FORMAL ENGINEERING DESIGN SYNTHESIS

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Chapter 9

KINEMATIC SYNTHESIS

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Abstract

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1. Introduction

This chapter surveys recent results in the kinematic synthesis of machines. A machine is generally considered to be a device that directs a source of power to a desired application of forces and movement. The earliest machines were simply wedges, levers, and wheels that amplified human and animal effort. Eventually wind and water power were captured to drive gearing that rotated millstones and pumps (Dimarogonas 1993). Today chemical, nuclear, hydro-electric and solar energy drive machines that build, manufacture, transport, and process items that affect all aspects of our lives.

Kinematic synthesis determines the configuration and size of the mechanical elements that shape this power flow in a machine. In this chapter we show how researchers in machine design and robotics use the concepts of workspace and mechanical advantage as the criteria for the kinematic synthesis of a broad range of machines.

The workspace of a machine part is the set of positions and orientations that it can reach. We will see that there are many representations of this workspace, however the best place to start is with the kinematics equations of the chain of bodies that connect the part to the base frame. The range of values of the configuration parameters that appear in these kinematics equations define the configuration space of the system. The velocity of the part is then obtained by computing the Jacobian of these kinematics equations in terms of the rate of change of these configuration parameters. The principle of virtual work links this Jacobian directly to the mechanical advantage of the system.

We begin by showing how virtual work relates the input-output speed ratios of a system to its mechanical advantage. This is developed for general serial and parallel chains of parts within a machine. Then we explore the current design theory for serial and parallel robots and find that workspace and mechanical advantage, clothed in various ways, are the primary considerations. We then develop in detail the current results in the kinematic synthesis of constrained movement which focusses on satisfying workspace constraints formulated as algebraic equations. Finally, we outline the configuration space analysis of machines that shows the versatility of these concepts for representing assembly, tolerances and fixtures.

A survey of computer-aided kinematic synthesis software shows that while specialized systems exist in research laboratories, there is very little available in engineering practice. Also, current design systems leave untouched many of the machine topologies that are available for invention. Designer/inventors need a software environment that allows them to specify a workspace and an associated distribution of mechanical advantage, and then simulate and display candidate designs to evaluate their performance, including issues of tolerancing and assembly. The result would be a remarkable opportunity for the design of new devices to serve our needs.

2. Kinematics and Kinetics

The term *kinematics* refers to geometric properties of movement such as position, velocity, and acceleration of components of a machine, while *kinetics* refers properties of forces acting on and exerted by the part. Newton's second law of motion relates kinetics and kinematics by defining the acceleration of a particle as proportional to the difference between the force applied to the particle and the force it exerts,

$$\mathbf{F}_{\text{in}} - \mathbf{F}_{\text{out}} = m\ddot{\mathbf{x}}. \tag{9.1}$$

If these forces are equal then the particle moves at a constant velocity.

Newton's law can summed over all particles in the machine and integrated as the system moves along a trajectory $\mathbf{x}(t)$ to define the change in energy of the system as the difference in the input and output work. It is a law of mechanics that this equality of work and energy remains unchanged for small variations $\delta\mathbf{x}$ of the trajectory. Thus, the variations in work and energy must cancel for all virtual displacements, that is

$$\delta W_{\text{in}} - \delta W_{\text{out}} = \delta E. \tag{9.2}$$

A machine is designed to minimize energy losses, often due to friction and fatigue, so $\delta E = 0$, which means the input and output variations in work must cancel. This is known as the *principle of virtual work* (Greenwood 1977, Moon 1998).

At a specific instant of time, we can introduce the virtual displacement $\delta\mathbf{x} = \mathbf{v}\delta t$, where $\mathbf{v}$ is the velocity and δt is a virtual time increment. This allows us to define the virtual work $\delta W_{\text{in}} = P_{\text{in}}\delta t$ and $\delta W_{\text{out}} = P_{\text{out}}\delta t$ where $P = \mathbf{F}\cdot\mathbf{v}$ is the instantaneous power. The result is that Equation (9.2) becomes

$$(P_{\text{in}} - P_{\text{out}})\delta t = 0. \tag{9.3}$$

Thus, the usual assumption is that the machine does not dissipate power and in every configuration the input and output instantaneous power are equal. Because power is force times velocity, we have that *the mechanical advantage of a machine is the inverse of its speed ratio.* In the next section we examine this in more detail.

2.1 Machine Topology

A machine is constructed from a variety of elements such as gears, cams, linkages, ratchets, brakes, and clutches. Each of these elements can be reduced to a set of *links* connected together by *joints*. Perhaps the simplest joint to construct, though difficult to analyze, is the cam-and-follower formed by one link pushing against a second follower link. In this case, the movement of the output link depends on the shape of the contacting surfaces. In contrast, pure rotary and sliding joints, and joints constructed from them, are considered simple joints because they are easy to analyze, though difficult to construct. These two classes of joints are often termed *higher* and *lower* pairs, respectively, (Reuleaux 1875, Waldron and Kinzel 1998).

Each component M of a machine is connected by a series of links and joints to the base frame F of the device, and the topology of the system is presented as

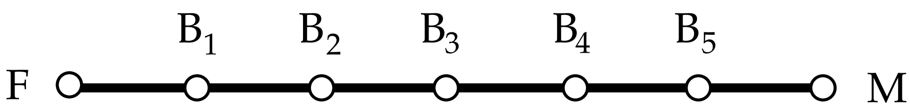


Figure 9.1 The part M is connected to the base frame F by a series of links (vertices) and joints (edges).

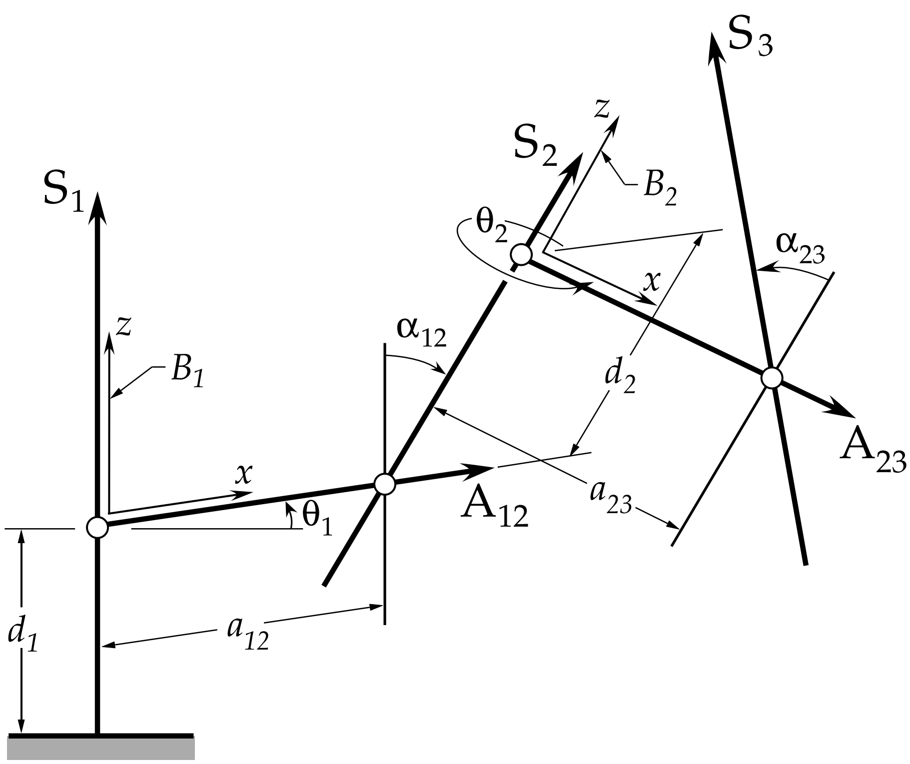


Figure 9.2 A machine part is located in space by a sequence of frames consisting of axes S_j and their common normals A_{ij}.

