Chapter 4 The Creation of Mechanisms

4-1 Introduction*

The most difficult phase of mechanical design is, perhaps, the conceptual phase, i.e. the creation of a mechanism to satisfy a given functional requirement. One approach for the generation of concepts, as shown in Fig. 4.1 is to identify the overall function of a device based on the customer's requirements, and decompose it into subfunctions. Then, various concepts that satisfy each of the functions are generated and combined into a complete design. Generally, this procedure depends on the designer's intuition and experience. Another attempt to proceed this conceptual design is to generate atlases of mechanisms grouped according to their functional characteristics. And this remains the primary source of ideas for mechanism synthesis. However, this design methodology does not ensure the identification of all the design alternatives nor does it necessarily result in an optimum design.

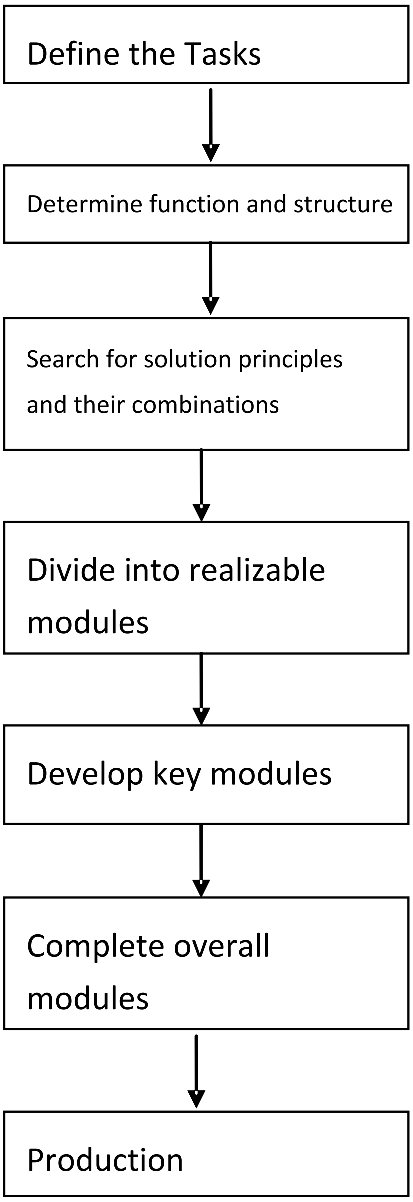


Figure 4.1 Traditional design process

Recently, a new approach which uses an abstract representation of the kinematic structure has been evolved. The kinematic structure contains the essential information about which link is connected to which other links by what types of joint. It can be conveniently represented by a graph and the graph can be enumerated systematically using combinatorial analysis and computer algorithms.

As shown in Fig. 4.2, the basic idea is based on the separation of mechanism structure from its function. First, the functional requirements of a class of mechanisms are identified. Then, kinematic structures of the same nature, i.e., same degree-of-freedom, number of links, and nature of the desired functions are enumerated in a systematic, unbiased manner using combinatorial analysis and graph theory. Third, each kinematic structure is sketched and evaluated according to functional requirements for the desired mechanism. Finally, a promising concept is chosen for dimensional synthesis, design optimization, computer simulation, and prototype demonstration. This process

* The material of this chapter is based on reference [1].

may be iterated several times until a final product is achieved. We summarize the methodology as follows:

Procedure for Systematic Creation of Mechanisms

(a) Identify the functional requirements, based on customer's requirements, of a class of mechanisms of interest
(b) Determine the nature of motion (i.e., planar, spherical, or spatial mechanism), degrees of freedom, type, and complexity of the mechanisms.
(c) Identify the structural characteristics associated with some of the functional requirements.
(d) Enumerate all possible kinematic structures that satisfy the structural characteristics using graph theory and combinatorial analysis.
(e) Sketch the corresponding mechanisms and evaluating each of them qualitatively in terms of its capability in satisfying the remaining functional requirements. This results in a set of feasible mechanisms.
(f) Select a most promising mechanism for dimensional synthesis, design optimization, computer simulation, and prototype demonstration.
(g) Enter the production phase.

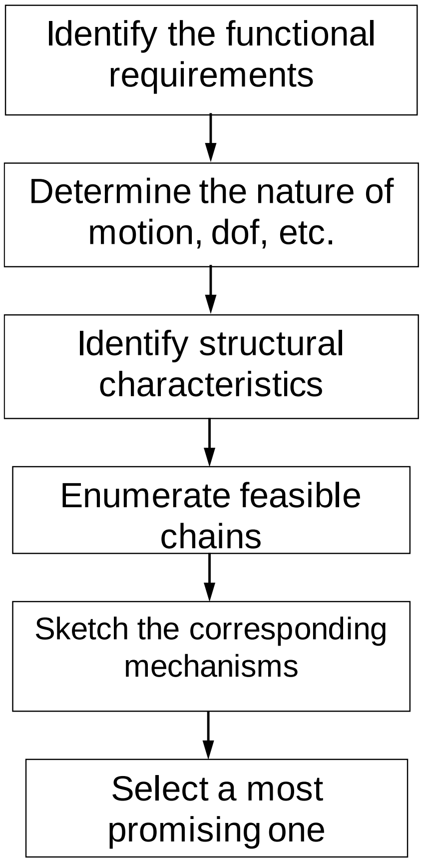


Figure 4.2 Procedure for mechanism creation according to function

We note that the methodology consists of two processors, generation and evaluation. In certain cases, some of the functional requirements are transformed into the structural characteristics and incorporated in the generation process as rules of enumeration. The generation processor enumerates all possible solutions with the aid of graph theory and combinatorial analysis. The remaining functional requirements are incorporated into the evaluation process as evaluation criteria for the selection of concepts. This results in a class of feasible mechanisms. Finally, a most promising candidate is chosen for the production design.

