Program FOURBAR calculates the equivalent geared fivebar configuration for any fourbar linkage and will export its data to a disk file that can be opened in program FIVEBAR for analysis. The file F03-28aAbr can be opened in FOURBAR to animate the linkage shown in Figure 3-28a. Then also open the file F03-28b.5br in program FIVEBAR to see the motion of the equivalent geared fivebar linkage. Note that the original fourbar linkage is a triple-rocker, so cannot reach all portions of the coupler curve when driven from one rocker. But, its geared fivebar equivalent linkage can make a full revolution and traverses the entire coupler path. To export a FIVEBAR disk file for the equivalent GFBM of any fourbar linkage from program FOURBAR, use the *Export* selection under the *File* pull-down menu.

3.8 STRAIGHT-LINE MECHANISMS

A very common application of coupler curves is the generation of approximate straight lines. Straight-line linkages have been known and used since the time of James Watt in the 18th century. Many kinematicians such as Watt, Chebyschev, Peaucellier, Kempe, Evans, and Hoeken (as well as others) over a century ago, developed or discovered either approximate or exact straight-line linkages, and their names are associated with those devices to this day.

The first recorded application of a coupler curve to a motion problem is that of **Watt's straight-line linkage,** patented in 1784, and shown in Figure 3-29a. Watt devised his straight-line linkage to guide the long-stroke piston of his steam engine at a time when metal-cutting machinery that could create a long, straight guideway did not yet exist.* This triple-rocker linkage is still used in automobile suspension systems to guide the rear axle up and down in a straight line as well as in many other applications.

Richard Roberts (1789-1864) (not to be confused with Samuel Roberts of the cognates) discovered the **Roberts' straight-line linkage** shown in Figure 3-29b. This is a triple-rocker. **Chebyschev** (1821-1894) also devised a **straight-line linkage**-a Grashof double-rocker-shown in Figure 3-29c.

The **Hoeken linkage** [16] in Figure 3-29d is a Grashof crank-rocker, which is a significant practical advantage. **In** addition, the Hoeken linkage has the feature of very *nearly constant velocity along the center portion of its straight-line motion.* It is interesting to note that the **Hoecken** and **Chebyschev** linkages are cognates of one another.† The cognates shown in Figure 3-26 (p. 116) are the Chebyschev and Hoeken linkages.

These straight-line linkages are provided as built-in examples in program FOURBAR. A quick look in the Hrones and Nelson atlas of coupler curves will reveal a large number of coupler curves with **approximate straight-line** segments. They are quite common.

To generate an **exact straight line** with only pin joints requires more than four links. At least six links and seven pin joints are needed to generate an exact straight line with a pure revolute-jointed linkage, i.e., a Watt's or Stephenson's sixbar. A geared fivebar mechanism, with a gear ratio of -1 and a phase angle of π radians, will generate an exact straight line at the joint between links 3 and 4. But this linkage is merely a transformed Watt's sixbar obtained by replacing one binary link with a higher joint in the form of a gear pair. This geared fivebar's straight-line motion can be seen by reading the file STRAIGHT.5BR into program FIVEBAR, calculating and animating the linkage.

* In Watt's time, straight-line motion was dubbed "parallel motion" though we use that term somewhat differently now. James Watt is reported to have told his son *"Though I am not over anxious after fame, yet I am more proud of the parallel motion than of any other mechanical invention I have made."* Quoted in Muirhead, J. P. (1854). *The Origin and Progress of the Mechanical Inventions of James Watt*, Vol. 3, London, p. 89.

† Hain [17] (1967) cites the Hoeken reference [16] (1926) for this linkage. Nolle [18] (1974) shows the Hoeken mechanism but refers to it as a Chebyschev crank-rocker without noting its cognate relationship to the Chebyschev double-rocker, which he also shows. It is certainly conceivable that Chebyschev, as one of the creators of the theorem of cognate linkages, would have discovered the "Hoeken" cognate of his own double-rocker. However, this author has been unable to find any mention of its genesis in the English literature other than the ones cited here.