a graph with the links as vertices and joints as edges (Kota 1993). For example, the typical robot arm has the graph shown in Figure 9.1. The mathematical relation that defines the position of each machine part M in the frame F is called its *kinematics equations*.

2.2 Kinematics Equations

If we model the local geometry of higher-pair joints using rotary and sliding joints, then the position of every component of a machine can be obtained from a sequence of lines representing the axes S_j of equivalent revolute or prismatic joints. Between successive joint axes, we have the common normal lines A_{ij} which together with the joint axes forms a serial chain, Figure 9.2. This construction allows the specification of the location of the part relative to the base of the machine by the matrix equation

$$[D] = [Z(\theta_1, \rho_1)][X(\alpha_{12}, a_{12})][Z(\theta_2, \rho_2)]...[X(\alpha_{m-1,m}, a_{m-1,m})][Z(\theta_m, \rho_m)], \tag{9.4}$$

known as the *kinematics equations* of the chain (Paul 1981, Craig 1989). The set of all positions $[D]$ obtained as the joint parameters vary over their range of movement defines the *workspace* of the component, also called its *configuration space* (Greenwood 1977, Arnold 1978).

The matrices $[Z(\theta_j, \rho_j)]$ and $[X(\alpha_{ij}, a_{ij})]$ are 4×4 matrices that define screw displacements around and along the joint axes S_j and A_{ij}, respectively (Bottema and Roth 1979). The parameters α_{ij}, a_{ij} define the dimensions of the links in the chain. The parameter θ_j is the joint variable for revolute joints and ρ_j is the variable for prismatic joints. The trajectory $\mathbf{P}(t)$ of a point $\mathbf{p}$ in any part of a machine is obtained from the joint trajectory, $\vec{\theta}(t) = (\theta_1(t), \dots, \theta_m(t))^T$, so we have

$$\mathbf{P}(t) = [D(\vec{\theta}(t))]\mathbf{p}. \tag{9.5}$$

A single part is often connected to the base frame by more than one serial chain, Figure 9.3. In this case we have a set of kinematics equations for each chain,

$$[D] = [G_j][D(\vec{\theta}_j)][H_j], \quad j = 1, \dots, n, \tag{9.6}$$

where $[G_j]$ locates the base of the jth chain and $[H_j]$ defines the position of its attachment to the part. The set of positions $[D]$ that simultaneously satisfy all of these equations is the workspace of the part. This imposes constraints on the joint variables that must be determined to completely define its workspace (McCarthy 1990, Tsai 1999).

2.3 Configuration Space

Configuration space is the set of values available to configuration parameters of a mechanical system. For a serial chain it is the set of values available $\vec{\theta}$. Configuration space is a fundamental tool in robot path planning for obstacle avoidance (Lozano-Perez 1983). Though any link in the chain forming a robot may hit an obstacle, it is the gripper that is intended to approach and move around obstacles such as the table supporting the robot and the fixtures for parts it is to pick up. Obstacles define forbidden positions and orientations in the workspace which map back to forbidden joint angles in the configuration

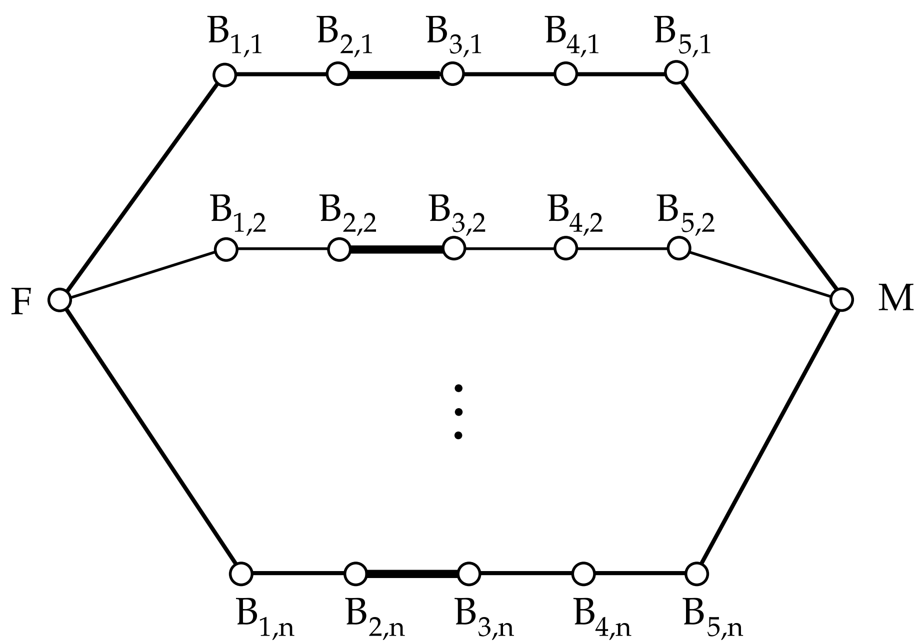


Figure 9.3 A part M is often connected to the base frame F by parallel series of links and joints.

space of the robot. Robot path planners seek trajectories to a goal position through the free space around these joint space obstacles (Latombe 1991).

The map of joint space obstacles provides a convenient illustration of the movement available to mechanical parts that are in close proximity (Joskowicz and Sacks 1999). For this reason it has found applications far from robot path planning, such as the modeling tolerances, assembly and fixtures.

2.4 Speed Ratios

The speed ratio for any component of a machine relates the velocity $\dot{\mathbf{P}}$ of a point $\mathbf{P}$ to the joint rates $\dot{\vec{\theta}} = (\dot{\theta}_1, \ldots, \dot{\theta}_m)^T$. The velocity of this point is given by

$$\dot{\mathbf{P}} = \mathbf{v} + \vec{\omega} \times (\mathbf{P} - \mathbf{d}), \tag{9.7}$$

where $\mathbf{d}$ and $\mathbf{v}$ are the position and velocity of a reference point and $\vec{\omega}$ is the angular velocity of the part.