4-2 Planar Three-Link Mechanisms

Let us assume that we are interested in creating all planar three-link, one Degree-of-Freedom mechanisms with R, P, G and Cam (Cp) pairs. Then, we have $\lambda= 3$, $n= 3$, and $F= 1$. We find from graph atlas that only one graph No.1 satisfies the above condition

$$F = 3n - 2j - 3 + j_2 = 1 \qquad (4\text{-}1)$$

Or, $j_2 = 1$ (4-2)

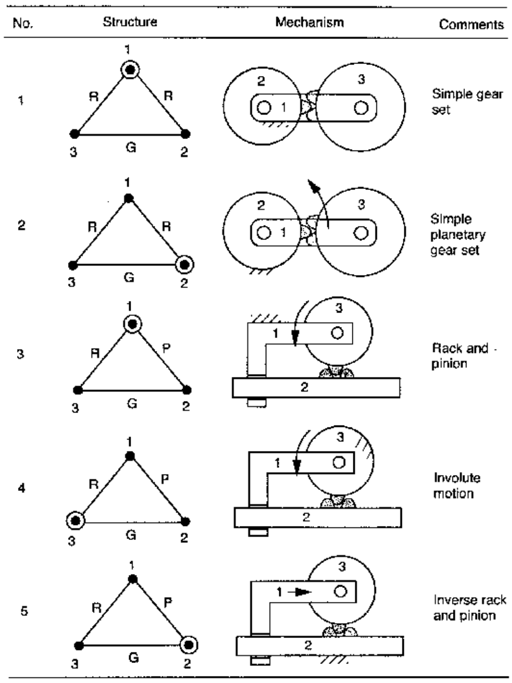
Therefore, only one gear pair or cam pair is allowed in a three-link chain, the other two joints must be R, P or their combinations, i.e. RR, PP, or RP. Making all the combinations of the j_1 joints with a j_2 joint pair, we obtain the following chains:

RRG, RRCp, RPG, RPCp, and PPCp, (PPG*)

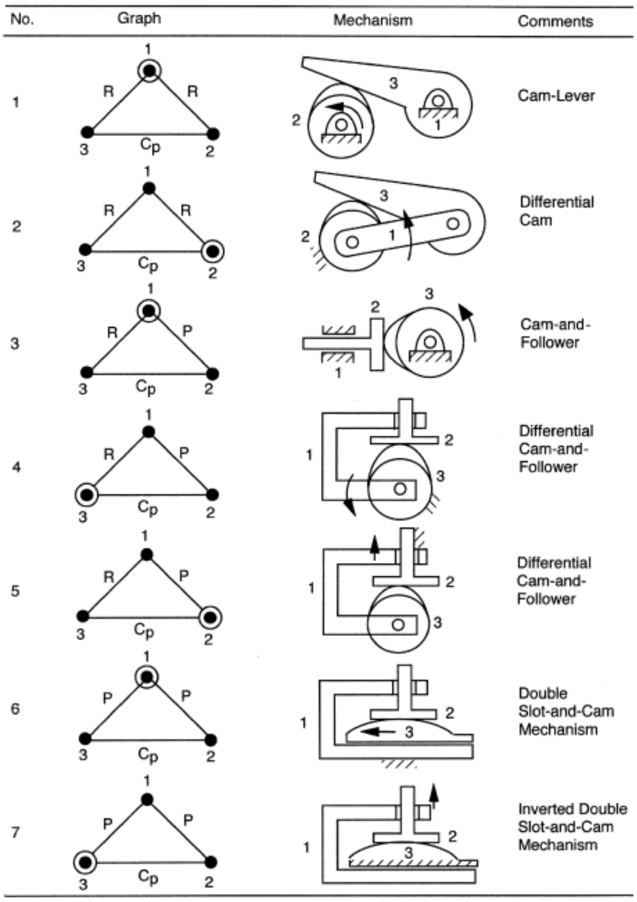
* PPG is excluded because it is judged impractical.

By choosing one of the three links as fixed link in as many different ways as possible, the resulting mechanisms are shown in Fig. 4.3. Notice that we have, in effect enumerated the number of kinematic chains as combinations of 3 objects (two of one specification and 1 of another). We then alternate the fixed link to obtain different mechanisms (kinematic inversion). We have excluded the non-circular gear pair and the disc-in-slot pair. If these high pairs are included, the number of three-link mechanisms will be approximately doubled.

It can be seen that there are no three-link planar mechanisms with just revolute, or prismatic, or a combination of both prismatic and revolute pairs, except for the three-link wedge mechanism discussed in Chapter 3.



(a) Geared three-link mechanism



(b) Three-link cam mechanisms

Figure 4.3 Three-link mechanisms

4-3 Planar Four-link mechanisms

From Table 3-4; there are two applicable graphs with four vertices. Graph No.2 (4,4) is a graph with single loop, and Graph No.3 (4,5) is a graph with two loops. Assume that we are interested in creating F=1 planar mechanisms, then we can enumerate mechanisms from these two graphs as follows

Case 1. Graph No.2 (4, 4)

In this case, n= 4, j= 4, L= 1, and F= 1. Substituting these into Eq. (3-20), yields j_2= 0. Hence, all joints must be either Revolute or Prismatic. Labeling the four edges with as many combinations of R and P as possible, we obtain the kinematic chains

RRRR, RRRP, RRPP, and RPRP

Note that the RPPP chain is excluded because it has two adjacent links with only sliding pairs. Choosing different fixed links results in seven mechanisms as shown in the Fig. 4.4. All of which are basic planar linkages.

