}{L} & & & \\ \frac{-EA}{L} & 0 & 0 & \frac{EA}{L} & & \\ 0 & \frac{-12EI}{L^3} & \frac{-6EI}{L^2} & 0 & \frac{12EI}{L^3} & \\ 0 & \frac{6EI}{L^2} & \frac{2EI}{L} & 0 & \frac{-6EI}{L^2} & \frac{4EI}{L} \end{bmatrix}$$

With the help of Lagrange's equations, the equation of motion for the beam element may be derived as,

$$[\overline{m}]\{\ddot{u}_a(t)\} + [\overline{k}]\{u(t)\} = \{\overline{Q}\} \tag{6}$$

5. Assembly of the element

In previous section we formulate the mass and stiffness matrices for the element in their local coordinates system. Now system matrices for the whole mechanism are found by define only one global coordinate system for a given mechanism. In four bar mechanism of Fig. 2, in which for the analysis purpose each link is to be consider as an element and joints of all links are to be consider as node. For the finite element analysis prepare the stiffness and mass matrix for all elements and these elements mass and stiffness matrixes are systematically superposed to develop the stiffness and mass matrix of the mechanism. Afterwards solve the coupled differential equation of motion with help of modal analysis. Solutions of equation give a displacement of each element at particular at that instant. With the help of displacement calculate the strain and stress at desire point on links.

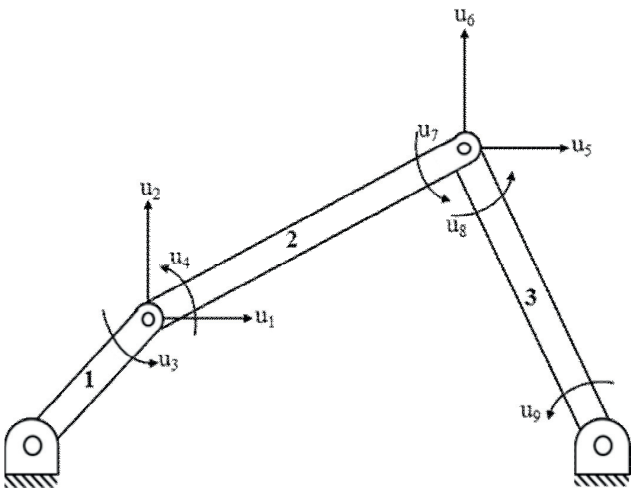


Fig. 2. Four bar mechanism with displacement and three elements

If the system oriented coordinates $\{U\}$ are used as the generalized coordinates to describe the structural deformation of the linkage from its rigid body position, as in Fig. 2. In first, second and the third terms on the left hand side of Lagrange's equation is reduce to $\sum_{j=1}^{9} M_{ij}\ddot{U}_{aj}$, 0 and $\sum_{j=1}^{9} K_{ij}U_j$ respectively.

In matrix form, the equations of motion can be written as,

$$[M]\{\ddot{U}_a\}+[k]\{U\}=\{Q\} \quad (7)$$

In the absence of damping forces in the mechanism and of external forces on the follower, the equation of motion may be written in matrix form as below

$$[M]\{\ddot{U}\}+[K]\{U\}=-[M]\{\ddot{U}_r\} \quad (8)$$

6. Solution of equation of motion

The system is excited by a periodic forcing function. The method is use for solving nine couple differential equations are modal analysis. Application of modal analysis for linear system only requires symmetry of the stiffness and mass matrix. In coupling equation first step is to determine the natural frequencies and mode shapes. When the principle coordinates are used as dependent variables, the governing differential equations are uncoupled. Each equation can be solved by any appropriate method.

7. Design problem

Specifications of four bar linkage for analysis are shown in Table 1.

Table 1. Dimensions of four bar mechanism

Parameters	Fixed link (1)	Crank (2)	Coupler (3)	Follower (4)
Length (mm)	250	110	280	260
C/S Area (mm^2)	-	108	40	40
Area moment of Inertia (mm^4)	-	160	9	9
Density = 2770 kg/m^3			Crank speed = 308.44 rpm	

8. Results and Conclusions

Modeling and simulation has been done in ANSYS software. Analysis type selected for four bar planar mechanism is transient with ANSYS Mechanical solver. Only coupler link to be considered as flexible and remaining all links is to be considered as rigid. Meshed modeled of four bar mechanism as shown in Fig. 3. Non linear effect of material properties and thermal strain effect should be included in analysis.

Meshing has been done with Quad/Tri element by taking 105 elements with 927 nodes. Simulation had done for 0.2 seconds in two steps in 500 sub steps by keeping large deflection on. Connections of Joint are to be considered as a rigid. Properties of material are to be taken for analysis are shown in Table 2.