The vectors $\mathbf{v}$ and $\vec{\omega}$ depend on the joint rates $\dot{\theta}_j$ by the formula

$$\begin{Bmatrix} \mathbf{v} \\ \vec{\omega} \end{Bmatrix} = \begin{bmatrix} \frac{\partial \mathbf{v}}{\partial \dot{\theta}_1} & \frac{\partial \mathbf{v}}{\partial \dot{\theta}_2} & \cdots & \frac{\partial \mathbf{v}}{\partial \dot{\theta}_m} \\ \frac{\partial \vec{\omega}}{\partial \dot{\theta}_1} & \frac{\partial \vec{\omega}}{\partial \dot{\theta}_2} & \cdots & \frac{\partial \vec{\omega}}{\partial \dot{\theta}_m} \end{bmatrix} \begin{Bmatrix} \dot{\theta}_1 \\ \vdots \\ \dot{\theta}_m \end{Bmatrix}, \tag{9.8}$$

or

$$\mathsf{V} = [J]\dot{\vec{\theta}}. \tag{9.9}$$

The coefficient matrix $[J]$ in this equation is called the *Jacobian* and is a matrix of speed ratios relating the velocity of the part to the input joint rotation rates (Craig 1989, Tsai 1999).

2.5 Mechanical Advantage

If the machine component exerts a force $\mathbf{F}$ at the point $\mathbf{P}$, then the power output is

$$P_{\text{out}} = \mathbf{F} \cdot \dot{\mathbf{P}} = \sum_{j=1}^{m} \mathbf{F} \cdot \left(\frac{\partial \mathbf{v}}{\partial \dot{\theta}_j} + \frac{\partial \vec{\omega}}{\partial \dot{\theta}_j} \times (\mathbf{P} - \mathbf{d}) \right) \dot{\theta}_j. \tag{9.10}$$

Each term in this sum is the portion of the output power that can be associated with an actuator at the joint S_j, if one exists.

The power input at joint S_j is the product $\tau_j \dot{\theta}_j$ of the torque τ_j and joint angular velocity $\dot{\theta}_j$. Using the principle of virtual work for each joint we can compute

$$\tau_j = \mathbf{F} \cdot \frac{\partial \mathbf{v}}{\partial \dot{\theta}_j} + (\mathbf{P} - \mathbf{d}) \times \mathbf{F} \cdot \frac{\partial \vec{\omega}}{\partial \dot{\theta}_j}, \quad j = 1, \ldots, m. \tag{9.11}$$

We have arranged this equation to introduce the six-vector $\mathsf{F} = (\mathbf{F}, (\mathbf{P} - \mathbf{d}) \times \mathbf{F})^T$ which is the resultant force and moment at the reference point $\mathbf{d}$.

The equations (9.11) can be assembled into the matrix equation

$$\vec{\tau} = [J^T]\mathsf{F}, \tag{9.12}$$

where $[J]$ is the *Jacobian* defined above in (9.8). For a chain with six joints this equation can be solved for the output force-torque vector F,

$$\mathsf{F} = [J^T]^{-1}\vec{\tau}. \tag{9.13}$$

Thus, we see that the matrix that defines the mechanical advantage for this system is the inverse of the matrix of speed ratios. This is a more general version of the statement that to increase mechanical advantage we must decrease the speed ratio.

3. Simple Machines

Here we illustrate the basic issues of kinematic synthesis that we will discuss in more detail later. The kinematics equations, the relation between mechanical advantage and speed ratio and configuration space are easily developed for the lever, wedge and planar RR chain (R denotes a revolute joint).

3.1 The Lever

A lever is a solid bar that rotates about a fixed hinge O called its *fulcrum*. This serial chain and it has the kinematics equations

$$[D] = [Z(\theta)], \tag{9.14}$$

which define a pure rotation about the fulcrum. Its configuration parameter is simply the rotation angle θ.

Let the input from a motor or applied force result in a torque $T_{\text{in}} = F_{\in} a$ about the fulcrum, resulting in an output force at $\mathbf{B}$. See Figure 9.4. If the angular velocity of the lever is $\dot{\theta}$, then the velocity of $\mathbf{B}$ is $v_{\text{out}} = b\dot{\theta}$, and the principle of virtual work yields the relationship

$$(F_{\text{in}} a\dot{\theta} - F_{\text{out}} b\dot{\theta})\delta t = 0, \tag{9.15}$$

or,

$$\frac{F_{\text{out}}}{F_{\text{in}}} = \frac{a}{b}. \tag{9.16}$$

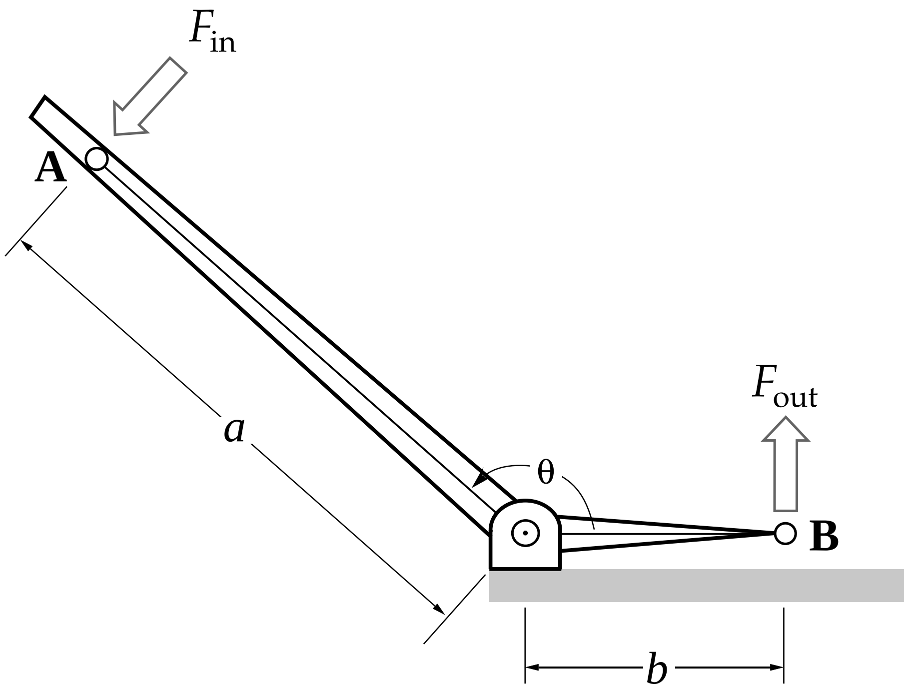


Figure 9.4 A lever is driven by a torque about its fulcrum to lift a load at **B**.

This is the well-known *law of the lever* which defines the mechanical advantage of the lever. The speed ratio of the output to input is given by

$$\frac{v_{\text{out}}}{v_{\text{in}}} = \frac{b\dot{\theta}}{a\dot{\theta}} = \frac{b}{a}, \tag{9.17}$$

which is inverse of the mechanical advantage. This is the simplest example of the relationship between the speed ratio and mechanical advantage.