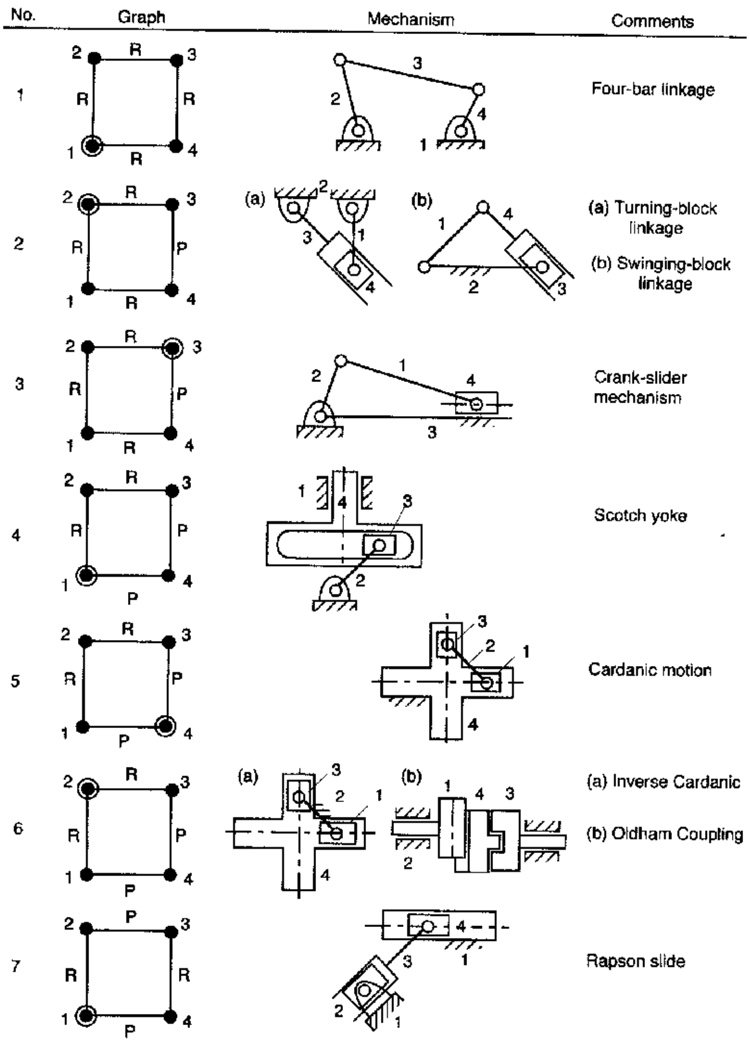


Figure 4.4 Four-bar linkages

The skill required in sketching the mechanism corresponding to a given structure is best illustrated in structure No. 6 inverse Cardanic motion and Oldham coupling. In this regard, creativity and ingenuity play an important role.

Case 2. Graph No.3 (4, 5)

Substituting n= 4, j= 5, and F= 1 into Eq. (3-20), becomes

$$j_2 = 2 \qquad (4\text{-}3)$$

Hence, there are two j_2 pairs and three j_1 pairs. Let us assume that only Revolute and Gear pairs are to be used. The problem becomes the determination of how many non-isomorphic ways we can label the graph with two G-edges and three R-edges.

In this simple case, the problem can be solved by inspection. There are four non-isomorphic graphs as shown in Figure 4.5. (Draw Figure 4.5)

4-4 Planar Six-Link Mechanisms

There are one (6, 6), three (6, 7), nine (6, 8) and thirteen (6, 9) graphs in Table 3-4. The search for all possible six-link mechanisms becomes a prohibited task. However, sometimes we would prefer certain type of mechanisms than others in an engineering application. Then, the task can be reduced to a manageable size. For example, if we limit ourselves to planar one-DOF linkages with revolute and prismatic joints, then the number of possible choices is considerably reduced. Substituting d= 3, n= 6, and j_2= 0 into Eq. 3-20, we have

$$j = j_1 = 7 \qquad (4\text{-}4)$$

This implies (6, 7) are needed for the creation of mechanism with six links and seven joints. There are three (6, 7) graphs and each has two loops. Exclude Graph No.15 for the triangular loop. The remaining two graphs are shown in the Figure 4.6. (Draw Figure 4.6)

4-5 Constant-Velocity Shaft Couplings

Constant-Velocity (CV) shaft couplings are used in automobiles and other machinery for transmitting power from one shaft to another to allow for small misalignments or relative motion between the two shafts.

<u>Functional requirement</u>

A mechanism for transmitting a one-to-one angular velocity ratio between two nonparallel intersecting shafts.

<u>Structural characteristics</u>

The principle of the C-V coupling is as follows

- one-dof mechanism
- one-to-one angular velocity ratio between the input and output shaft associated with a

symmetry of the coupling about a plane called the homokinetic plane, which bisects the two shaft axes perpendicularly.

Figure 4.7 shows the most elementary form of C-V coupling where the axes of two identical shafts intersect at a point O. The homokinetic plane is the plane passing through O, perpendicular to the paper, and bisecting the angle between the two shaft axes. As the shaft rotate, the contact point Q lies in the homokinetic plane for all phases.

Since the perpendicular distances from the contact point Q to the two shaft axes r_1 and r_2, are always equal to each other, the angular velocity ratio of the two shafts remains constant at all times. This mechanism is not very practical because it involves a five-dof higher pair.

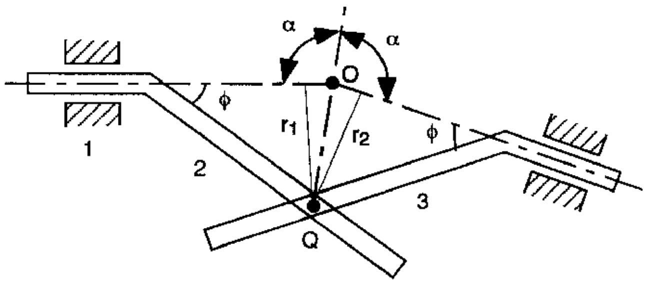


Figure 4.7 Elementary form of C-V coupling

There are two basic types of C-V shaft couplings: Ball type and linkage type. The ball type is characterized by point contact between the balls and their races in the yokes of the P shafts, whereas the linkage type is characterized by surfaces contact between the links. In the following, we limit ourselves to the linkage type. The structural characteristics can be summarized as follows:

1. Type of mechanism: spatial single-loop linkage
2. Degree of freedom: F = 1
3. Symmetrical about a homokinetic plane
4. Available joint types: R, P, C, S, and E