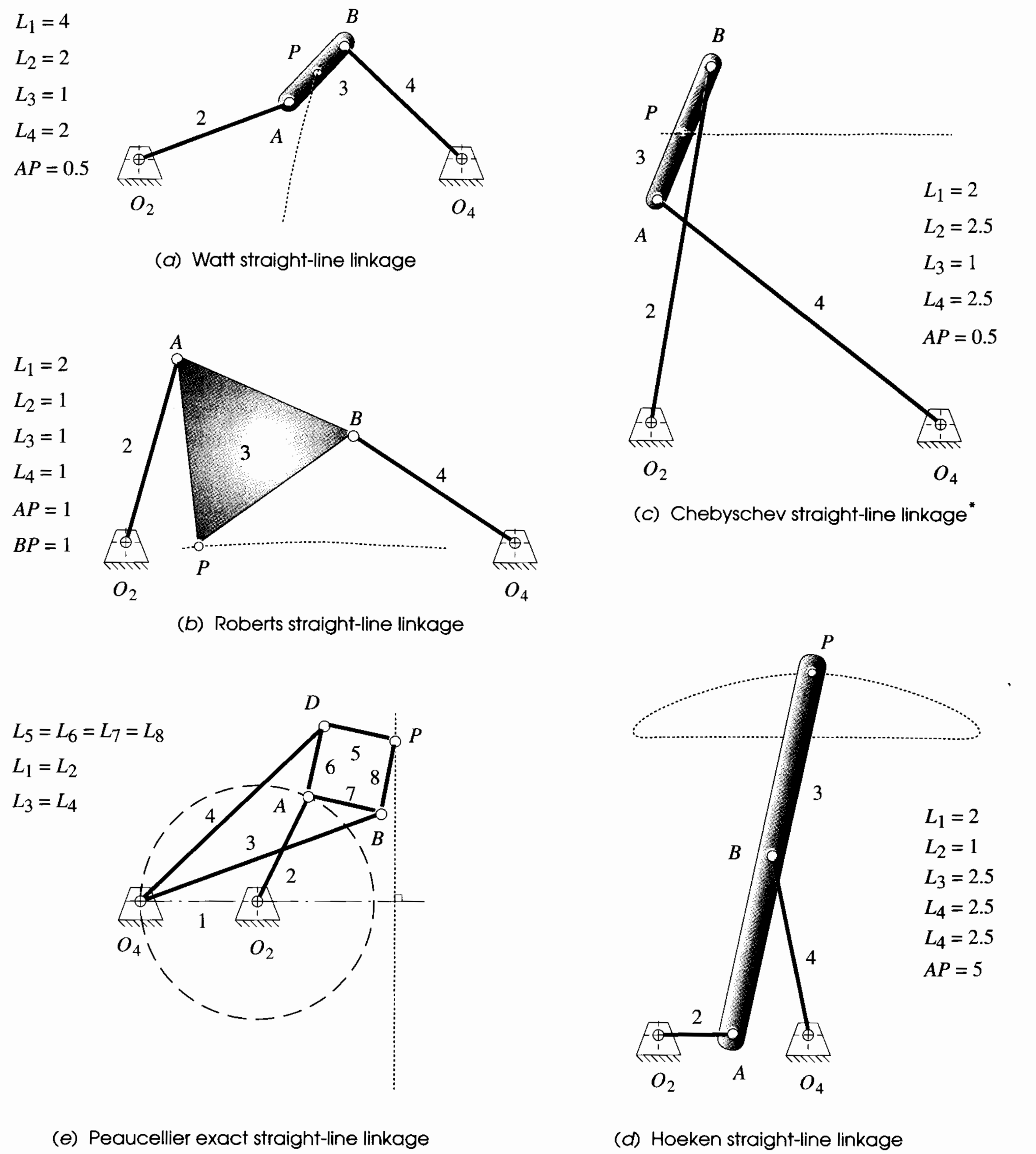


FIGURE 3-29

Some common and classic approximate, and one exact, straight-line linkages

* The link ratios of the Chebyschev straight-line linkage have been reported differently by various authors. The ratios used here are those first reported (in English) by Kempe (1877). But Kennedy (1893) describes the same linkage, reportedly "as Chebyschev demonstrated it at the Vienna Exhibition of 1893" as having the link ratios 1, 3.25, 2.5, 3.25. We will assume the earliest reference by Kempe to be correct as listed in the figure.

Peaucellier * (1864) discovered an exact straight-line mechanism of eight bars and six pins, shown in Figure 3-2ge. Links 5, 6, 7, 8 form a rhombus of convenient size. Links 3 and 4 can be any convenient but equal lengths. When OZ04 exactly equals OzA, point C generates an *arc of infinite radius,* i.e., an exact straight line. By moving the pivot Oz left or right from the position shown, changing only the length of link 1, this mechanism *will generate true circle arcs with radii much larger than the link lengths.*

Designing Optimum Straight-Line Fourbar Linkages

Given the fact that an exact straight line can be generated with six or more links using only revolute joints, why use a fourbar approximate straight-line linkage at all? One reason is the desire for simplicity in machine design. The pin-jointed fourbar is the simplest possible one-DOF mechanism. Another reason is that a very good approximation to a true straight line can be obtained with just four links, and this is often "good enough" for the needs of the machine being designed. Manufacturing tolerances will, after all, cause any mechanism's performance to be less than ideal. As the number of links and joints increases, the probability that an exact-straight-line mechanism will deliver its theoretical performance in practice is obviously reduced.