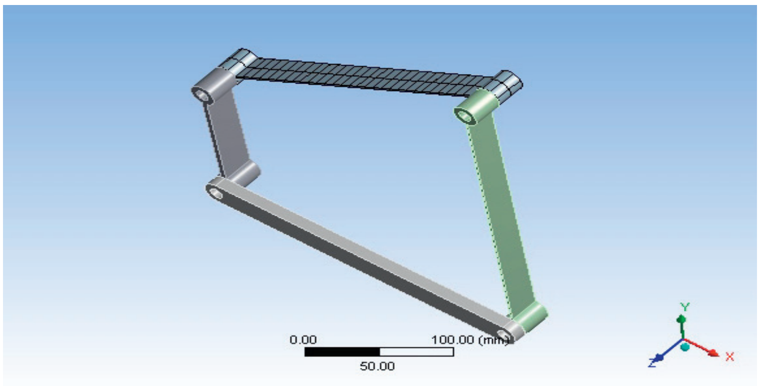


Fig. 3. Meshed model of four bar mechanism

Table 2. Material properties of link

Temperature	Young's Modulus	Poisson's Ratio	Bulk Modulus	Shear Modulus
22 C	71000 MPa	0.33	69608 MPa	26692 MPa

Results of analysis as depicted in Fig. 4 has been compared with experimental result mention in literature [6] and found to be closer by comparing the value of root mean square of experimental and ANSYS results given in Table 3.

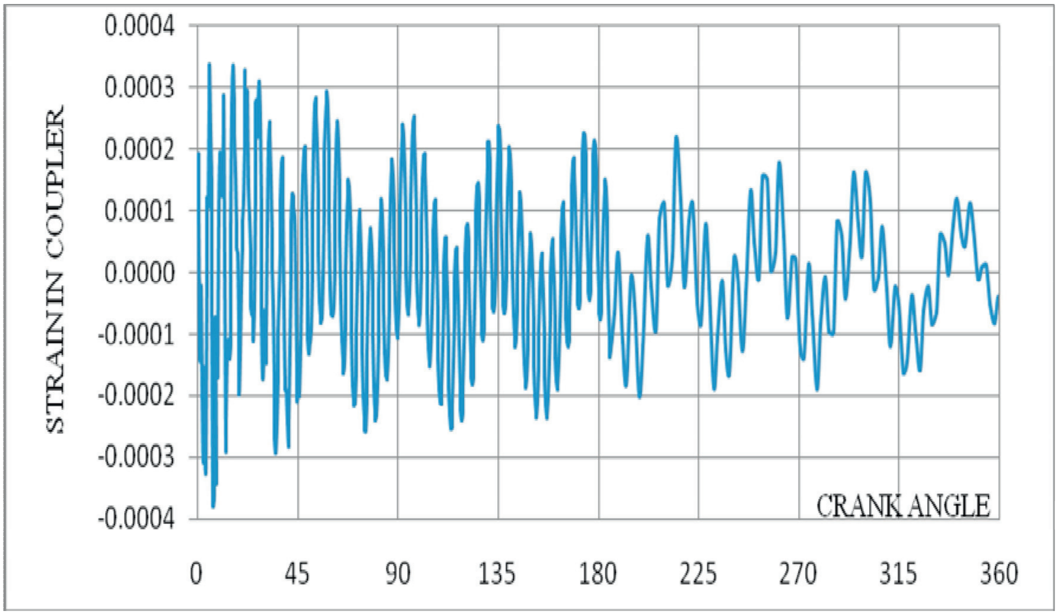
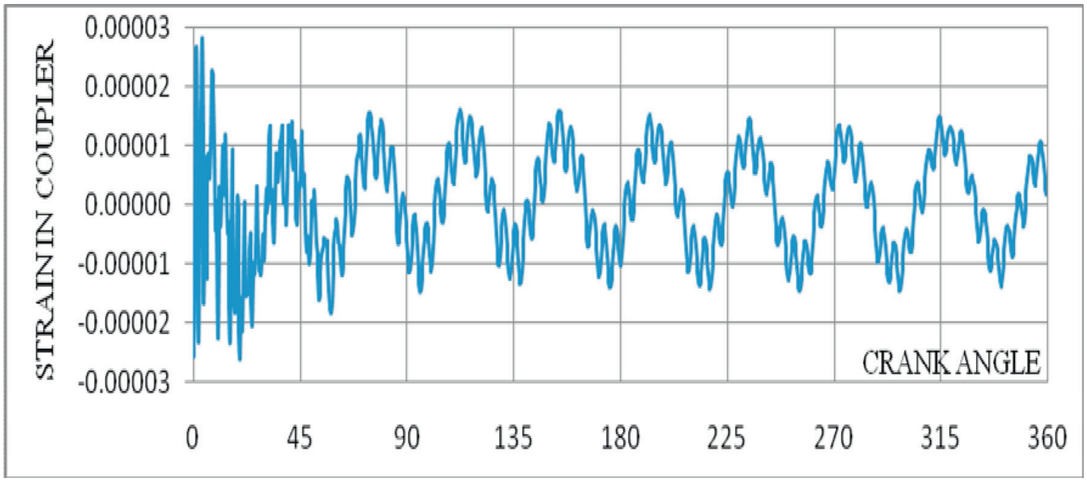