Enumeration of C-V Shaft Couplings

Figure 4.8 shows the general configuration of a C-V shaft coupling, where the fixed link is denoted as link 1, the input link as link 2, and the output link as link 3. Since we are interested in single-loop shaft couplings, the number of links is equal to the number of joints and all links are necessarily binary. The loop mobility equation requires that

$$\sum f_i = 7 \qquad (4\text{-}5)$$

Since the minimum degree of freedom in any joint is one, the number of joints and the number of links should not exceed 7;

$$n = j \leq 7 \qquad (4\text{-}6)$$

Since the first and last joints are preassigned as revolute joints and the mechanism is symmetrical about the homokinetic plane, the number of links (and joints) should be odd:

$n = j = 3$, 5 , or 7

The case $n = 3$ requires a five-dof joint, which is judged to be impractical. Hence

$n = j = 5$ or 7

The graph representations of these two families of mechanisms are sketched in Figure 4.9. The two ground connected joints are prelabeled as revolute joints. The other joint types are labeled symmetrically with respect to the fixed link as X and Y for the five-link chain, and X, Y, and Z for the seven-link chain. Let the degrees of freedom associated with the X, Y, and Z joints be denoted by f_x, f_y, and f_z, respectively. The enumeration of each family of C-V shaft couplings is discussed as follows.

<u>Five-link C-V Shaft Couplings</u>

Solving the joint dof for unknown joints X and Y,

$$2f_x + f_y = 5 \quad (4\text{-}7)$$

yields $(f_x , f_y) = (1, 3)$ or $(2, 1)$. Labeling the graph with these joint distributions results in six distinct mechanisms

RRERR (Tracta couplings)
RRSRR (Clements couplings)
RPEPR
RPSPR (Altmann)
RCRCR (Myard)
RCPCR

<u>Seven-link C-V Shaft Couplings</u>

$$2f_x + 2f_y + f_z = 5 \quad (4\text{-}8)$$

We have only one solution to the above equation as

$f_x = f_y = f_z = 1$

Labeling the graph with the joint distribution yields

RRRRRRR (Myard, Voss, Wachter and Reiger)
RRRPRRR
RRPRPRR (Derby, S.W. Industries)
RPRRRPR
RPRPRPR

Overall, a total of 12 kinematic structures of C-V shaft couplings are found. Six well-known C-V shaft couplings are sketched in the Figure 4.10.

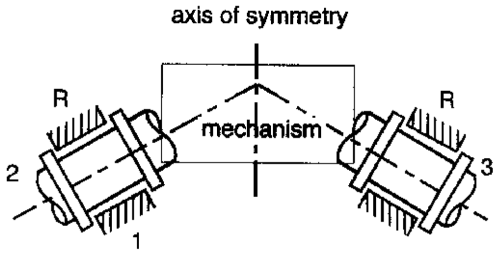


Figure 4.8 General configuration of a C-V shaft coupling

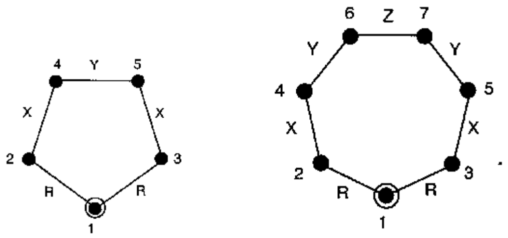


Figure 4.9 Graph representations of two families of mechanisms

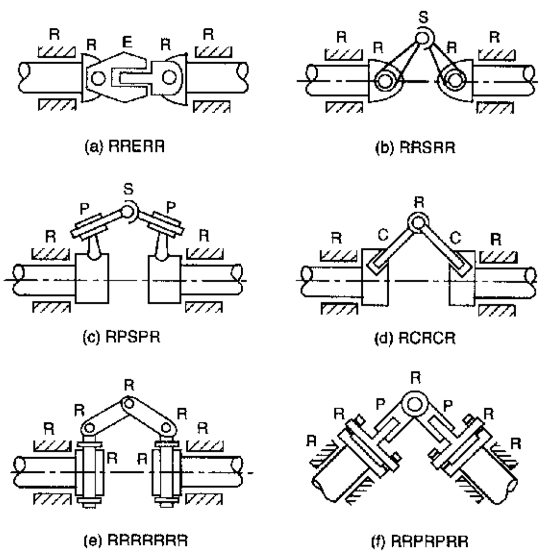


Figure 4.10 Five-link and seven-link C-V coupling

4-6 Design of a Mechanical Hand [2]

Initiative requirement: In a general condition, an object is to be manipulated with five (5) motions, i.e. three translations and two rotations. As shown in the Figure 4.11 (Draw Fig.4.11), one rotation is desired with respect to a major axis of the object and the second rotation is with respect to an axis perpendicular to that major axis. In a commercially available robot arm, the wrist always can provide the third rotation and it is, therefore excluded from the design objectives here.

Design Methodology

1. Search through a library of hands

   This method will be efficient, if an appropriate search algorithm is incorporated in the system. This algorithm should identify designs in the library which satisfy some specified set of requirements.
2. Design of a new hand – when a library search fails, an efficient design technique needs to be developed to aid in design of mechanical hands with the desired properties. Following are the integrated phases of mechanical hand design.
   - Structural and Dimensional Design
   - Development of a Control System
   - Selection and development of sensors
   - Design of mechanical actuators

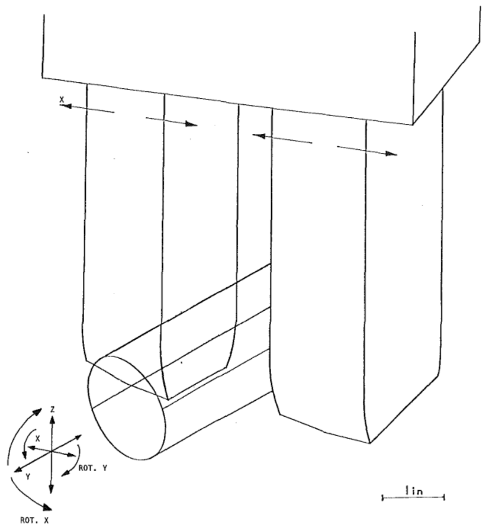


Fig. 4.11 Schematic of a mechanical hand with five dof

Structural Design of Hands with Manipulative Capabilities

- Two-fingered mechanical hands are considered (as shown in Fig. 4.11.)
- They can translate independently of each other, to provide for both grasping of different object sizes and for required translation manipulation along the line of grasping. If excluding the translating movement of fingers, then the remaining object manipulations are two rotations, ROT X and ROT Y and translation in the plane Y-Z. Then, the number of degrees of freedom of object motion is then four.
- Assume each finger will be equipped with identical mechanisms for object manipulation, we then conclude that the mechanism on each finger needs to have at least two degrees of freedom.
- Only consider mechanisms which satisfy the general degree-of-freedom equation.

