Chapter – 6 Results and Discussions

The acceleration analysis at different crank position must be performed prior to the dynamic analysis of the any mechanisms. Therefore, in the present study position analysis, velocity analysis and acceleration analysis have been performed using the Eqs. (3.1) to (3.16).

The Table 5.2 provides the detailed specifications for four bar planar mechanism used for analysis purpose during this study. These specifications are selected as per the experimental data available in literature.

6.1 Rigid dynamics analysis of four bar planar mechanism

The Eqs. (3.7) and (3.8) were solved using MATLAB code to compute the angular velocity of coupler and rocker. The Fig. 6.1(a) shows the angular velocity of coupler and rocker at different crank position. It can be observed that the direction of angular velocity of coupler and rocker is opposite to crank during the crank rotation from 92° to 310° and 40° to 260° respectively. Rigid dynamic system tool was selected in ANSYS to perform the rigid analysis of mechanism. Geometry of four bar mechanism was prepared as per specification mentioned in Table 5.1 in Pro-E and imported in ANSYS and all connections were defined as revolute joint. Angular velocity of coupler and rocker were obtained through the probe located on both links are shown in the Fig. 6.1(b). The angular velocities of coupler and rocker obtained through MATLAB and ANSYS are seems to be identical.

The Fig. 6.2(a) shows the angular acceleration of coupler and rocker. These angular accelerations were obtained through MATLAB by solving Eqs. (3.12) and (3.13). The same results were obtained (refer Fig. 6.2(b)) by ANSYS analysis. Angular acceleration probe were selected for this purpose.
Fig. 6.1 Coupler and rocker angular velocity using (a) MATLAB  (b) ANSYS
Fig. 6.2 Coupler and rocker angular acceleration using (a) MATLAB (b) ANSYS
This angular acceleration of coupler and rocker arm ($\ddot{r}_{c}$) at different crank positions is used in equations of motions (Eq. (4.81)) for the dynamic analysis.

The torque required to operate the mechanism depends on transmission angle, angular acceleration and reaction forces at pin joints. As mentioned in section 3.1.1 the transmission angle is angular difference of link 3 and link 4. Transmission angle and driving torque at different crank position is presented in Fig. 6.3 and Fig. 6.4, respectively. From these figures, it is found that maximum driving torque is required at maximum transmission angle i.e. 180°.
The rigid dynamic analysis has been carried out to determine the forces acting at all pin joints. As mentioned in section 3.2, the D’Alembert’s principle and Newton’s second law is used for this rigid analysis in MATLAB (refer Eqs. (3.17) to (3.29)) and joints probe in ANSYS (rigid dynamics system). The pin forces developed at all joint for different crank angles are shown in the Fig. 6.5. The pin forces are found slightly more during ANSYS analysis (Fig. 6.5(b)) as compared to MATLAB analysis (Fig. 6.5(a)). This variation in pin forces occurred due to friction consideration at joints in ANSYS.

Fig. 6.5 Pin joints force using (a) MATLAB    (b) ANSYS
6.2 Flexible dynamics analysis of four bar planar mechanism

In the previous analysis the linkages are assumed to be rigid. But in reality linkages of mechanisms deform due to application of forces and they should be modeled as elastic bodies. It is worth to mention here, that the generalized equations of motion (Eq. (4.81)) for an elastic mechanism system have been derived using displacement finite element theory. The equations of motion were first developed for a single finite bar and beam element, then for a single link of the mechanism and finally for the entire mechanism system.

The strain resulted due to the elasticity or flexibility in the links has been computed and the results are presented in Fig. 6.6. During the computation of the coupler strain, each link has been considered as (i) single element in MATLAB analysis and (ii) 102 elements for coupler in ANSYS analysis. It is found that the Root Means Square (RMS) values of coupler strain are 0.000535 and 0.000138 in MATLAB (Fig. 6.6(a)) and ANSYS (Fig. 6.6(b)), respectively. The simulated results have been validated with the experimental results presented by Turcic et al. [13]. Their experimental results are re-plotted in Fig. 6.7. For more clarity and understating the RMS values of strain in coupler obtained from MATLAB, ANSYS and Experimentations are mentioned in Table 6.1. It is noticed that, the simulated results are in good agreement with experimental results in form of amplitude and phase.
Fig. 6.6 Strain in coupler using (a) MATLAB  (b) ANSYS
The Strain produce in the flexible coupler depends on (i) its cross section shape and orientation, (ii) flexibility of other links (crank, rocker) and (iii) the length of links. Therefore, in this study the above parameters have been varied during the analysis and results are discussed in the following paragraphs.

The cross section shapes under studies are rectangular, circular and elliptical cross section with cross section area of 40 mm². Moreover, to change the orientation of rectangular cross section, the width of coupler was kept parallel to axis of rotation and perpendicular to axis of rotation (refer Figs. 5.12 and 5.13). This change in orientation of coupler’s cross section has changed its moment of inertia.

The strain of rectangular cross section coupler has been compared with circular cross section and elliptical cross section couplers in Figs. 6.8 to 6.10. The
Fig. 6.8 shows the strain in coupler for different orientation of rectangular cross section. From Fig. 6.8 it is observed that the amplitude of the strain has decreased when the orientation 1 is changed to orientation 2. From Figs. 6.9 and 6.10 it is noticed that the amplitude of the strain has reduced for circular cross section while it has increased for elliptical cross section. The RMS values of coupler strain for the rectangular, circular and elliptical cross sections have been mentioned in Table 6.2 for ready reference.