There is a real need for straight-line motions in machinery of all kinds, especially in automated production machinery. Many consumer products such as cameras, film, toiletries, razors, and bottles are manufactured, decorated, or assembled on sophisticated and complicated machines that contain a myriad of linkages and cam-follower systems. Traditionally, most of this kind of production equipment has been of the intermittent-motion variety. This means that the product is carried through the machine on a linear or rotary conveyor that stops for any operation to be done on the product, and then indexes the product to the next work station where it again stops for another operation to be performed. The forces and power required to accelerate and decelerate the large mass of the conveyor (which is independent of, and typically larger than, the mass of the product) severely limit the speeds at which these machines can be run.

Economic considerations continually demand higher production rates, requiring higher speeds or additional, expensive machines. This economic pressure has caused many manufacturers to redesign their assembly equipment for continuous conveyor motion. When the product is in continuous motion in a straight line and at constant velocity, every workhead that operates on the product must be articulated to chase the product and match both its straight-line path and its constant velocity while performing the task. These factors have increased the need for straight-line mechanisms, including ones capable of near-constant velocity over the straight-line path.

A (near) perfect straight-line motion is easily obtained with a fourbar slider-crank mechanism. Ball-bushings (Figure 2-26, p. 57) and hardened ways are available commercially at moderate cost and make this a reasonable, low-friction solution to the straight-line path guidance problem. But, the cost and lubrication problems of a properly guided slider-crank mechanism are still greater than those of a pin-jointed fourbar linkage. Moreover, a crank-slider-block has a velocity profile that is nearly sinusoidal (with some harmonic content) and is far from having constant velocity over any of its motion.

The Hoeken-type linkage offers an 0l?timum combination of straightness and near constant velocity and is a crank-rocker, so it can be motor driven. Its geometry, dimen-

* Peaucellier was a French army captain and military engineer who first proposed his "compas compose" or *compound compass* in 1864 but received no immediate recognition therefor. The British-American mathematician, James Sylvester, reported on it to the *Atheneum Club* in London in 1874. He observed that *"The perfect parallel motion of Peaucellier looks so simple and moves so easily that people who see it at work almost universally express astonishment that it waited so long to be discovered."* A model of the Peaucellier linkage was passed around the table. The famous physicist, Sir William Thomson (later Lord Kelvin), refused to relinquish it, declaring *"No. I have not had nearly enough of it-it is the most beautiful thing I have ever seen in my life."* Source: Strandh, S. (1979). *A History of the Machine.* A&W Publishers: New York, p. 67.

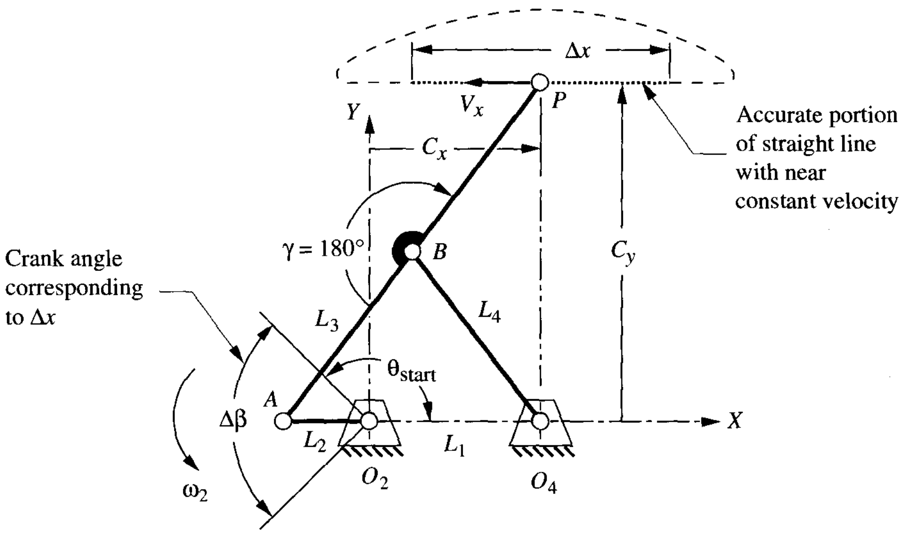


FIGURE 3-30

Hoekens linkage geometry. Linkage shown with *P* at center of straight-line portion of path

sions, and coupler path are shown in Figure 3-30. This is a symmetrical fourbar linkage. Since the angle γ of line BP is specified and $L_3 = L_4 = BP$, only two link ratios are needed to define its geometry, say L_1 / L_2 and L_3 / L_2. If the crank L_2 is driven at constant angular velocity ω_2, the linear velocity V_x along the straight line portion Δx of the coupler path will be very close to constant over a significant portion of crank rotation $\Delta\beta$.