Mobility equation:

$$F= 3(n-j-1)+ 2j_g+ j_p+ j_r \qquad (4\text{-}9)$$

F (=2), n, j, j_g, j_p, j_r are defined as such.

Loop equation:

$$L= j- n+ 1 \qquad (4\text{-}10a)$$

$$j= j_g+ j_p+ j_r \qquad (4\text{-}10b)$$

The number of links, n, and the number of joints, j, can be expressed in terms of L and j_g

$$j= 3L- j_g+ 2 \qquad (4\text{-}11a)$$

$$n= 2L- j_g+ 3 \qquad (4\text{-}11b)$$

Eqs.(4-11) depend on the number of independent loop, L and the number of gear pairs j_g

For a specified number of gear pairs, one can show that

$$j_{L+1}=j_L +3 \qquad (4\text{-}12a)$$

$$n_{L+1}=n_L+2 \qquad (4\text{-}12b)$$

It is felt that more than two loops and more than two gear pairs would add unnecessary complexity to the search. Thus, we examine mechanisms with up to two gear pairs only and with up to two loops.

Case1: L= 1, j_g= 1,

From Eq.(4-11), j= 4 and n= 4. It is the basic planetary gear train. This case is sketched with spur and bevel gears.

Case2: L= 1, j_g= 2

From Eq.(4-11), j= 3, n= 3. The case has three links and three joints, which is not acceptable since each loop should have only one gear pair.

Case3: L= 2, j_g= 1

$j= 3\times2- 1+ 2= 7$

$n= 2\times2- 1+ 3= 6$

The case represents six-link mechanisms with seven kinematic pairs, one of which is gear pair. There are two types of six-link kinematic chains. They are sketched by labeling the graph representation of these kinematic chains in all possible ways by designing the type of kinematic pairs, P, R, or G, and by identifying the fixed link. (refer to Fig. 4.12a ~ 4.12g)

Case4: L= 2, j_g= 2

$j= 3\times2- 2+ 2= 6$

$n = 2\times2 - 2 + 3 = 5$

(refer to Fig. 4.13a ~ 4.13c)

The number of inversions alone each mechanism sketched will be over 200. This number is large. It is then necessary to formulate some rules or criteria based on which some cases will not be examined. (Usually, these rules require intuition, good understanding of the mechanism function, and in general they involve kinematic structure specifications. For example, they include specifications of mechanism type, plane or spatial, the type of kinematic pairs, and limits on the specific type of kinematic pairs.)

Some design rules are defined as follows

1. Consider only plane mechanisms that obey general DOF equation, and which have six links and seven joints, one of which is a gear pair, or have five links and six joints, two of which are gear pairs, with the other joints being R and P.
2. Gear rack is not permitted for six-link mechanisms.
3. One gear rack only is permitted for five-link mechanism.
4. One prismatic pair only is permitted to provide the required translation manipulation.
5. Revolute pairs between gears are permitted only if gears are concentric, or if one gear is the frame.
6. Gear rack cannot have R-pairs.
7. One gear is permitted in any independent loop.

The design criteria are used to screen out some undesirable designs. The number of mechanisms that satisfy the criteria outlined above is usually large. One can now impose mechanism performance criteria to identify an optimum design with respect to these performance criteria. These specifications may involve size restrictions, smooth and infinite motions, ease of locating power actuators, and others. Following are the outlines of the performance criteria for optimum selection of mechanisms for object manipulation.

Criteria for optimum selection

1. Both translation and rotation should be provided as a direct output and controlled independently of each other.
2. The output translation and rotation should be easy to control.
3. The output displacement should be infinite. This ensures versatility and eliminates position initialization procedures.
4. Both drive motors should be fixed on the frame for minimum inertia, and largely for size restrictions.
5. The output translation and rotation should be smooth in terms of velocity variation assuming reasonable driving characteristics.

Optimum mechanism selection for object manipulation

1. Cases with only R- and G- pair are rejected, since a translation output is not directly available in the mechanism.
2. Special straight-line motions, such as cardanic motion in internal gearing are rejected, since they violate criteria 1 and 3.
3. Mechanisms involving crank-and-slider and its inversions are rejected. This reduces six-link mechanisms with one gear pair and one prismatic pair. (Due to velocity variations of the slides for constant input speed and cause problems in secure object grasping during manipulation; furthermore, an initialization procedure would have to be invoked for the slider which may add to the complexity of the control system; and limited stroke for slider motion.)
4. For five-link mechanisms, some were rejected because they can not be driven by actuators fixed on the frame; or translation and rotation cannot be controlled independently.
5. No.8 in five-link mechanism is optimum since rack 2 can be driven for any specified displacement by driving gear 4, while the drive of arm 1 is locked. In addition, arm 1 can be driven directly to rotate for any specified amount, but gear 4 has to be driven simultaneously to prevent translation of link 2, if rotation only is desired. In normal operation, combined motion occurs to rotate and translate link 2 by prespecified amounts.

Functional representation

1. Motors can be installed on link 4 (to drive gear rack) and on link 1 (to rotate the mechanism. (Fig. 4.14 ~4.16)
2. Translation in Y-Z plane and rotation about X-axis can be obtained.
3. Rotation about Y-axis — belts translate in opposite sense with respect to each other to roll the object.

