A Design Method for Four-bar Mechanisms with Variable Speeds and Length-adjustable Driving Links*

Ren-Chung SOONG **
** Department of Mechanical and Automation Engineering, Kao Yuan University
No.1821, Jhongshan Rd., Lujhu Township, Kaohsiung County 821, Taiwan (R.O.C.)
E-mail: soongrc@cc.kyu.edu.tw

Abstract
This paper proposes a design method to improve or obtain the desired output motion characteristics of single degree-of-freedom (DOF) Four-bar mechanisms by varying the speed trajectory and the length of the input link. This approach adds a ternary link to the original mechanism. The ternary link is adjacent to the input link, roller link, and connecting link, and makes use of a sliding pair, revolute pair, and revolute pair, respectively. The roller link, which is adjacent to the ternary link and fixed link by a revolute pair and a rolling pair, is guided by a designed slot embedded in the fixed link while the input link is driven by a servomotor with a specific speed trajectory. A Bezier curve is applied to determine the input speed trajectory and contour of the guiding slot. Examples are given to verify the feasibility and effectiveness of this work.

Key words: Adjustable; Variable Input Speed; Four-bar Mechanism; Single DOF; Bezier Curve

1. Introduction

Four-bar mechanisms are commonly used in mechanical devices such as four-bar linkages and slider-crank mechanisms. Traditionally, the input speed trajectory and link lengths of a linkage mechanism are designed to be constant and fixed. When the kinematic requirements of an existing linkage mechanism need to be changed or improved for new applications, from a kinematic point of view, the output motion characteristics of the linkage mechanisms, such as displacement, velocity, and acceleration, are functions of the input motion characteristics and link lengths. Therefore, one can either vary just the input speed trajectories of the original mechanism, or vary both the input speed trajectories and the link lengths. The feasibility and effectiveness of the former has been verified by several researchers. In comparison, the main advantage of the latter is its increased flexibility in practical applications. But this strategy has higher costs because the degrees of freedom (DOF) of the original mechanism will increase to two or more, depending on how many links are adjustable.

Some researchers have improved the performance and raised the flexibility of the linkage mechanisms by varying the speed trajectory of the input link. Yan et al. (1)-(4) proposed eliminating the discontinuity in motion characteristics and lowering the peak values of the follower acceleration in cam-follower systems by implementing servo control. Yossifon and Shivpuri (5)-(6) discussed the design, analysis, and construction of a servomotor-controlled mechanical press for precision forming. Doege and Hindersmann (7) optimized the kinematics by designing non-circular gears to drive mechanical presses. Yao et al. (8)-(9) applied optimal control theory to control the cam speed and improve the motion characteristics of the follower. Yan and Chen (10) designed a general input speed trajectory...
for slider-crank mechanisms that led to an arbitrarily designated output motion. Yan and Chen\textsuperscript{(11)} further proposed a novel approach of varying the crank input speed such that the ram’s motion was suitable for both deep-drawing and precision-cutting processes.

Other researchers have focused on obtaining the desired kinematic requirements by adjusting the link length or the position of the moving or fixed pivot in the mechanisms. Naik and Amarnath\textsuperscript{(12)} presented a synthesis of adjustable four-bar function generators through five-bar loop closure equations. Wang and Sodhi\textsuperscript{(13)} proposed a kinematic synthesis method for four-bar mechanisms by adjusting the moving pivot for multi-phase motion generation. Zhou and Ting\textsuperscript{(14)} presented an optimal synthesis method that made simple adjustments to the slider guider position and used adjustable slider-crank linkages to generate multiple continuous paths. Cheung and Zhou\textsuperscript{(15)} developed an optimal synthesis method that utilized adjustable four-bar linkages for path generation by adjusting the position of a driven side-link fixed pivot. Zhou and Cheung\textsuperscript{(16)} proposed an optimal synthesis method that used adjustable four-bar linkages for multi-phase motion generation by adjusting the position of a driven side-link fixed pivot. Wu and Chen\textsuperscript{(17)} developed a mathematical model and ran a simulation for four-bar linkages. They exactly synthesized any satisfied input-output relationship by adjusting the link length of the coupler link. Shimojima et al.\textsuperscript{(18)} developed a synthesis method for straight line and L-shaped path generation by using fixed pivot positions as adjustable parameters. Chuenchom and Kota\textsuperscript{(19)} presented a synthesis method for adjustable mechanisms using adjustable dyads. Chang\textsuperscript{(20)} proposed a synthesis method for adjustable mechanisms to trace variable arcs with prescribed velocities. Russell and Sodhi\textsuperscript{(21)(22)} presented a method to design a slider path that could achieve multi-phase path, function, and motion generation applications for adjustable slider-crank mechanisms using seventh-order polynomials. Zhou\textsuperscript{(23)} proposed an optimal synthesis method for generating precise functions by adjusting the link length of the coupler link or driven link.