A study was done to determine the errors in straightness and constant velocity of the Hoeken-type linkage over various fractions $\Delta\beta$ of the crank cycle as a function of the link ratios. [19] The structural error in position (i.e., straightness) ε_S and the structural error in velocity ε_V are defined using notation from Figure 3-29 (p. 121) as:

$$\varepsilon_S = \frac{MAX_{i=1}^{n}\left(C_{y_i}\right) - MIN_{i=1}^{n}\left(C_{y_i}\right)}{\Delta x}$$

$$\varepsilon_V = \frac{MAX_{i=1}^{n}\left(V_{x_i}\right) - MIN_{i=1}^{n}\left(V_{x_i}\right)}{\overline{V}_x} \qquad (3.5)^*$$

The structural errors were computed separately for each of nine crank angle ranges $\Delta\beta$ from 20° to 180°. Table 3-1 shows the link ratios that give the smallest possible structural error in either position or velocity over values of $\Delta\beta$ from 20° to 180°. Note that one cannot attain optimum straightness and minimum velocity error in the same linkage. However, reasonable compromises between the two criteria can be achieved, especially for smaller ranges of crank angle. The errors in both straightness and velocity increase as longer portions of the curve are used (larger $\Delta\beta$). The use of Table 3-1 to design a straight-line linkage will be shown with an example.

* See reference [19] for the derivation of equations 3.5.

TABLE 3-1 Link Ratios for Smallest Attainable Errors in Straightness and Velocity for Various Crank-Angle Ranges of a Hoeken-Type Fourbar Approximate Straight-Line Linkage (19)

Range of Motion			Optimized for Straightness						Optimized for Constant Velocity					
$\Delta\beta$ (deg)	θ_{start} (deg)	% of cycle	Minimum ΔC_y %	ΔV %	$\frac{V_x}{(L_2\omega_2)}$	Link Ratios L_1/L_2	L_3/L_2	$\Delta x/L_2$	Minimum ΔV_x %	ΔC_y %	$\frac{V_x}{(L_2\omega_2)}$	Link Ratios L_1/L_2	L_3/L_2	$\Delta x/L_2$
20	170	5.6%	0.00001%	0.38%	1.436	2.975	3.963	0.601	0.006%	0.137%	1.045	2.075	2.613	0.480
40	160	11.1%	0.00004%	1.53%	1.504	2.950	3.925	1.193	0.038%	0.274%	1.124	2.050	2.575	0.950
60	150	16.7%	0.00027%	3.48%	1.565	2.900	3.850	1.763	0.106%	0.387%	1.178	2.025	2.538	1.411
80	140	22.2%	0.001%	6.27%	1.611	2.825	3.738	2.299	0.340%	0.503%	1.229	1.975	2.463	1.845
100	130	27.8%	0.004%	9.90%	1.646	2.725	3.588	2.790	0.910%	0.640%	1.275	1.900	2.350	2.237
120	120	33.3%	0.010%	14.68%	1.679	2.625	3.438	3.238	1.885%	0.752%	1.319	1.825	2.238	2.600
140	110	38.9%	0.023%	20.48%	1.702	2.500	3.250	3.623	3.327%	0.888%	1.347	1.750	2.125	2.932
160	100	44.4%	0.047%	27.15%	1.717	2.350	3.025	3.933	5.878%	1.067%	1.361	1.675	2.013	3.232
180	90	50.0%	0.096%	35.31%	1.725	2.200	2.800	4.181	9.299%	1.446%	1.374	1.575	1.863	3.456

EXAMPLE 3-12

Designing a Hoeken-Type Straight-Line Linkage.

Problem: A 100-mm-long straight line motion is needed over 1/3 of the total cycle (120° of crank rotation). Determine the dimensions of a Hoeken-type linkage that will

(a) Provide minimum deviation from a straight line. Determine its maximum deviation from constant velocity.
(b) Provide minimum deviation from constant velocity. Determine its maximum deviation from a straight line.

Solution: (see Figure 3-30, p. 123, and Table 3-1)

1 Part (a) requires the most accurate straight line. Enter Table 3-1 at the 6th row which is for a crank angle duration $\Delta\beta$ of the required 120°. The 4th column shows the minimum possible deviation from straight to be 0.01% of the length of the straight line portion used. For a 100-mm length the absolute deviation will then be 0.01 mm (0.0004 in). The 5th column shows that its velocity error will be 14.68% of the average velocity over the 100-mm length. The absolute value of this velocity error of course depends on the speed of the crank.