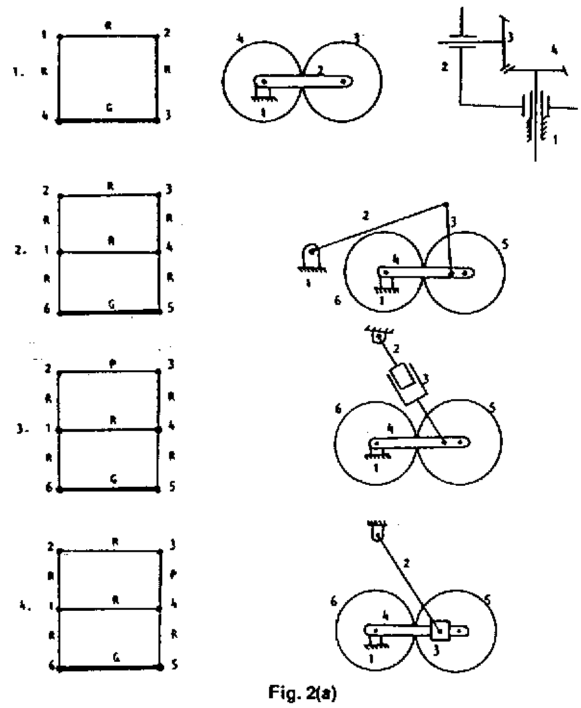
Reference

1. Tsai, L.W. *Mechanism Design- Enumeration of Kinematic Structures According to Function*, CRC Press, 2001.

2. Datseris, P. and Palm, W., "Principles on the Development of Mechanical Hands Which Can Manipulate Objects by Means of Active Control," ASME J. of Mechanisms, Transmissions, and Automation in Design, 107, June 1985, pp. 148-156.

Fig. 4.12(a)

Fig. 4.12(b)



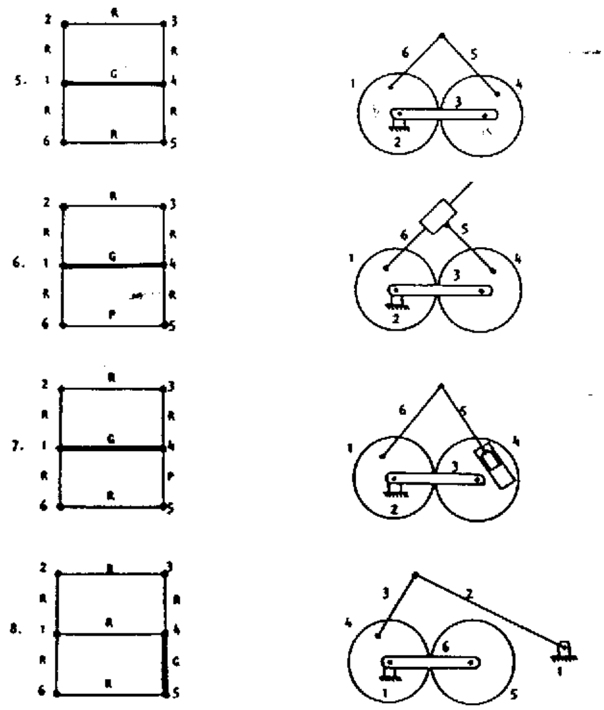


Fig. 4.12(e)

Fig. 4.12(f)

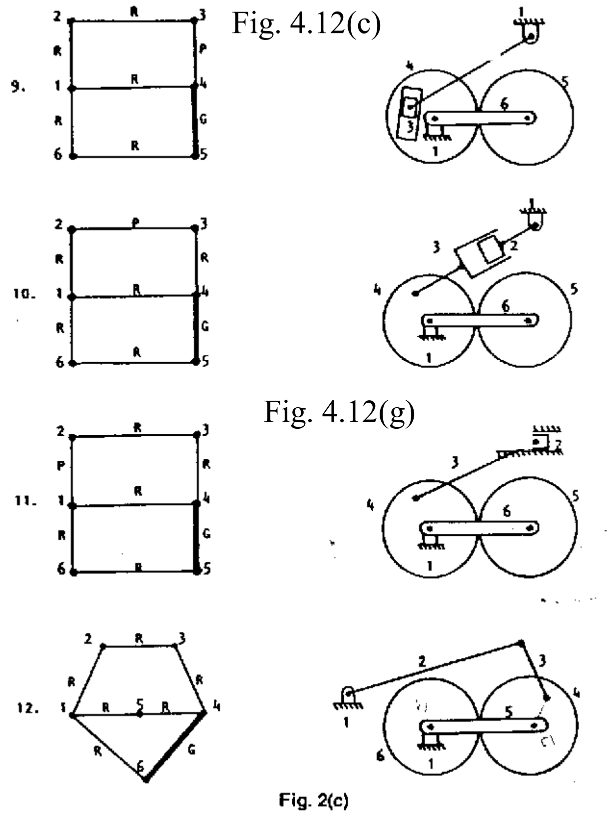


Fig. 4.12(c)

Fig. 4.12(g)

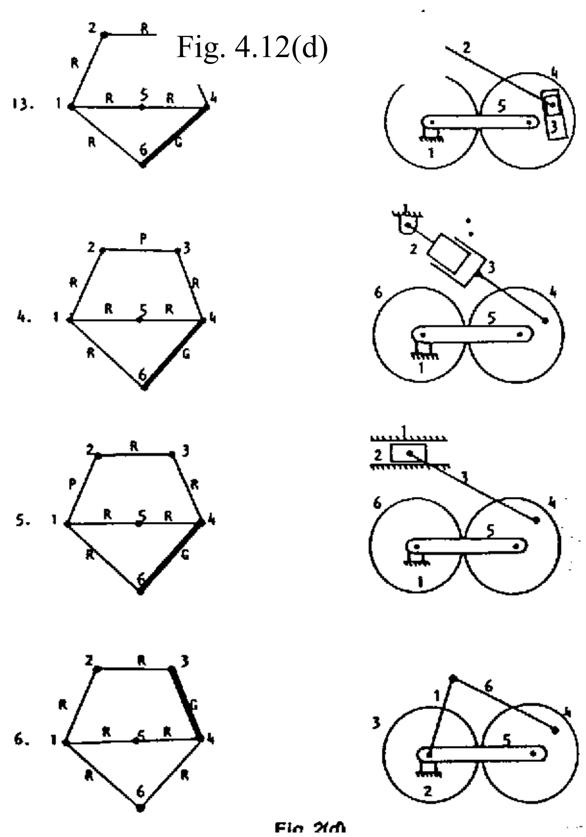


Fig. 4.12(d)