3.2 The Wedge

A wedge is a right triangle with apex angle α that slides horizontally along a flat surface and lifts a load vertically by sliding it along its inclined face and against a vertical wall, Figure 9.5. This system consists of two parallel chains that support a load. One consists of the PP chain formed by the wedge itself, and the second P joint formed by load sliding against the wall (P denotes a prismatic, or sliding, joint). The kinematics equations of the system are

$$\begin{aligned} &[D] = [G_1][Z(0,x)][X(\alpha,0)][Z(0,a)][H_1] \\ \text{and}\quad &[D] = [G_2][Z(0,y)][H_2], \end{aligned} \tag{9.18}$$

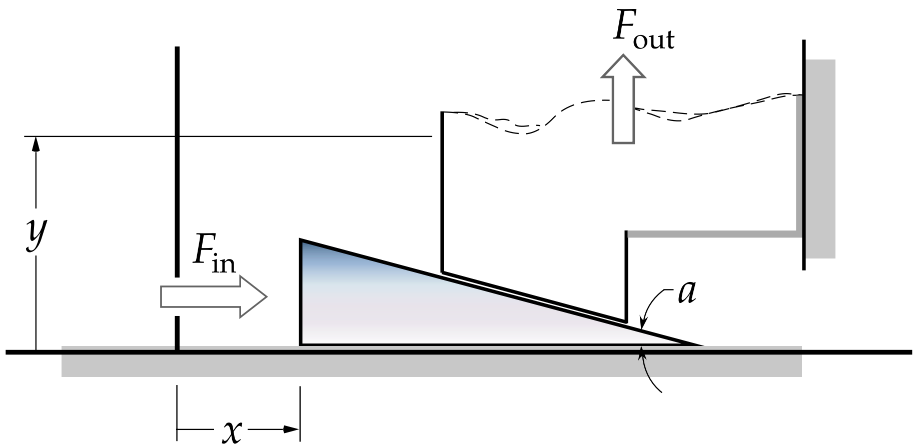


Figure 9.5 A wedge is driven in the x direction to slide a load along the y direction.

where $[G_i]$ and $[H_i]$ locate the base and moving frames for the two chains. The horizontal slide x and vertical slide y define the configuration of the system

The configuration parameters x and y must satisfy a constraint equation associated with the geometry of the two chains. The slide y is related to x by the slope $\tan\alpha$ of the wedge, that is

$$y = x\tan\alpha + k, \tag{9.19}$$

where k is a constant. This equation yields the speed ratio

$$\frac{\dot{y}}{\dot{x}} = \tan\alpha. \tag{9.20}$$

The principle of virtual work now yields

$$(F_{\text{in}}\dot{x} - F_{\text{out}}\dot{y})\delta t = 0, \tag{9.21}$$

and we obtain the mechanical advantage

$$\frac{F_{\text{out}}}{F_{\text{in}}} = \frac{1}{\tan\alpha}. \tag{9.22}$$

The fundamental issues shown here for a lever and wedge appear in the kinematic synthesis of general serial and parallel robot systems. The primary concerns are the workspace and mechanical advantage, or speed ratio, of the system.

3.3 The Planar RR Chain

Mechanical systems, including robots that interact with the world, often have components that make intermittent contact with other components. Thus,

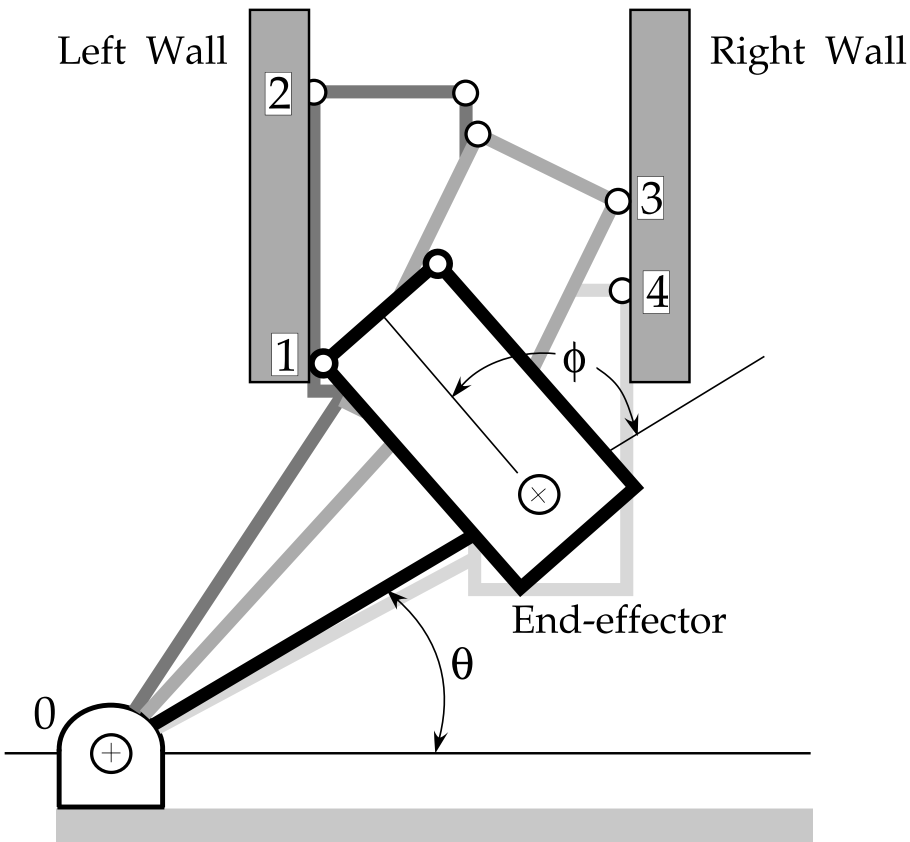


Figure 9.6 A planar RR robot moves its end-effector between two walls.