2 The linkage dimensions for part (a) are found from the ratios in columns 7, 8, and 9. The crank length required to obtain the 100-mm length of straight line Δx is:

$$\text{from Table 3-1:} \qquad \frac{\Delta x}{L_2} = 3.238$$

$$L_2 = \frac{\Delta x}{3.238} = \frac{100 \text{ mm}}{3.23} = 30.88 \text{ mm} \tag{a}$$

The other link lengths are then:

from Table 3-1:
$$\frac{L_1}{L_2} = 2.625$$
$$L_1 = 2.625L_2 = 2.625(30.88 \text{ mm}) = 81.07 \text{ mm} \tag{b}$$

from Table 3-1:
$$\frac{L_3}{L_2} = 3.438$$
$$L_3 = 3.438L_2 = 3.438(30.88 \text{ mm}) = 106.18 \text{ mm} \tag{c}$$

The complete linkage is then: $L_1 = 81.07$, $L_2 = 30.88$, $L_3 = L_4 = BP = 106.18$ mm. The nominal velocity V_x of the coupler point at the center of the straight line ($\theta_2 = 180°$) can be found from the factor in the 6th column which must be multiplied by the crank length L_2 and the crank angular velocity ω_2 in radians per second (rad/sec).

3 Part (b) requires the most accurate velocity. Again enter Table 3-1 at the 6th row which is for a crank angle duration $\Delta\beta$ of the required 120°. The 10th column shows the minimum possible deviation from constant velocity to be 1.885% of the average velocity V_x over the length of the straight line portion used. The 11th column shows the deviation from straight to be 0.752% of the length of the straight line portion used. For a 100-mm length the absolute deviation in straightness for this optimum constant velocity linkage will then be 0.75 mm (0.030 in).

The link lengths for this mechanism are found in the same way as was done in step 2 above except that the link ratios 1.825, 2.238, and 2.600 from columns 13, 14, and 15 are used. The result is: $L_1 = 70.19$, $L_2 = 38.46$, $L_3 = L_4 = BP = 86.08$ mm. The nominal velocity V_x of the coupler point at the center of the straight line ($\theta_2 = 180°$) can be found from the factor in the 12th column which must be multiplied by the crank length L_2 and the crank angular velocity ω_2 in rad/sec.

4 The first solution (step 2) gives an extremely accurate straight line over a significant part of the cycle but its 15% deviation in velocity would probably be unacceptable if that factor was considered important. The second solution (step 3) gives less than 2% deviation from constant velocity, which may be viable for a design application. Its 3/4% deviation from straightness, while much greater than the first design, may be acceptable in some situations.

3.9 DWELL MECHANISMS

A common requirement in machine design problems is the need for a dwell in the output motion. A **dwell** is defined as *zero output motionfor some nonzero input motion.* In other words, the motor keeps going, but the output link stops moving. Many production machines perform a series of operations which involve feeding a part or tool into a workspace, and then holding it there (in a dwell) while some task is performed. Then the part must be removed from the workspace, and perhaps held in a second dwell while the rest of the machine "catches up" by indexing or performing some other tasks. Cams and followers (Chapter 8) are often used for these tasks because it is trivially easy to create a dwell with a earn. But, there is always a trade-off in engineering design, and cams have their problems of high cost and wear as described in Section 2.15 (p. 55).

It is also possible to obtain dwells with "pure" linkages of only links and pin joints, which have the advantage over cams of low cost and high reliability. Dwell linkages are more difficult to design than are cams with dwells. Linkages will usually yield only an approximate dwell but will be much cheaper to make and maintain than cams. Thus they may be well worth the effort.

Single-Dwell Linkages

There are two usual approaches to designing single-dwell linkages. Both result in **six-bar mechanisms,** and both require first finding a fourbar with a suitable coupler curve. A **dyad** is then added to provide an output link with the desired dwell characteristic. The first approach to be discussed requires the design or definition of a fourbar with a coupler curve that contains an approximate circle arc portion, which "are" occupies the desired portion of the input link (crank) cycle designated as the dwell. An atlas of coupler curves is invaluable for this part of the task. Symmetrical coupler curves are also well suited to this task, and the information in Figure 3-21 (p. 110) can be used to find them.

EXAMPLE 3-13

Single-Dwell Mechanism with Only Revolute Joints.

Problem: Design a sixbar linkage for 90° rocker motion over 300 crank degrees with dwell for the remaining 60°.

Solution: (see Figure 3-31)

I Search the H&N atlas for a fourbar linkage with a coupler curve having an approximate (pseudo) circle arc portion which occupies 60° of crank motion (12 dashes). The chosen fourbar is shown in Figure 3-3Ia.