In this paper, we propose a new design method to improve or obtain the desired output motion characteristics. By simultaneously varying the speed trajectory of the driving link and adjusting its link length, the output motion characteristics of linkage mechanisms can be changed to satisfy new kinematic design requirements.

2. The four-bar mechanisms with variable speeds and length-adjustable driving links

The four-bar mechanisms with variable speeds and length-adjustable driving links are examined in this paper. Our approach adds a ternary link to the original mechanism. The ternary link is adjacent to the input link, roller link, and connecting link, and makes use of a sliding pair, revolute pair, and revolute pair, respectively, as shown in Figs. 1 and 2.

![Fig. 1 A slider-crank mechanism with length-adjustable driving link](image)
The roller link, which is adjacent to the ternary link and fixed link by a revolute pair and a rolling pair, respectively, is guided by a designed slot embedded in the fixed link, while the input link is driven by a servomotor with a specific speed trajectory. By properly designing the speed trajectory of the driving link and the contour of the roller guiding slot, improved or newly desired output motion characteristics can be generated.

3. Kinematic Analysis

A coordinate system of the slider-crank mechanism as shown in Fig. 1, the vector loop equation can be written as

\[ \vec{r}_2 + \vec{r}_3 - \vec{r}_1 - \vec{r}_4 = 0 \]  

(1)

Here, we represent each vector by a length \( r_i \) and an angle \( \phi_i \) as shown in Fig. 1. The length of driving link can be expressed as

\[ r_5 = r_{5_{\text{ol}}} + \Delta r_{5_{\text{int}}} + \Delta r_2 \]  

(2)

where \( r_{5_{\text{ol}}} \) is the original length, \( \Delta r_2(t) \) is the instantaneous length adjustment magnitude of the driving link, and \( \Delta r_{5_{\text{int}}} \) is the initial length adjustment magnitude of the driving link.

This vector equation can be separated into two scalar component equations in the \( x \)- and \( y \)-directions, respectively, as

\[ r_2 \cos \phi_2 + r_3 \cos \phi_3 - r_1 = 0 \]  

(3)

\[ r_2 \sin \phi_2 + r_3 \sin \phi_3 + r_4 = 0 \]  

(4)

Solving Eqs. (3) and (4) for \( \phi_3 \) and \( r_4 \) give

\[ \phi_3 = \cos^{-1} \left( \frac{r_1 - r_2 \cos \phi_2}{r_3} \right) \]  

(5)

\[ r_4 = -(r_2 \sin \phi_2 + r_3 \sin \phi_3) \]  

(6)

Differentiating Eqs. (3) and (4) with respect to time and rearranging results in \( \omega_3 \) and \( v_4 \).

\[ \omega_3 = \frac{d\phi_3}{dt} = \frac{\Delta r_2 \cos \phi_2 - r_2 \sin \phi_2 \omega_2}{r_3 \sin \phi_3} \]  

(7)

\[ v_4 = \frac{dr_4}{dt} = -(\Delta \dot{r}_2 \sin \phi_2 + r_2 \cos \phi_2 \omega_2 + r_3 \cos \phi_3 \omega_2) \]  

(8)

Where \( \omega_2 = \frac{d\phi_2}{dt} \) and \( \Delta r_2 = \frac{d\Delta r_2}{dt} \).
Differentiating Eqs. (7) and (8) with respect to time and rearranging results in $\alpha_3$ and $\alpha_4$:

$$\alpha_3 = \frac{d\alpha_3}{dt} = \frac{\Delta r_2 \cos \phi - \Delta r_2 \sin \phi \omega_3 - r_2 \cos \phi \omega_3^2 - r_2 \sin \phi \alpha_3 - r_2 \cos \phi \omega_3^2}{r_2 \sin \phi}$$ (9)

$$\alpha_4 = \frac{d\alpha_4}{dt} = -(\Delta r_2 \sin \phi + \Delta r_2 \cos \phi \omega_2 - r_2 \sin \phi \omega_2^2 + r_2 \cos \phi \alpha_2 - r_2 \sin \phi \omega_2^2 + r_3 \cos \phi \alpha_3)$$ (10)

Where $\alpha_2 = \frac{d\omega_2}{dt}$ and $\Delta r_2 = \frac{d^2 \alpha_2}{dt^2}$.