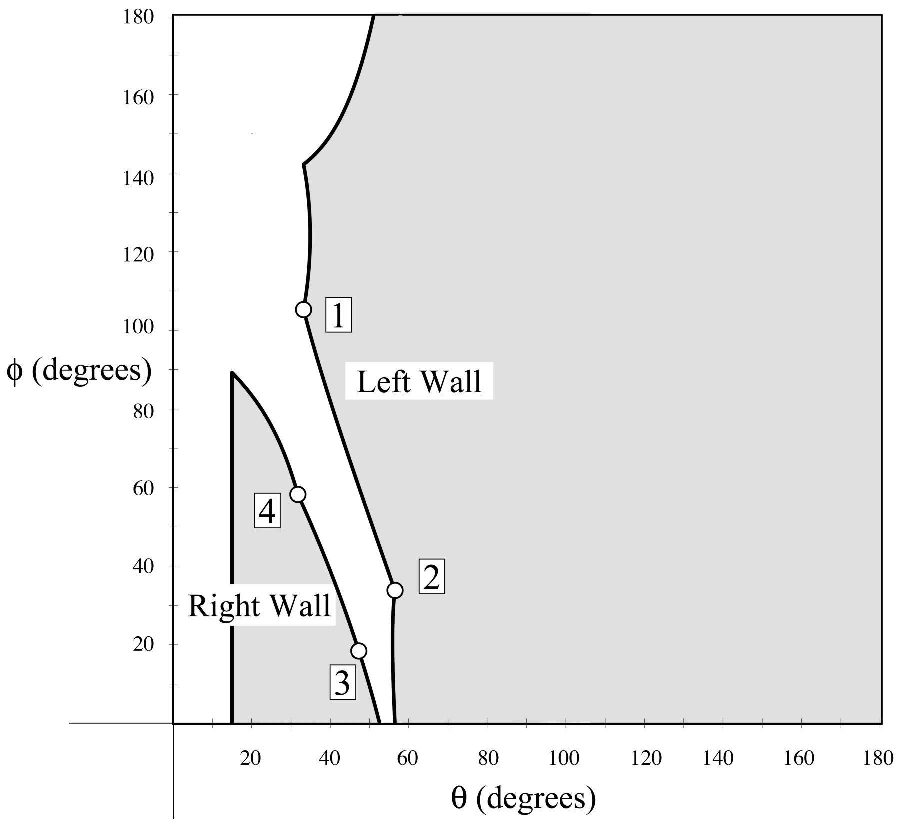


Figure 9.7 The joint angle values available to the robot are restricted by the presence of the two walls.

the system may be either and open chain or a closed chain depending on the contact configuration.

Perhaps the simplest system that illustrates this issue is an RR chain with its rectangular end-effector moving between two walls. See Figure 9.6. The position and orientation of the rectangle is defined by the joint variables θ and ϕ, therefore its configuration space is two-dimensional. The configurations excluded by the presence of the walls are said to define joint space obstacles. We show only the quadrant $0 \leq \theta \leq 180^\circ$ and $0 \leq \phi \leq 180^\circ$ in Figure 9.7 with the free space in white.

The boundary curves of a joint space obstacle are defined by the modes of contact of the end-effector and a wall. For obstacles and links that are polygons and circles, the robot-obstacle system forms a planar linkage that can be analyzed to determine this boundary (Ge and McCarthy 1989). For example, when a vertex of the rectangular end-effector moves along the left wall from position 1 to 2, as shown, the system forms a slider-crank linkage that is easily analyzed to determine $\phi(\theta)$. Similar calculations can be done for spatial polyhedra and spheres in contact in order to compute obstacle boundaries for spatial systems (Ge and McCarthy 1990).

4. Serial Robots

A serial chain robot is a sequence links and joints that begins at a base and ends with an gripper. See Figure 9.8. The position of the gripper is defined by the kinematics equations of the robot, which generally have the form

$$[D] = [Z(\theta_1, \rho_1)][X(\alpha_{12}, a_{12})][Z(\theta_2, \rho_2)]...[X(\alpha_{56}, a_{56})][Z(\theta_6, \rho_6)], \tag{9.23}$$

because the robot has six joints. The set of positions $[D]$ reachable by the robot is called its *workspace*.

The links and joints of a robot are usually configured to provide separate translation and orientation structures. Usually, the first three joints are used to position a reference point in space and the last three form the *wrist* which orients the gripper around this point (Vijaykumar *et al.*, 1987, Gupta 1987). This reference point is called the *wrist center*. The volume of space in which the wrist center can be placed is called the *reachable workspace* of the robot. The rotations available at each of these points is called the *dextrous workspace*.

The design of a robot is often based on the symmetry of its reachable workspace. From this point of view there are three basic shapes: rectangular, cylindrical and spherical (Craig, 1989). A rectangular workspace is provided by three mutually perpendicular sliding, or prismatic, joints which form a PPPS chain called a *Cartesian* robot—S denotes a spherical wrist which allows all rotations about its center point. A rotary base joint combined with a vertical and horizontal prismatic joints forms a CPS chain with a cylindrical workspace—

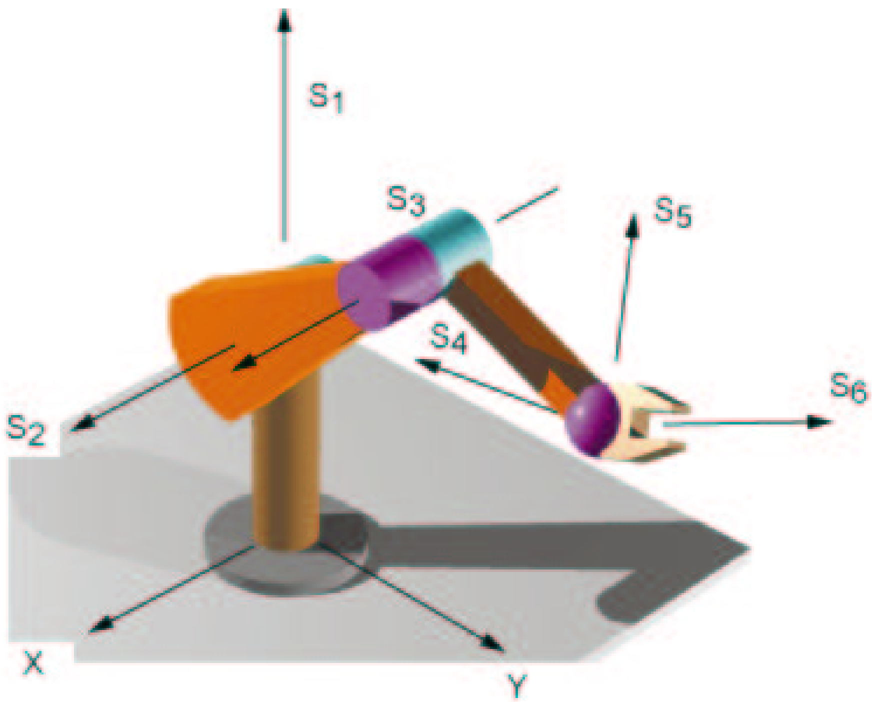


Figure 9.8 A serial robot is defined a set of joint axes S_i and the common normals A_{ij} between them.

C denotes a rotary and sliding joint with the same axis. The P-joint can be replaced by a revolute joint that acts as an elbow in order to provide the same radial movement. Finally, two rotary joints at right angles form a T-joint at the base of the robot that supports rotations about a vertical and horizontal axes. Radial movement is provided either by a P-joint, or by an R-joint configured as an elbow. The result is a TPS or TRS chain with a spherical workspace.

It is rare that the workspace is completely symmetrical because joint axes are often offset to avoid link collision and there are limits to joint travel which combine to distort the shape of the workspace.

4.1 Design Optimization

Another approach to robot design uses a direct specification of the workspace as a set of positions for the end-effector of a robotic system (Chen and Burdick 1995, Chedmail and Ramstein 1996, Chedmail 1998, Leger and Bares 1998) which we call the *taskspace*. A general serial robot arm has two design parameters, link offset and twist, for each of five links combined with four parameters each that locate the base of the robot and the workpiece in its gripper, for a total of 18 design variables. The link parameters are often specified so the chain has a spherical wrist and specific workspace shape. The design goal is usually to determine the workspace volume and locate the base and workpiece frames so that the workspace encloses the specified taskspace.