Fig. 4. Results of analysis

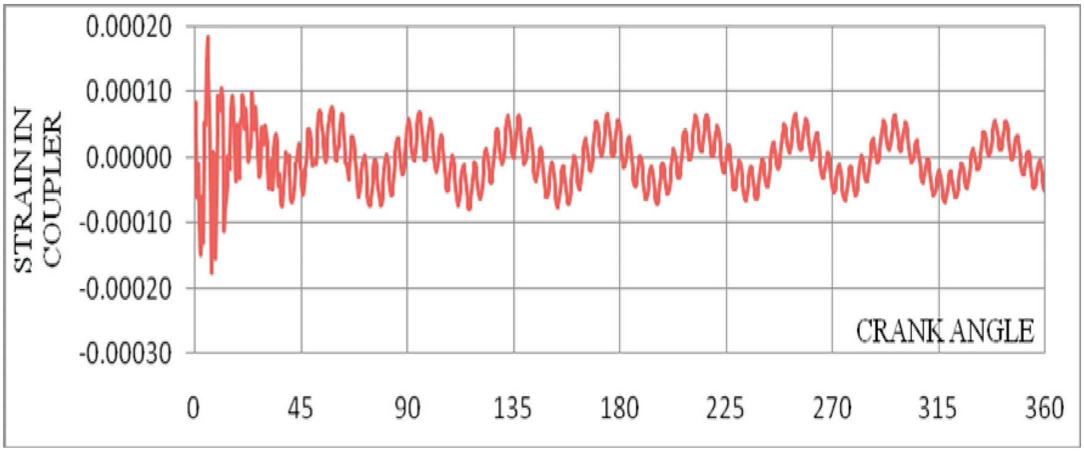
Table 3. Root Mean Square of strain in coupler link (Rocker length = 260 mm)

	Experimental Results	ANSYS Results
RMS Value	0.000134	0.000138

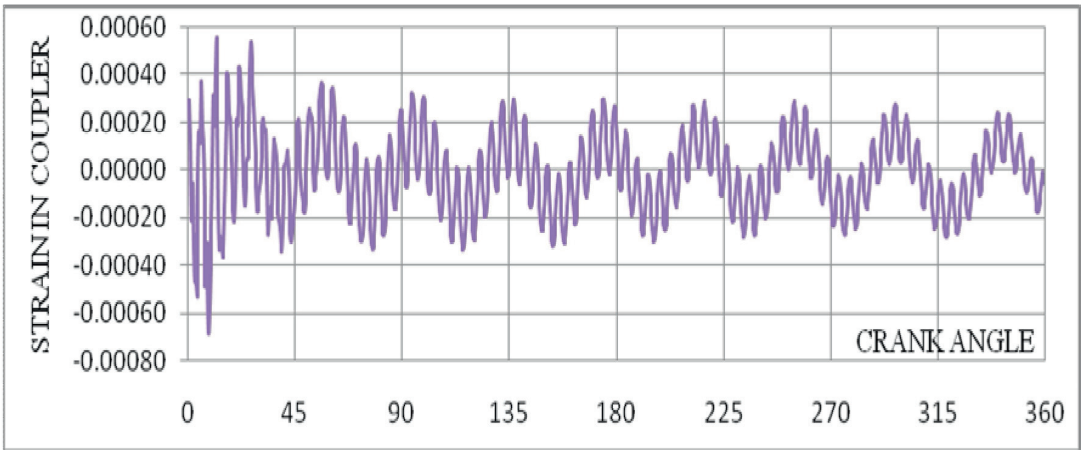
After simulation results are validated with experimental results work has been extended to see the effect of length of rocker link on flexible coupler. By taking all the variable of analysis and dimensions of link are kept constant (expect the length of rocker link), analysis carried out for the length 240, 250, 260, 270, 280 mm of rocker link. The strain in coupler link for different length of rocker at different crank angle is represents in the form of a graph as follow in Fig. 5.