A coordinate system of the four-bar mechanism as shown in Fig. 2, the transmission angle be expressed as

$$\mu = \cos^{-1}\left(\frac{-r_1^2 - r_2^2 + r_3^2 + 2rr_2 \cos \phi}{2rr_2}\right).$$ (11)

If the four-bar linkage is a crank-rocker mechanism, its limit positions of the output link can be expressed as

$$\phi_{d1} = \cos^{-1}\left(\frac{(r_2 - r_2)^2 - r_3^2 - r_4^2}{2rr_4}\right), \quad \phi_{d2} = \cos^{-1}\left(\frac{(r_2 + r_2)^2 - r_3^2 - r_4^2}{2rr_4}\right).$$ (12)

4. Speed trajectory of the input crank

We assume that the driving link of the four-bar mechanisms is a crank. In this paper, the angular position trajectory of the crank is defined by an $n^{th}$ order Bezier curve $\phi_2(t)$ with parameter $t$ as follows:

$$\phi_2(t) = \sum_{i=0}^{n} \theta_i \cdot B_{i,n}(t)$$ (13)

where

$$B_{i,n}(t) = \frac{n!}{i!(n-i)!} \cdot t^i \cdot (1-t)^{n-i}, \quad t \in [0,1],$$ (14)

in which $\phi_2(t)$ is a Bezier curve that represents the angular position of the driving link defined by control points $\theta_i$. Parameter $t$ is regarded as the normalized time from 0 to 1. The $n$th-order differentiability of the Bezier curve guarantees the smoothness of the entire motion. Hence, the angular velocity $\omega_2(t)$ and acceleration $\alpha_2(t)$ of the driving link can be derived by continuously differentiating Eqs. (17) and (18) with respect to time as

$$\omega_2(t) = \frac{d\phi_2(t)}{dt} = \sum_{i=0}^{n} \theta_i \cdot \frac{dB_{i,n}(t)}{dt}$$ (15)

$$\alpha_2(t) = \frac{d^2\phi_2(t)}{dt^2} = \sum_{i=0}^{n} \theta_i \cdot \frac{d^2B_{i,n}(t)}{dt^2},$$ (16)

where

$$\frac{dB_{i,n}(t)}{dt} = \frac{n!}{i!(n-i)!} \cdot t^{i-1} \cdot (1-t)^{n-i} - \frac{n!}{i!(n-i-1)!} \cdot t^i \cdot (1-t)^{n-i-1}$$ (17)

$$\frac{d^2B_{i,n}(t)}{dt^2} = \frac{n!}{(i-2)!(n-i)!} \cdot t^{i-2} \cdot (1-t)^{n-i} - \frac{n!}{(i-1)!(n-i-1)!} \cdot t^{i-1} \cdot (1-t)^{n-i-1}$$

$$+ \frac{n!}{(i-1)!(n-i-2)!} \cdot t^{i-3} \cdot (1-t)^{n-i-2} + \frac{n!}{i!(n-i-2)!} \cdot t^{i-4} \cdot (1-t)^{n-i-3}$$ (18)
5. Contour design of the guiding slot

To correspond with the angular position of the driving link, the simultaneous length-adjustable magnitude of the driving link is also created by an \( n \)th order Bezier curve \( \Delta r_2(t) \) with parameter \( t \), similar to Eq. (13), as

\[
\Delta r_2(t) = \sum_{i=0}^{n} \lambda_i \cdot B_{i,n}(t)
\]  
(19)

\[
\frac{d\Delta r_2(t)}{dt} = \sum_{i=0}^{n} \lambda_i \cdot \frac{dB_{i,n}(t)}{dt}
\]  
(20)

\[
\frac{d^2\Delta r_2(t)}{dt^2} = \sum_{i=0}^{n} \lambda_i \cdot \frac{d^2B_{i,n}(t)}{dt^2}
\]  
(21)

where \( \Delta r_2(t) \) is a Bezier curve that represents the simultaneous length-adjustable magnitude of the driving link defined by control points \( \lambda_i \). The coordinates of the \( i \)th point on the contour of the guiding slot, \( x_{ip} \) and \( y_{ip} \), can be written as follows:

\[
x_{ip} = r_{i0\Delta} + \Delta r_{2i} + \Delta r_{m0}\cos\phi_{ipl},
\]
\[
y_{ip} = r_{i0\Delta} + \Delta r_{2i} + \Delta r_{m0}\sin\phi_{ipl}
\]  
(22)

where \( \phi_{ipl} \) and \( \Delta r_{2i} \) represent the angular position and the simultaneous length-adjustable magnitude of the driving link, respectively, corresponding to the \( i \)th point on the contour of the guiding slot.