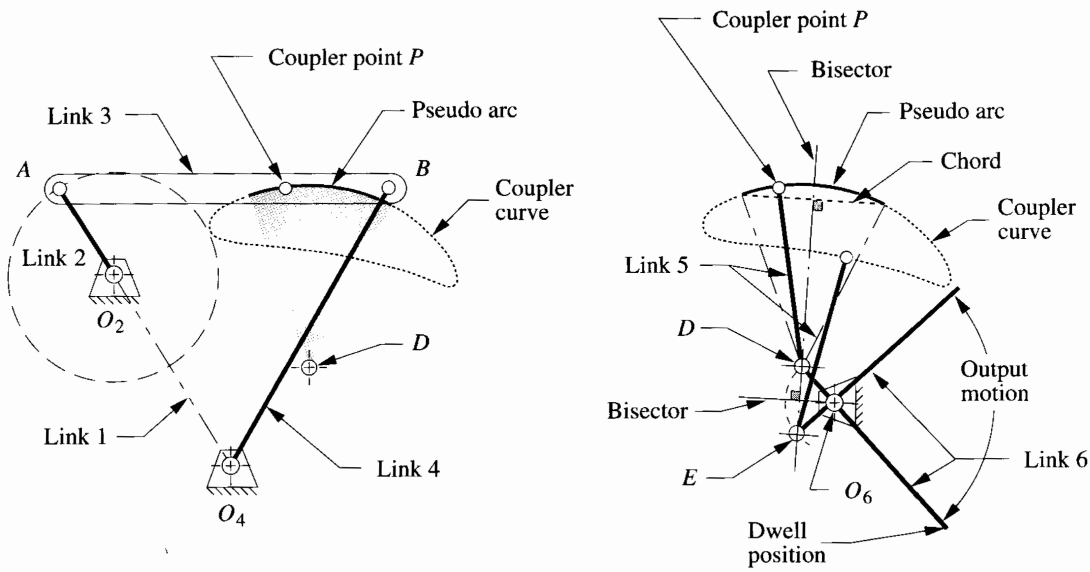
2 Layout this linkage to scale including the coupler curve and find the approximate center of the chosen coupler curve pseudo-arc using graphical geometric techniques. To do so, draw the chord of the arc and construct its perpendicular bisector as shown in Figure 3-31b. The center will lie on the bisector. Label this point D.

3 Set your compass to the approximate radius of the coupler arc. This will be the length of link 5 which is to be attached at the coupler point *P.*

4 Trace the coupler curve with the compass point, while keeping the compass pencil lead on the perpendicular bisector, and find the extreme location along the bisector that the compass lead will reach. Label this point *E.*

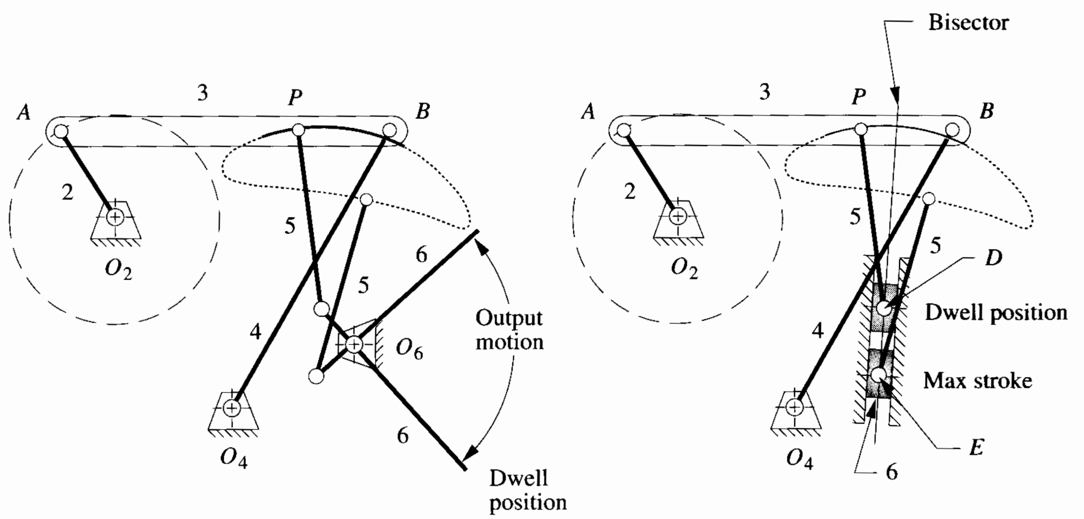
5 The line segment *DE* represents the maximum displacement that a link of length *CD,* attached at *P,* will reach along the bisector.

6 Construct a perpendicular bisector of the line segment *DE,* and extend it in a convenient direction.



(a) Chosen fourbar crank-rocker with pseudo-arc section for 60° of link 2 rotation

(b) Construction of the output-dwell dyad



(c) Completed sixbar single-dwell linkage with rocker output option

(d) Completed sixbar single-dwell linkage with slider output option

FIGURE 3-31

Design of a sixbar single-dwell mechanism with rocker output or slider output, using a pseudo-arc coupler curve

7 Locate fixed pivot O_6 on the bisector of DE such that lines O_6D and O_6E subtend the desired output angle, in this example, 90°.

8 Draw link 6 from D (or E) through O_6 and extend to any convenient length. This is the output link which will dwell for the specified portion of the crank cycle.