0.00003
0.00002
0.00001
0.00000
-0.00001
-0.00002
-0.00003
STRAININ COUPLER
CRANK ANGLE
0
45
90
135
180
225
270
315
360

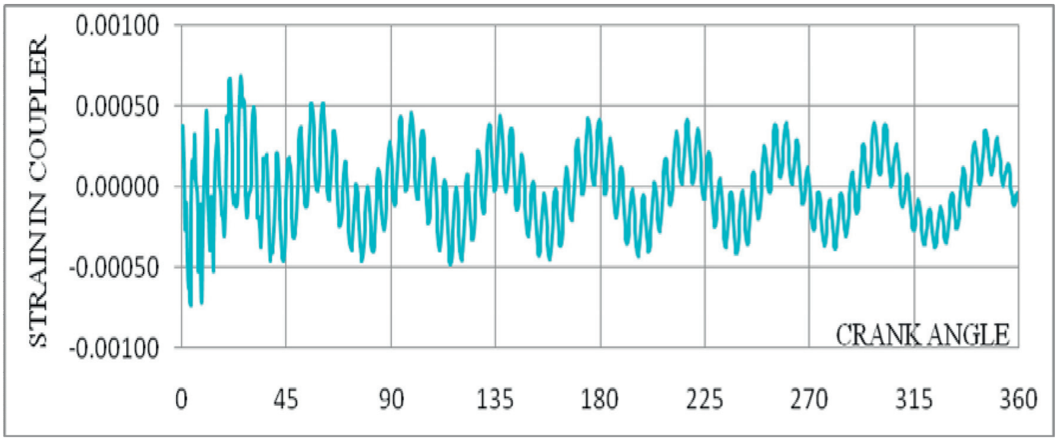
(a)


0.00020
0.00010
0.00000
-0.00010
-0.00020
-0.00030
STRAININ COUPLER
CRANK ANGLE
0
45
90
135
180
225
270
315
360

(b)


0.00060
0.00040
0.00020
0.00000
-0.00020
-0.00040
-0.00060
-0.00080
STRAININ COUPLER
CRANK ANGLE
0
45
90
135
180
225
270
315
360

(c)



(d)

Fig. 5. Effect of rocker length on coupler strain, rocker length (a) 240 mm (b) 250 mm (c) 270 mm (d) 280 mm

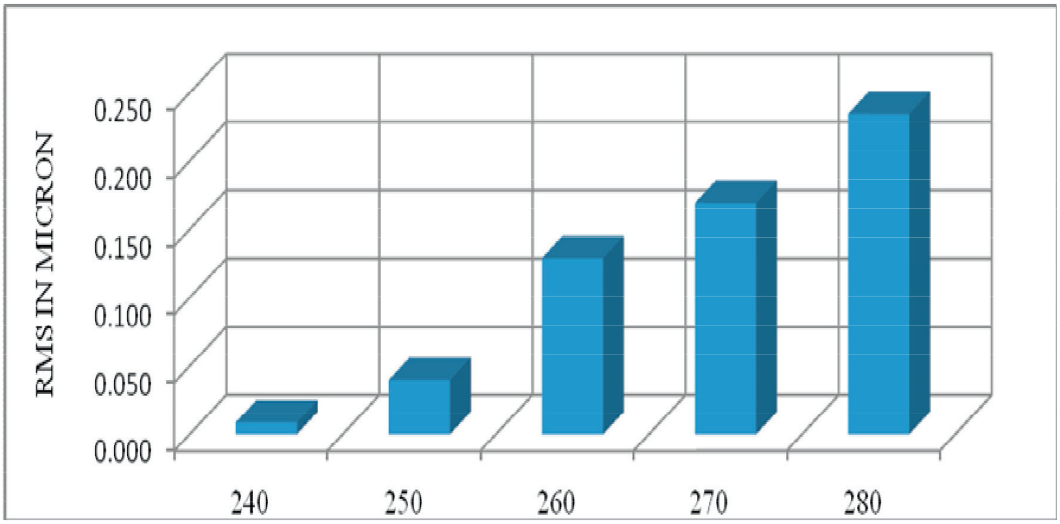


Fig. 6. Comparison of RMS of coupler strain for different rocker length (mm)

RMS value for the different length of rocker link is mention in Fig. 6 and it can be concludes that by increasing the length of rocker link strain produced in flexible coupler links is increased. It means that length of rocker link keep as minimum as possible.

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