6. Optimum design

Based on the kinematic analysis, we know that the speed trajectory is determined by variables \( \theta_0, ..., \theta_n \), and the total length of the adjustable driving link is determined by variables \( \lambda_0, ..., \lambda_n \) and \( r_{m0} \). An optimization procedure was used in this approach to determine all the design variables, the general optimization equations for which can be defined as

\[
\text{minimize } f(\theta_0, \theta_1, ..., \theta_{n-1}, \lambda_0, \lambda_1, ..., \lambda_{n-1}, \lambda_n + \Delta r_{m0}) = \sum_{j=1}^{n} obj_j
\]  
(23)

subject to the constraints of equality

\[
c_j(\theta_0, \theta_1, ..., \theta_{n-1}, \lambda_0, \lambda_1, ..., \lambda_{n-1}, \lambda_n + \Delta r_{m0}) = 0 \quad j = 1, ..., n_e
\]  
(24)

and the constraints of inequality

\[
g_j(\theta_0, \theta_1, ..., \theta_{n-1}, \lambda_0, \lambda_1, ..., \lambda_{n-1}, \lambda_n + \Delta r_{m0}) < 0 \quad j = 1, ..., n_i
\]  
(25)

where \( obj_j \) is the objective function, \( n_e \) denotes the number of objective functions, and \( n_e \) and \( n_i \) denote the number of equality and inequality constraint equations, respectively.

Eqs. (23) to (25) are the general expressions of the objective function, constrains of equality and inequality equations, respectively, their exact expressions depend on design requirements and constraints. In this paper, Eq. (23) refers to Eqs. (2), (10), (13), (15)-(16) and (19)-(21), Eq. (24) refers to Eqs. (2), (6), (13), (15)-(16) and (19), Eq. (25) refers to Eqs. (2), (8), (13), (15) and (19)-(20) for Example 1 to Example 3, Eq. (23) refers to Eqs. (2), (11), (13) and (19), Eq. (24) refers to Eqs. (2), (12), (15)-(16) and (19), Eq. (25) refers to Eqs. (2) and (19) for Example 4.

Any optimization method can be used to determine the design variables. In this study, all of the information required for the optimization was derived, and a sequential quadratic programming subroutine was applied to solve the design variables.
7. Examples and discussion

In this section, examples are provided to demonstrate the feasibility of our proposed approach. A tenth-order Bezier curve (with 11 control points) was used to represent the speed trajectory and the length of the adjustable driving link. It is clear that $\theta_0$, $\lambda_0$, $\theta_n$, and $\lambda_n$ are the boundary conditions in consecutive cycles. Therefore, $\theta_0 = 0$ and $\theta_n = 360^\circ$, and $\lambda_0 = 0$ and $\lambda_n = 0$ in order to obtain a closed contour for the guiding slot. The average speed of the driving link was set to 60 rpm for all examples.

Example 1

For the punch press with dimensions $r_2 = 50, r_3 = 160$ in Ref. (10), we assumed that constant velocity drawing processes were desired during the forward stroke between $t = 0.2$ and 0.3. We added a ternary link and its guiding slot, as shown in Fig. 1, to the original mechanism. The task was to design the speed trajectory and length of the driving link to satisfy the desired kinematic design requirements and minimize the peak acceleration of the ram. The optimization problem was defined as minimizing

$$f(\theta_1, \ldots, \theta_9, \lambda_1, \ldots, \lambda_9 + \Delta r_{ini}) = \text{peak of } a_{ram}$$