9 Check the transmission angles.

10 Make a cardboard model of the linkage and articulate it to check its function.

This linkage dwells because, during the time that the coupler point P is traversing the pseudo-arc portion of the coupler curve, the other end of link 5, attached to P and the same length as the arc radius, is essentially stationary at its other end, which is the arc center. However the dwell at point D will have some "jitter" or oscillation, due to the fact that D is only an approximate center of the pseudo-arc on the sixth-degree coupler curve. When point P leaves the arc portion, it will smoothly drive link 5 from point D to point E, which will in turn rotate the output link 6 through its arc as shown in Figure 3-31c (p. 127). Note that we can have any angular displacement of link 6 we desire with the same links 2 to 5, as they alone completely define the dwell aspect. Moving pivot 06 left and right along the bisector of line DE will change the angular displacement of link 6 but not its timing. In fact, a slider block could be substituted for link 6 as shown in Figure 3-31d, and linear translation along line DE with the same timing and dwell at D will result. Input the file F03-31c.6br to program SIXBARand animate to see the linkage of Example 3-13 in motion. The dwell in the motion of link 6 can be clearly seen in the animation, including the jitter due to its approximate nature.

Double-Dwell Linkages

It is also possible, using a fourbar coupler curve, to create a double-dwell output motion. One approach is the same as that used in the single-dwell of Example 3-11. Now a coupler curve is needed which has *two* approximate circle arcs of the same radius but with different centers, both convex or both concave. A link 5 of length equal to the radius of the two arcs will be added such that it and link 6 will remain nearly stationary at the center of each of the arcs, while the coupler point traverses the circular parts of its path. Motion of the output link 6 will occur only when the coupler point is between those arc portions. Higher-order linkages, such as the geared fivebar, can be used to create multiple-dwell outputs by a similar technique since they possess coupler curves with multiple, approximate circle arcs. See the built-in example double-dwell linkage in program SIXBAR for a demonstration of this approach.

A second approach uses a coupler curve with two approximate straight-line segments of appropriate duration. If a pivoted slider block (link 5) is attached to the coupler at this point, and link 6 is allowed to slide in link 5, it only remains to choose a pivot 06 at the intersection of the straight-line segments extended. The result is shown in Figure 3-32. While block 5 is traversing the "straight-line" segments of the curve, it will not impart any angular motion to link 6. The approximate nature of the fourbar straight line causes some jitter in these dwells also.