The taskspace is defined by a set of 4×4 transformations $[T_i], i = 1, \dots, k$. The problem is solved iteratively by selecting a design and using the associated kinematics equations $[D(\vec{\theta})]$ to compute the minimum relative displacements $[T_i D^{-1}(\vec{\theta}_i)]$. The invariants of each of these relative displacements are used to construct an objective function

$$f(\mathbf{r}) = \sum_{i=1}^{k} \|[T_i D^{-1}(\vec{\theta}_i)]\|. \tag{9.24}$$

Parameter optimization yields the design parameter vector $\mathbf{r}$ that minimizes this objective function.

Clearly, this optimization relies on the way the invariants are used to define of a distance measure between the positions reached by the gripper and the desired workspace. Park (1995), Martinez and Duffy (1995), Zefran *et al.*, (1996), Lin and Burdick (2000) and others have shown that there is no such distance metric that is coordinate frame invariant. This means that unless this objective function can be forced to zero so the workspace completely contains the taskspace, the resulting design will not be "geometric" in the sense that the same design is obtained for any choice of coordinates.

If the goal is a design that best approximates the taskspace, then we cannot allow both the location for the base frame of the robot and location of the workpiece in the gripper to be design variables. For example, in the above example, if the location of the base of the robot is given then we can find a coordinate invariant solution for the location of the workpiece in the gripper. On the other hand, if the position of the workpiece is specified the entire formulation may be inverted to allow a coordinate frame invariant solution for the location of the base frame (Bobrow and Park 1995).

4.2 4×4 Transforms as 4×4 Rotations

Etzel and McCarthy (1996) and Ahlers and McCarthy (2000) embed 4×4 homogeneous transformations in the set of 4×4 rotation matrices in order to provide the designer control over the error associated with coordinate frame variation within a specific volume of the world that contains the taskspace.

Consider the 4×4 screw displacement about the Z axis defined by

$$[Z(\theta, \rho)] = \begin{bmatrix} \cos\theta & -\sin\theta & 0 & 0 \\ \sin\theta & \cos\theta & 0 & 0 \\ 0 & 0 & 1 & \rho \\ 0 & 0 & 0 & 1 \end{bmatrix}. \tag{9.25}$$

This defines a rigid displacement in the three-dimensional hyperplane $x_4 = 1$ of four dimensional Euclidean space, E^4. The same displacement can be defined in parallel hyperplanes, $x_4 = R$, simply by dividing the translation component by R, that is

$$[Z(\theta, \rho)] = \begin{bmatrix} \cos\theta & -\sin\theta & 0 & 0 \\ \sin\theta & \cos\theta & 0 & 0 \\ 0 & 0 & 1 & \frac{\rho}{R} \\ 0 & 0 & 0 & 1 \end{bmatrix}. \tag{9.26}$$

This matrix can be viewed as derived from the 4×4 rotation:

$$[Z(\theta, \gamma)] = \begin{bmatrix} \cos\theta & -\sin\theta & 0 & 0 \\ \sin\theta & \cos\theta & 0 & 0 \\ 0 & 0 & \cos\gamma & \sin\gamma \\ 0 & 0 & -\sin\gamma & \cos\gamma \end{bmatrix}. \tag{9.27}$$

where $\tan\gamma = \rho/R$. This formalism views spatial translations as rotations of small angular values. The error ϵ associated with this approximation is less than $\sqrt{L/R}$ where L is the maximum dimension of the world space.

The result is an optimization strategy that computes both the base and workpiece frames yielding essentially the same design for all coordinate changes within the given volume of space. This approach also provides the opportunity to tailor the number of joints in the chain and set the values of internal link parameters to fit a desired task.

4.3 Speed Ratios

A six axis robot has a 6×6 Jacobian $[J]$ obtained from (9.8) that is an array of speed ratios relating the components of the velocity $\mathbf{v}$ of the wrist center and the angular velocity $\vec{\omega}$ of the gripper to each of the joint velocities. The principle of virtual work yields the relationship

$$\mathsf{F} = [J^T]^{-1}\vec{\tau}, \tag{9.28}$$

which defines the force-torque F exerted at the wrist center in terms the torque applied by each of the actuators. The link parameters of the robot can be selected to provide a Jacobian $[J]$ with specific properties.

The sum of the squares of the actuator torques of robot is often used as a measure of "effort" (Gosselin 1998, Albro *et al.*, 2000). From (9.28) we have

$$\vec{\tau}^T\vec{\tau} = \mathsf{F}^T[J][J^T]\mathsf{F}. \tag{9.29}$$

The matrix $[J][J^T]$ is square and positive definite. Therefore, it can be viewed as defining an hyper-ellipsoid in six-dimensional space (Shilov 1974). The lengths of the semi-diameters of this ellipsoid are the inverse of the absolute value of the eigenvalues of the Jacobian $[J]$. These eigenvalues may be viewed as "modal" speed ratios that define the amplification associated with each joint velocity. Their reciprocals are the associated "modal" mechanical advantage, so the shape of this ellipsoid illustrates the force amplification properties of the robot.

The ratio of the largest of these eigenvalues to the smallest, called the condition number, gives a measure of the anisotropy or "out-of-roundness" of the ellipsoid. A six-sphere has a condition number of one and is termed *isotropic*. When the gripper of a robot is in a position with an isotropic Jacobian there is no amplification of the speed ratios or mechanical advantage. This is considered to provide high-fidelity coupling between the input and output because errors are not amplified (Salisbury and Craig 1982, Angeles and Lopez-Cajun 1992). Thus, the condition number is used as a criterion in a robot design (Angeles and Chablat 2000).

In this case, it is assumed that the basic design of the robot provides a workspace that includes the taskspace. Parameter optimization finds the internal link parameters that yield the desired properties for the Jacobian. As in minimizing the distance to a desired workspace, optimization based on the Jacobian depends on a careful formulation to avoid coordinate dependency.