Fig. 4.13(a)

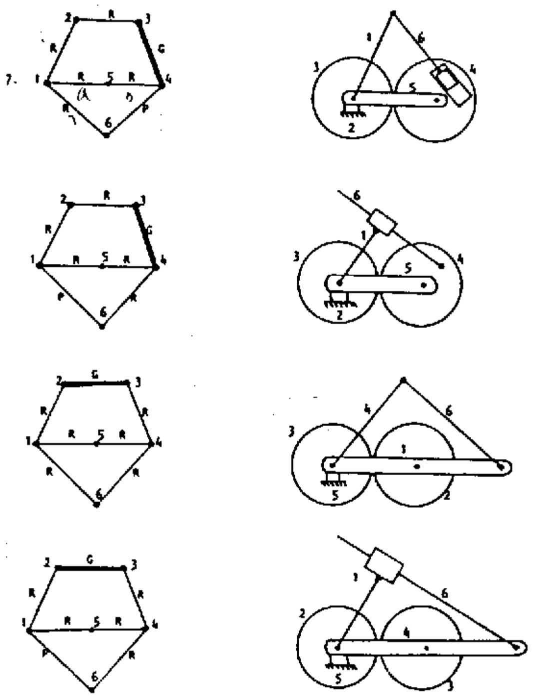


Fig. 2(e)

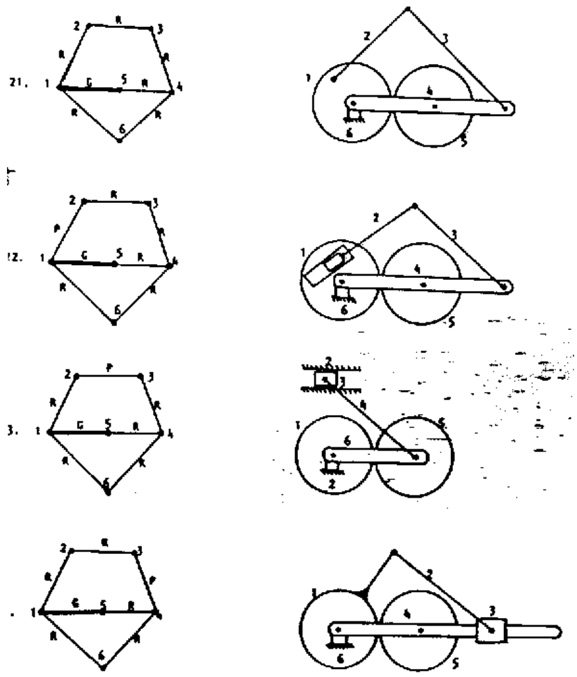


Fig. 2(f)

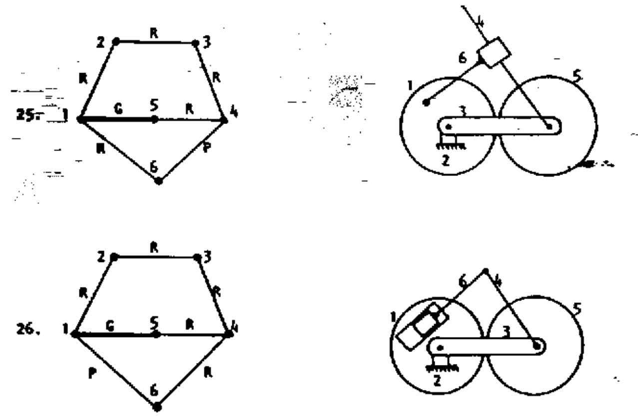


Fig. 2(g)

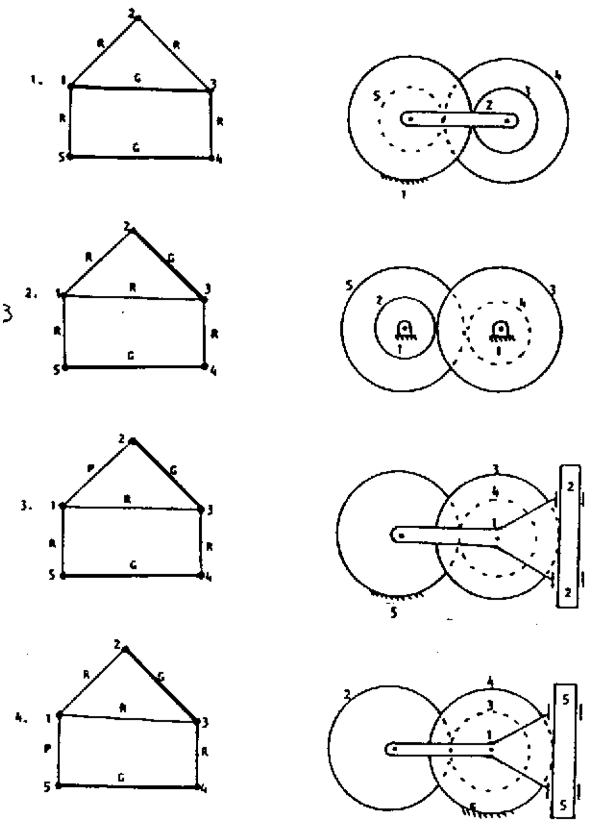


Fig. 3(a)