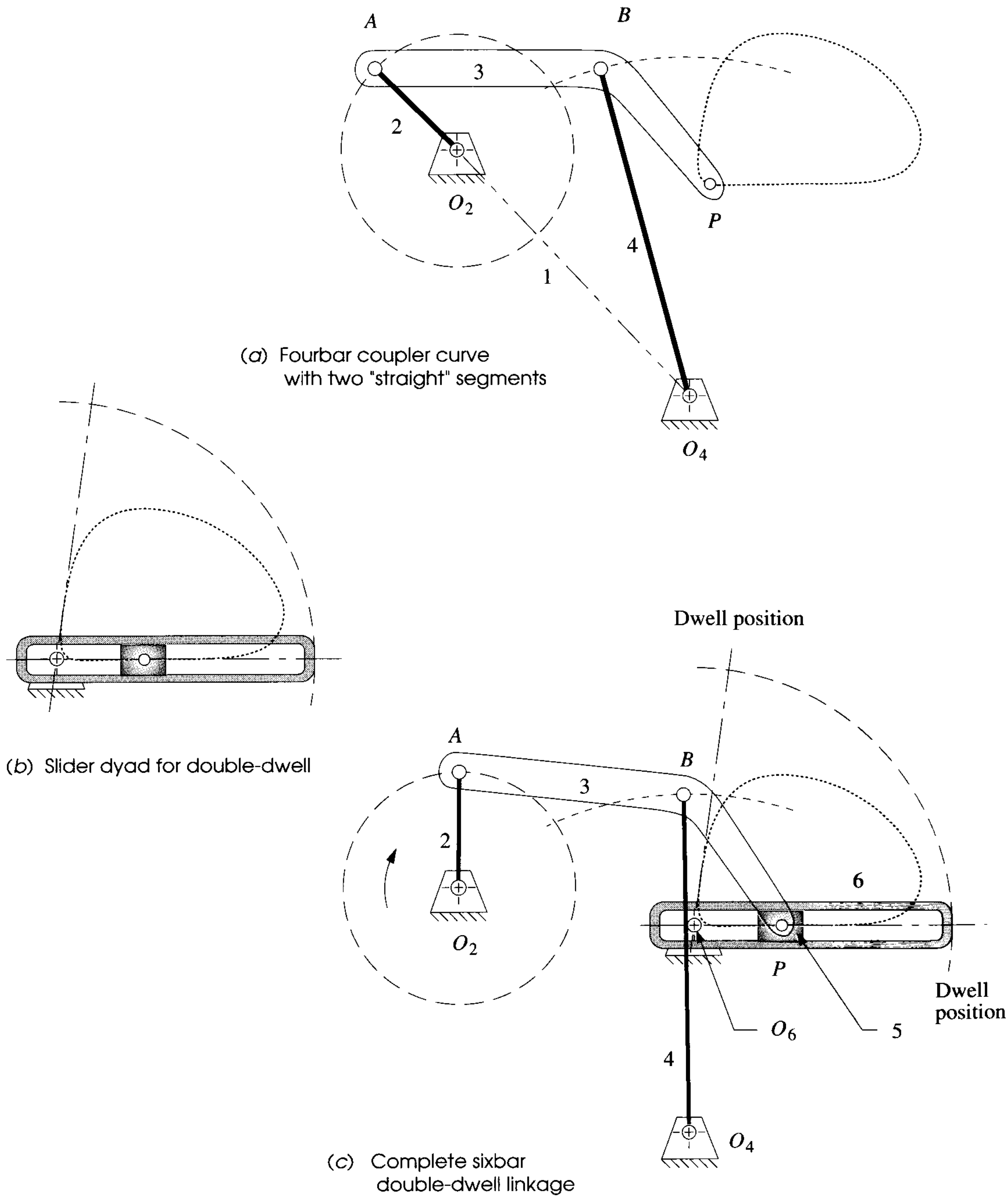


FIGURE 3-32

Double-dwell sixbar linkage

EXAMPLE 3-14

Double-Dwell Mechanism.

Problem: Design a sixbar linkage for 80° rocker output motion over 20 crank degrees with dwell for 160°, return motion over 140° and second dwell for 40°.

Solution: (see Figure 3-32, p. 129)

1 Search the H&N atlas for a fourbar linkage with a coupler curve having two approximate straight-line portions. One should occupy 160° of crank motion (32 dashes), and the second 40° of crank motion (8 dashes). This is a wedge-shaped curve as shown in Figure 3-32a.

2 Lay out this linkage to scale including the coupler curve and find the intersection of two tangent lines colinear with the straight segments. Label this point O_6.

3 Design link 6 to lie along these straight tangents, pivoted at O_6. Provide a slot in link 6 to accommodate slider block 5 as shown in Figure 3-32b.

4 Connect slider block 5 to the coupler point P on link 3 with a pin joint. The finished sixbar is shown in Figure 3-32c.

5 Check the transmission angles.

It should be apparent that these linkage dwell mechanisms have some disadvantages. Besides being difficult to synthesize, they give only approximate dwells which have some jitter on them. Also, they tend to be large for the output motions obtained, so do not package well. The acceleration of the output link can also be very high as in Figure 3-32 (p. 129), when block 5 is near pivot O_6. (Note the large angular displacement of link 6 resulting from a small motion of link 5.) Nevertheless they may be of value in situations where a completely stationary dwell is not required, and the low cost and high reliability of a linkage are important factors. Program SIXBAR has both single-dwell and double-dwell example linkages built in.

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3.12 PROBLEMS

*3-1 Define the following examples as path, motion, or function generation cases.

a. A telescope aiming (star tracking) mechanism
b. A backhoe bucket control mechanism
c. A thermostat adjusting mechanism
d. A computer printer head moving mechanism
e. An XY plotter pen control mechanism

3-2 Design a fourbar Grashof crank-rocker for 90° of output rocker motion with no quick-return. (See Example 3-1.) Build a cardboard model and determine the toggle positions and the minimum transmission angle.

*3-3 Design a fourbar mechanism to give the two positions shown in Figure P3-1 of output rocker motion with no quick-return. (See Example 3-2.) Build a cardboard model and determine the toggle positions and the minimum transmission angle.

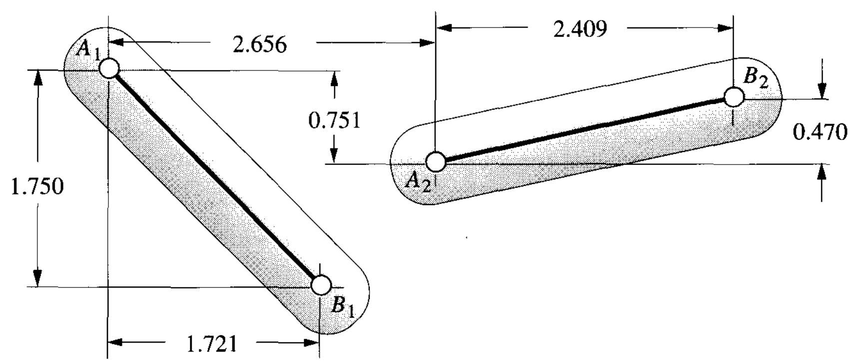

FIGURE P3-1
Problems 3-3 to 3-4

* Answers in Appendix F.