5. Parallel Robots

A robotic system in which two or more serial chain robots support an end-effector is called a *parallel robot*. Each of the serial chains must have six

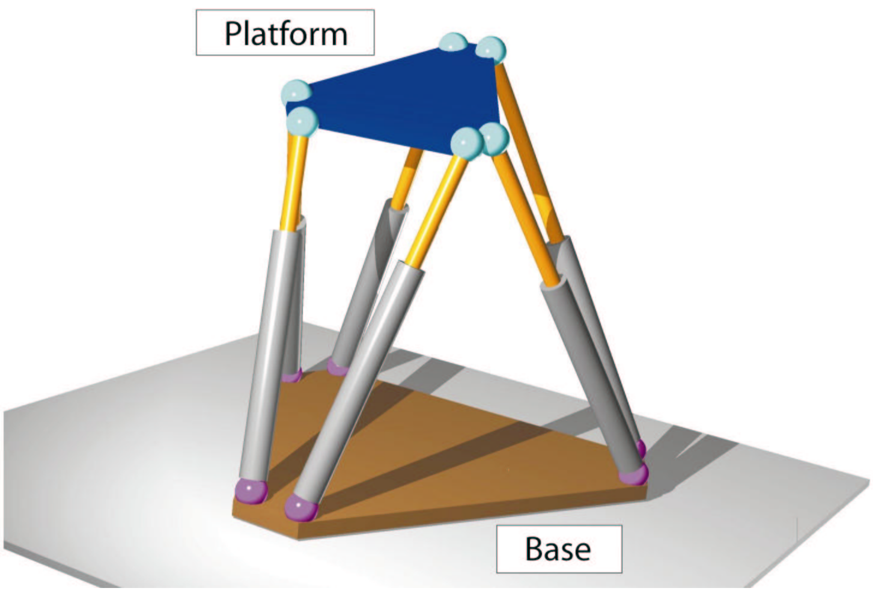


Figure 9.9 A parallel robot can have as many as six serial chains that connect a platform to the base frame.

degrees-of-freedom, however, in general only a total of six joints in the entire system are actuated. A good example is the Stewart platform formed from six TPS robots in which usually only the P-joint in each chain is actuated, Figure 9.9 (Fichter 1987, Tsai 1999, Merlet 1999).

The kinematics equations of the TPS legs are

$$[D] = [G_j][D(\vec{\theta}_j)][H_j], \quad j = 1, \ldots, 6, \tag{9.30}$$

where $[G_j]$ locates the base of the leg and $[H_j]$ defines the position of its attachment to the end-effector. The set of positions $[D]$ that simultaneously satisfy all of these equations is the workspace of the parallel robot.

Often the workspace of an individual chain of a parallel robot can be defined by geometric constraints. For example, a position $[D]$ is in the workspace of the jth supporting TPS chain if it satisfies the constraint equation

$$([D]\mathbf{q}_j - \mathbf{P}_j) \cdot ([D]\mathbf{q}_j - \mathbf{P}_j) = \rho_j^2. \tag{9.31}$$

This equation defines the distance between the base joint $\mathbf{P}_j$ and the point of attachment $\mathbf{Q}_j = [D]\mathbf{q}_j$ to the platform as the length ρ_j is controlled by the actuated prismatic joint. In this case the workspace is the set of positions $[D]$ that satisfy all six equations, one for each leg.

5.1 Workspace

The workspace of a parallel robot is the intersection of the workspaces of the individual supporting chains. However, it is not the intersection of the reachable and dextrous workspaces separately. These workspaces are intimately combined in parallel robots. The dextrous workspace is usually largest near the center of the reachable workspace and shrinks as the reference point moves toward the edge. A focus on the symmetry of movement allowed by supporting leg designs has been an important design tool resulting in many novel parallel designs (Hervé 1977, 1999). Simulation of the system is used to evaluate its workspace in terms of design parameters.

Another approach is to specify directly the positions and orientations that are to lie in the workspace and solve the algebraic equations that define the leg constraints to determine the design parameters (Murray *et al.*, 1997, Murray and Hanchak 2000). This yields parallel robots that are asymmetric but have a specified reachable and dextrous workspace.

5.2 Mechanical Advantage

The force amplification properties of a parallel robot are obtained by considering the Jacobians of the individual supporting chains. Let the linear and angular velocity of the platform be defined by the six-vector $\mathsf{V} = (\mathbf{v}, \vec{\omega})^T$, then from the kinematics equations of each of the support legs we have

$$\mathsf{V} = [J_1]\dot{\vec{\rho}}_1 = [J_2]\dot{\vec{\rho}}_2 = \cdots = [J_6]\dot{\vec{\rho}}_6. \tag{9.32}$$

Here we assume that the platform is supported by six chains, but it can be less. This occurs when the fingers of a mechanical hand grasp an object (Mason and Salisbury 1985).

The force on the platform applied by each chain is obtained from the principle of virtual work as

$$\mathsf{F}_j = [J_j^T]^{-1}\vec{\tau}_j, \quad j = 1, \ldots, 6. \tag{9.33}$$

There are only six actuated joints in the system so we assemble the associated joint torques into the vector $\vec{\tau} = (\tau_1, \ldots, \tau_6)^T$. If F_i is the force-torque obtained from (9.33) for $\tau_i j = 1$, then, the resultant force-torque W applied to the platform is

$$\mathsf{W} = [\mathsf{F}_1, \mathsf{F}_2, \cdots, \mathsf{F}_6]\vec{\tau}, \tag{9.34}$$

or

$$\mathsf{W} = [\Gamma]\vec{\tau}. \tag{9.35}$$

The elements of the coefficient matrix $[\Gamma]$ define the mechanical advantage for each of the actuated joints. In the case of a Stewart platform the columns of this matrix are the Plücker coordinates of the lines along each leg (Merlet 1989).

The principle of virtual work yields the velocity of the platform in terms of the joints rates $\dot{\vec{\rho}}$ as

$$[\Gamma]^T \mathsf{V} = \dot{\vec{\rho}}. \tag{9.36}$$

Thus, the inverse of $[\Gamma]$ defines the speed ratios between the actuated joints and the end-effector. The same equation can be obtained by computing the derivative of the geometric constraint equations (9.31), and $[\Gamma]$ is the *Jacobian* of the parallel robot system (Kumar 1992, Tsai 1999).

The Jacobian $[\Gamma]$ is used in parameter optimization algorithms to design parallel robots (Gosselin and Angeles 1988) with isotropic mechanical advantage. The square root of the determinant $|[\Gamma][\Gamma]^T|$ measures the six-dimensional volume spanned by the column vectors F_j. The distribution of the percentage of this volume compared to its maximum within the workspace is also used as a measure of the overall performance (Lee *et al.*, 1996, 1998). A similar performance measure normalizes this Jacobian by the maximum joint torques available and the maximum component of force and torque desired, and then seeks an isotropic design (Salcudean and Stocco 2000).

6. Linkage Design Theory

An assembly links and joints, which is our general definition of a machine, can be called a *linkage*. However, this term is generally restricted to machine elements that have much less than the six-degrees-of-freedom typical of a robotic system. Often they are one degree-of-freedom single-input-single-output devices such as the four-bar linkage.

The kinematic synthesis theory presented above for robots is actually a generalization of an approach originally developed for linkages. Beginning with a set of task positions, Burmester (1886) obtained an exact geometric solution to the constraint equations of a planar RR chain which he then assembled into a four-bar linkage. This has grown into a rich theory for the exact solution of the geometric constraint equations for RR and RP planar chains, RR spherical chains, and TS, CC and RR spatial chains. See Chen and Roth (1967) and Suh and Radcliffe (1978).