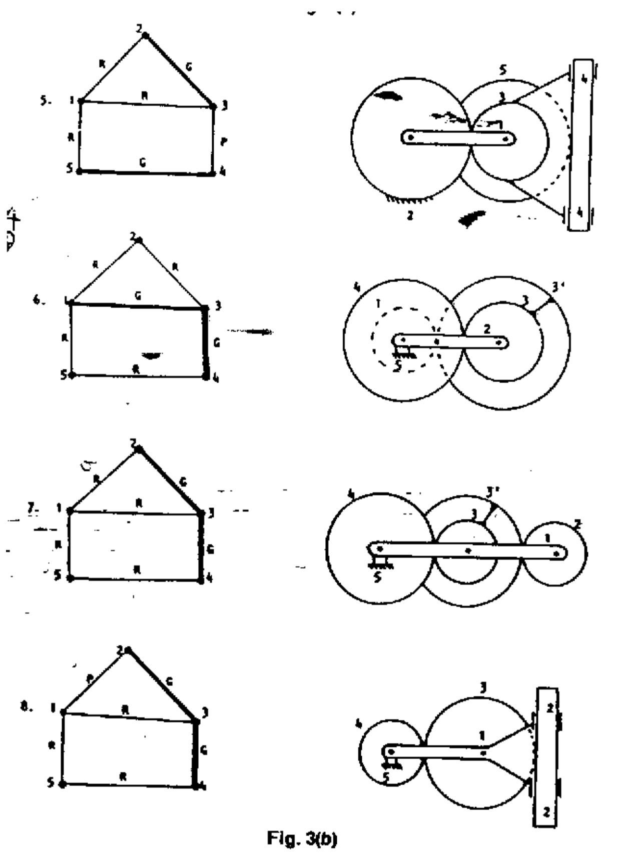


Fig. 4.13(b)

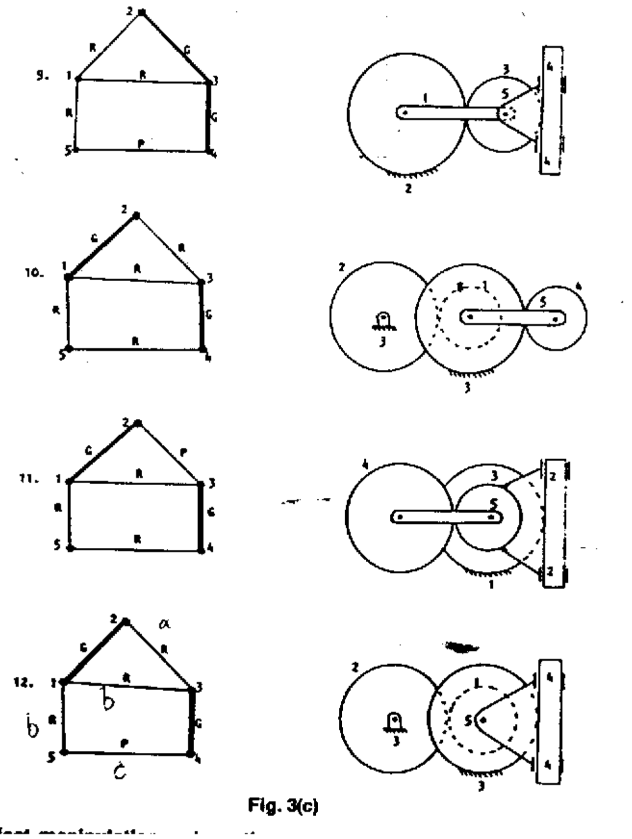


Fig. 4.13(c)

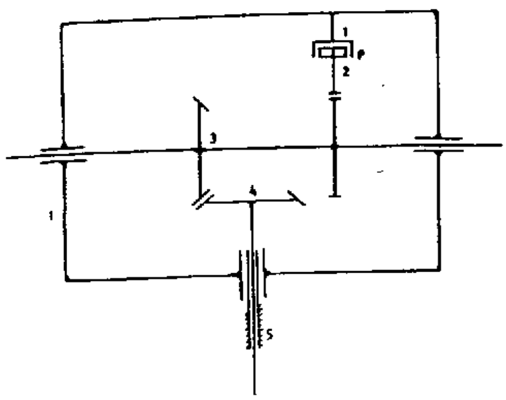


Figure 4.14 Optimum mechanism for manipulation

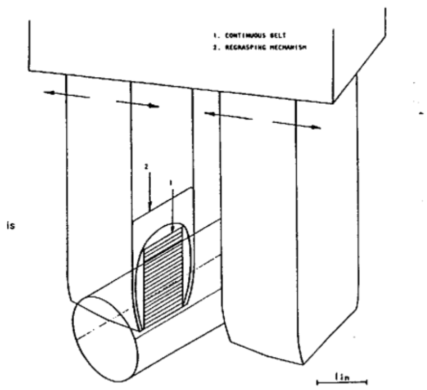


Figure 4.16 Schematic of a hand design

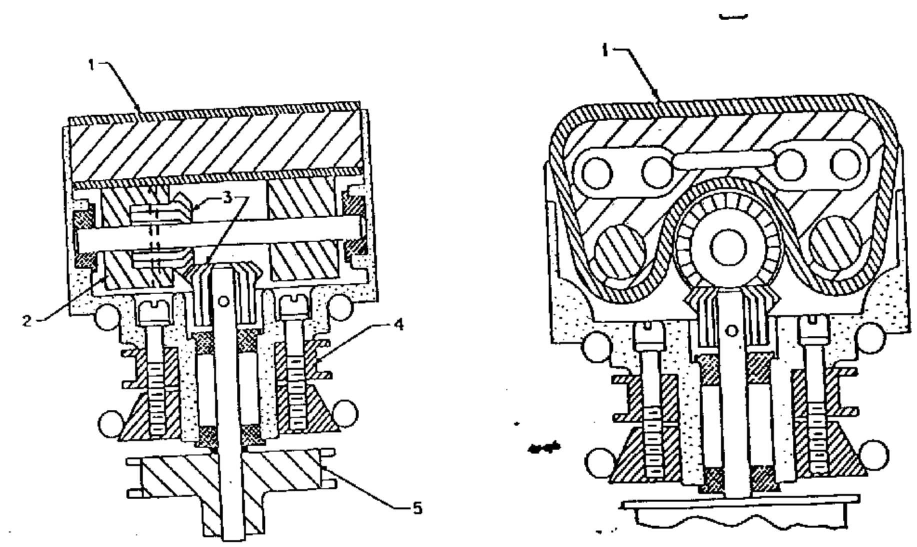


Fig. 4.15 Two views of detailed drawing of mechanisms for manipulation, Case 8 from Fig. 4.13(b)