subject to

$$c_1(\theta_1, \ldots, \theta_9) = \omega_2(0) - \omega_2(1) = 0$$
$$c_2(\theta_1, \ldots, \theta_9) = \alpha_2(0) - \alpha_2(1) = 0$$
$$c_3(\theta_1, \ldots, \theta_9, \lambda_1, \ldots, \lambda_9 + \Delta r_{ini}) = s(t_a) = s_{a_{max}} = s_{e_{max}}$$
$$g_1(\theta_1, \ldots, \theta_9, \lambda_1, \ldots, \lambda_9, \Delta r_{ini}) = \int_{t=tdf+0.1}^{t=tdf+0.2} |v(t) - v(t_{ds} = 0.2)| \, dt - \varepsilon_r < 0$$

where $a_{ram}$ is the linear acceleration of the ram; $s$ denotes the linear position of the ram; $s_{a_{max}}$ and $s_{e_{max}}$ are the maximum linear displacement of the ram for the new press and the original press, respectively; $t_a$ represents the normalized time corresponding to $s_{a_{max}}$ and $s_{e_{max}}$; $v$ denotes the linear velocity of the ram; $t_{ds}$ and $t_{de}$ represent the time of the beginning and end of a specific period; and $\varepsilon_r$ is a small number.

In this example the optimal control points of the speed trajectory and the simultaneous length-adjustable magnitude of the driving link are shown in Tables 1 and 2, respectively. The input and output motion characteristics are shown in Fig. 3, and the length-adjustable magnitude of the driving link and the contour of the guiding slot are shown in Fig. 4.

Table 1 Control points of the speed trajectory of the driving link in all examples (°)

<table>
<thead>
<tr>
<th>Example</th>
<th>$\theta_1$</th>
<th>$\theta_2$</th>
<th>$\theta_3$</th>
<th>$\theta_4$</th>
<th>$\theta_5$</th>
<th>$\theta_6$</th>
<th>$\theta_7$</th>
<th>$\theta_8$</th>
<th>$\theta_9$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>30.7</td>
<td>59.5</td>
<td>128.8</td>
<td>61.1</td>
<td>192.6</td>
<td>211.6</td>
<td>246.6</td>
<td>298.8</td>
<td>329.9</td>
</tr>
<tr>
<td>2</td>
<td>27</td>
<td>52.9</td>
<td>101.1</td>
<td>168.7</td>
<td>179.5</td>
<td>189.9</td>
<td>262.5</td>
<td>305.5</td>
<td>333</td>
</tr>
<tr>
<td>3</td>
<td>40.8</td>
<td>79.7</td>
<td>125.3</td>
<td>329.1</td>
<td>186.5</td>
<td>15.2</td>
<td>244</td>
<td>276.5</td>
<td>319.2</td>
</tr>
<tr>
<td>4</td>
<td>92</td>
<td>82</td>
<td>237.2</td>
<td>0</td>
<td>16.8</td>
<td>310.7</td>
<td>360</td>
<td>74</td>
<td>268</td>
</tr>
</tbody>
</table>

Table 2 Control points of the simultaneous length-adjustable magnitude of the driving link in all examples (mm)

<table>
<thead>
<tr>
<th>Example</th>
<th>$\lambda_1$</th>
<th>$\lambda_2$</th>
<th>$\lambda_3$</th>
<th>$\lambda_4$</th>
<th>$\lambda_5$</th>
<th>$\lambda_6$</th>
<th>$\lambda_7$</th>
<th>$\lambda_8$</th>
<th>$\lambda_9$</th>
<th>$\Delta r_{ini}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>-1.29</td>
<td>-15</td>
<td>-15</td>
<td>-15</td>
<td>31.77</td>
<td>-0.02</td>
<td>-14.9</td>
<td>-15</td>
<td>-3</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>-5</td>
<td>-50</td>
<td>-39.09</td>
<td>-10.51</td>
<td>69.86</td>
<td>0.28</td>
<td>-49.68</td>
<td>-50</td>
<td>-5</td>
<td>0</td>
</tr>
<tr>
<td>3</td>
<td>2</td>
<td>-4.96</td>
<td>-4.92</td>
<td>0.268</td>
<td>3.8</td>
<td>-2</td>
<td>-4.99</td>
<td>-5</td>
<td>-3</td>
<td>0</td>
</tr>
<tr>
<td>4</td>
<td>0</td>
<td>15</td>
<td>15</td>
<td>0</td>
<td>-15</td>
<td>-15</td>
<td>-13.28</td>
<td>15</td>
<td>0.45</td>
<td>-2.62</td>
</tr>
</tbody>
</table>
Example 2