Fig. 6.8 Comparison of rectangle cross section of coupler with different orientation
Fig. 6.9 Comparison of rectangular and circular cross section of coupler

Fig. 6.10 Comparison of rectangular and elliptical cross section of coupler
Table 6.2 Root Mean Square of strain for different shape of cross section

<table>
<thead>
<tr>
<th>Cross section</th>
<th>Rectangular (Orient. 1)</th>
<th>Rectangular (Orient. 2)</th>
<th>Circular</th>
<th>Elliptical</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMS Value</td>
<td>0.0001382</td>
<td>0.000021</td>
<td>0.000027</td>
<td>0.000272</td>
</tr>
</tbody>
</table>

To study the effect of the rocker length, the length of rocker was kept as 240 mm, 250 mm, 260 mm, 270 mm, 280 mm with rectangular cross section of coupler. The effect of rocker length on coupler strain has been shown in Fig. 6.11.
Fig. 6.11 Effect of rocker length on coupler strain, rocker length
(a) 240 mm (b) 250 mm (c) 270 mm (d) 280 mm
The RMS values of coupler strain for different rocker length have been presented in Fig. 6.12 in form of bar chart and also tabulated in Table 6.3. It has been observed that the amplitude of coupler strain has increased with length.

![Bar chart showing RMS values for different rocker lengths](image)

**Fig. 6.12 Comparison of RMS of coupler strain for different rocker length (mm)**

<table>
<thead>
<tr>
<th>Length of rocker (mm)</th>
<th>240</th>
<th>250</th>
<th>260</th>
<th>270</th>
<th>280</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMS Value</td>
<td>0.00864</td>
<td>0.03985</td>
<td>0.12905</td>
<td>0.16946</td>
<td>0.23461</td>
</tr>
</tbody>
</table>

**Table 6.3 Root Mean Square of strain for different length of rocker**

Fig. 6.13 shows the coupler strain for coupler length of 260 mm, 270 mm, 280 mm, 290 mm and 300 mm. The RMS values of coupler strain are tabulated in Table 6.4 and also represented in bar chart (Fig. 6.14) shows that the coupler strain reduces with the increase of coupler length.

<table>
<thead>
<tr>
<th>Length of coupler (mm)</th>
<th>260</th>
<th>270</th>
<th>280</th>
<th>290</th>
<th>300</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMS Value</td>
<td>0.19460</td>
<td>0.02020</td>
<td>0.12900</td>
<td>0.00950</td>
<td>0.00860</td>
</tr>
</tbody>
</table>

**Table 6.4 Root Mean Square of strain for different length of coupler**

6.13
Fig. 6.13 Effect of coupler length on coupler strain, coupler length
(a) 260 mm (b) 270 mm (c) 280 mm  (d) 290 mm (e) 300 mm

Fig. 6.14 Comparison of RMS of coupler strain for different coupler length (mm)
The results from the model were confirmed in that the stresses were reduced when the length of coupler was increased. Also it was confirmed that when the transmission angle was beyond a certain limit, the stresses becomes critical at an input angle of 180º, the angle at which the maximum transmission angle occurs and the effect of increasing coupler length is reversed.

To study the effect of links flexibility on coupler strain the following combination of links were kept flexible during analysis and results have been plotted in Figs 6.15 to 6.20.

Case I: coupler  
Case II: coupler and rocker  
Case III: coupler and fixed

Case IV: coupler and crank  
Case V: coupler, crank and rocker

Case VI: all links

The comparison of RMS value for coupler strain in Fig. 6.21 shows that coupler strain is least when coupler and rocker are flexible and also mention in Table 6.5.
Fig. 6.16 Strain in coupler for case II

Fig. 6.17 Strain in coupler for case III
Fig. 6.18 Strain in coupler for case IV

Fig. 6.19 Strain in coupler for case V
Fig. 6.20 Strain in coupler for case VI

Fig. 6.21 Comparison of RMS of coupler strain for different cases
Table 6.5 Root Mean Square of coupler strain for different cases

<table>
<thead>
<tr>
<th>Cases</th>
<th>Case I</th>
<th>Case II</th>
<th>Case III</th>
<th>Case IV</th>
<th>Case V</th>
<th>Case VI</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMS Value</td>
<td>0.127</td>
<td>0.006</td>
<td>0.021</td>
<td>0.01</td>
<td>0.01</td>
<td>0.012</td>
</tr>
</tbody>
</table>

6.3 Flexible dynamic analysis of Watt’s mechanism

The good agreement of ANSYS result with experimental results available in literature has proved the effectiveness of methodology used in dynamic analysis of four bar planar mechanism. Therefore, same methodology has been adopted for the dynamic analysis of Watt’s mechanism which is combination of two four bar mechanism connected in series.

During dynamic analysis of Watt’s mechanism link 3 and link 5 were kept flexible. The simulated results of strain generated in link-3 and link-5 are shown in Figs. 6.22 and 6.23.

![Strain in link-3 vs Crank angle](image-url)

Fig. 6.22 Crank angle versus strain in link-3
Fig. 6.23 Crank angle versus strain in link-5