In this example, the objective was to minimize the peak acceleration of the ram. Hence, this optimization problem is similar to that in Example 1, except that we delete Eq. (29). The optimal control points of the speed trajectory and the simultaneous length-adjustable magnitude of the driving link as those given in Tables 1 and Table 2, respectively. The input and output motion characteristics are shown in Fig. 5. The length-adjustable magnitude of the driving link and the contour of the guiding slot are shown in Fig. 6.
Example 3

In this example, the task was to force the ram to undergo a specific dwelling period from before to after bottom dead center (BDC), and to minimize the ram peak acceleration for forging processes. Hence, this optimization problem is similar to that in Example 1 except for the constant-speed time period from \( t_{sl} = 0.45 \) to \( t_{sl} = 0.55 \) in Eq. (29).

The optimal control points of the speed trajectory and the simultaneous length-adjustable...
magnitude of the driving link as those shown in Tables 1 and Table 2, respectively. The input and output motion characteristics are shown in Fig. 7. The length-adjustable magnitude of the driving link and the contour of the guiding slot are shown in Fig. 8.

Example 4
In this example, we assumed that the limit positions of the crank-rocker mechanism with...
dimensions \( r_1 = 60, r_2 = 20, r_3 = 50 \) and \( r_4 = 40 \) had \( \phi_{11} = 150^\circ, \phi_{12} = 101.5^\circ \), and better transmission angles were expected in a cycle. We added a ternary link and its guiding slot, as shown in Fig. 2, to the original mechanism. The task was to define a speed trajectory and length-adjustable magnitude for the driving link that satisfied the new requirements. The optimization problem was defined as minimizing

\[
f(\theta_1, \ldots, \theta_9, \theta_0 + \Delta r_{\text{ini}}) = |90^\circ - \mu|
\]

subject to

\[
c_1(\theta_1, \ldots, \theta_9) = \omega_2(0) - \omega_2(1) = 0, \quad c_2(\theta_1, \ldots, \theta_9) = \alpha_2(0) - \alpha_2(1) = 0
\]

\[
c_3(\lambda_1, \ldots, \lambda_9, \Delta r_{\text{ini}}) = \phi_{11} = 155, \quad c_4(\lambda_1, \ldots, \lambda_9, \Delta r_{\text{ini}}) = \phi_{12} = 101.5
\]

\[
g_1(\lambda_1, \ldots, \lambda_9, \Delta r_{\text{ini}}) = (r_{2\text{ot}} + \Delta r_{\text{ini}} + \Delta r_2) < 30
\]

Where \( \mu \) is the transmission angle of the four-bar linkage; Eq. (32) guarantees the mechanism satisfying Grashof’s law such that the driving link serves as a crank.

The optimal control points of the speed trajectory and the simultaneous length-adjustable magnitude of the driving link remained as those shown in Tables 1 and 2, respectively. The input and output motion characteristics are shown in Fig. 9. The length-adjustable magnitude of the driving link and the contour of the guiding slot are shown in Fig. 10.
The length-adjustable magnitude of the driving link

The contour of the guiding slot

Fig. 10 The length-adjustable magnitude of the driving link and the contour of the guiding slot - Example 4

The desired output motion characteristics for Examples 1 and 2 were obtained, as shown in Figs. 3 and 5, respectively, and the ram peak acceleration was minimized substantially. These results are better than those reported in Ref. 10. The results of Example 3 (Fig. 7) indicate that during the forging process, the ram had a specific dwelling period from before to after BDC. Ref (10) reports that this dwelling period was difficult to attain. The limit positions of the output link in Example 4, shown in Fig. 9, were satisfied while maintaining a transmission angle near 90° throughout most of the cycle.

The results of these four examples suggest that it is not necessary to redesign the link dimensions when the design requirements are changed. We only have to determine a new set of control points for the speed trajectory and adjustable length of the driving link by using the optimization procedure for an existing mechanism. This design method makes the four-bar mechanisms programmable and flexible in industrial applications.

8. Conclusion

A new design method to improve or obtain desired output motion characteristics by varying the speed trajectory and the adjustable length of the driving link for single DOF four-bar mechanisms was proposed in this paper. Single DOF four-bar mechanisms with length-adjustable driving links were presented. This approach makes single DOF four-bar mechanisms programmable and adjustable, and increases their flexibility in practical applications. A Bezier curve was applied to design the input speed trajectory and the slot contour. Examples were provided to verify the feasibility and effectiveness of this work.

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