6.1 The Spatial RR Chain

The principles of kinematic synthesis of linkages can be seen in the direct solution of the constraint equations of an RR chain. Figure 9.10 shows a spatial RR chain, which can be considered the simplest robot. It has 10 design

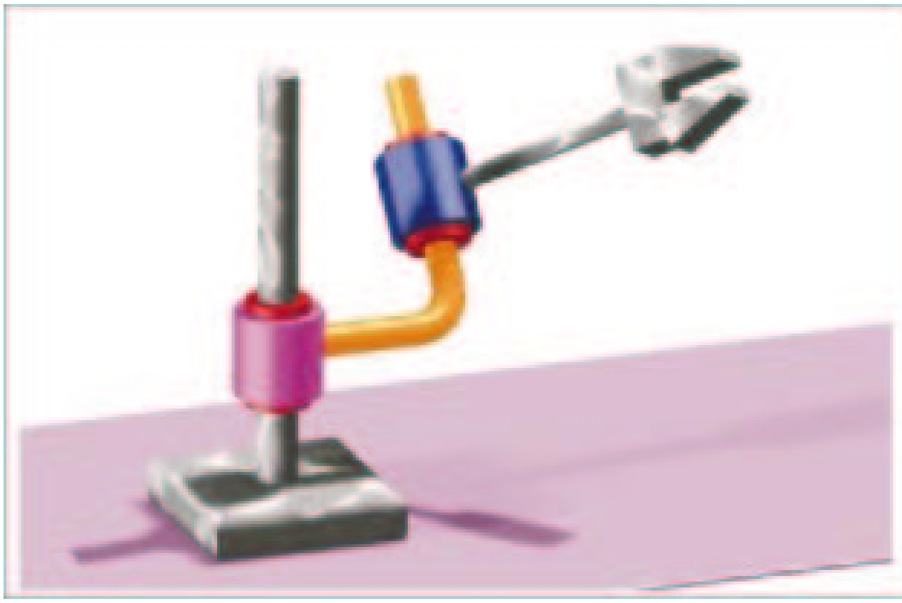


Figure 9.10 A spatial RR chain.

parameters, four for the fixed axis G and the moving axis W and the offset ρ and twist angle α.

The kinematics equations of the spatial RR chain are given by

$$[D] = [G][Z(\theta, 0)][X(\alpha, \rho)][Z(\phi, 0)][H], \tag{9.37}$$

which define its workspace. Choose a reference position $[D_1]$ and right translate the entire workspace to obtain

$$[D_{1k}] = [D][D_1]^{-1} = \\ ([G][Z(\theta, 0)][X(\alpha, \rho)][Z(\phi, 0)][H])([G][Z(\theta_1, 0)][X(\alpha, \rho)][Z(\phi_1, 0)][H])^{-1}. \tag{9.38}$$

This equation can be simplified to obtain

$$[D_{1k}] = [T(\Delta\theta, \mathsf{G})][T(\Delta\phi, \mathsf{W})], \tag{9.39}$$

where

$$[T(\Delta\theta, \mathsf{G})] = [G][Z(\Delta\theta, 0)][G]^{-1}, \\ [T(\Delta\phi, \mathsf{W})] = ([G][[Z(\theta_1, 0)][X(\alpha, \rho)]])[Z(\Delta\phi, 0)]([G][[Z(\theta_1, 0)][X(\alpha, \rho)]])^{-1}. \tag{9.40}$$

This defines the workspace of the RR chain as the composition of a rotation about the moving axis W in its reference position followed by a rotation about the fixed axis G, ($\Delta\theta = \theta - \theta_1$ and $\Delta\phi = \phi - \phi_1$ measure the rotation from the reference position).

To design a spatial RR chain we determine G, W, α and ρ such that (9.39) includes the desired taskspace. The general case requires the solution of 10 algebraic equations in 10 unknowns. We can simplify the presentation significantly by restricting attention to planar RR chains for which the axes G and W are parallel, which means the twist angle $\alpha = 0$.

6.2 The Planar RR Chain

If G and W are parallel, then they can be located in the plane perpendicular to their common direction by the coordinates $\mathbf{P} = (x, y)^T$ and $\mathbf{Q} = (\lambda, \mu)^T$. The kinematics equations of the chain become

$$[D_{1k}] = [T(\Delta\theta, \mathbf{P})][T(\Delta\phi, \mathbf{Q})], \tag{9.41}$$

which is the composition of rotations parallel to this plane about the points $\mathbf{Q}$ then $\mathbf{P}$.

The workspace of this chain can also be defined as the set of displacements $[D_{1k}]$ that satisfy the algebraic equation

$$([D_{1k}]\mathbf{Q} - \mathbf{P})^T([D_{1k}]\mathbf{Q} - \mathbf{P}) = \rho^2. \tag{9.42}$$

This is the geometric constraint that the displaced moving pivot $[D_{1k}]\mathbf{Q}$ must remain at a constant distance ρ from the fixed pivot $\mathbf{P}$.

We use Equation (9.42) to directly determine the five design parameters $\mathbf{r} = (x, y, \lambda, \mu, \rho)$. This is done by evaluating this equation at five task positions $[T_i], i = 1, \ldots, 5$, so we have

$$([D_{1i}]\mathbf{Q} - \mathbf{P})^T([D_{1i}]\mathbf{Q} - \mathbf{P}) = \rho^2, i = 1, \ldots, 5. \tag{9.43}$$

The result is a set of five equations in the five unknown design parameters. The distance ρ is easily eliminated by subtracting the first equation from the remaining four. This also cancels the squared terms u^2, v^2, λ^2 and μ^2. The resulting four equations are bilinear in the variables (x, y) and (λ, μ), and can be written as

$$\begin{bmatrix} A_2(x,y) & B_2(x,y) \\ A_3(x,y) & B_3(x,y) \\ A_4(x,y) & B_4(x,y) \\ A_5(x,y) & B_5(x,y) \end{bmatrix} \begin{Bmatrix} \lambda \\ \mu \end{Bmatrix} = \begin{Bmatrix} C_2(x,y) \\ C_3(x,y) \\ C_4(x,y) \\ C_5(x,y) \end{Bmatrix}. \tag{9.44}$$

In order for this equation to have a solution, the four 3×3 minors of the augmented coefficient matrix must all be identically zero. This yields four cubic equations in the coordinates x and y of the fixed pivot. These cubic equations can be further manipulated to yield a quartic polynomial in x (McCarthy 2000). Each real root of this polynomial defines a planar RR chain that reaches the five specified task positions.

6.3 Design Software

Kaufman (1978) was the first to transform this mathematical result into an interactive graphics program for linkage design called KINSYN. He used a modified game controller to provide the designer the ability to input a set of task positions. An important feature of this software was the decision to only allow the designer to specify four, not five, positions. Rather than obtain a finite number of RR chains, his software determined the cubic curve of solutions known as the *center-point curve*. This curve is obtained by setting the minor obtained from the first three equations in (9.44) to zero. Kaufman's software would ask the designer to select two points on this curve in order to define two RR chains that it assembled into the one degree-of-freedom 4R linkage, or *four-bar* linkage. Analysis routines evaluate the performance of the design and provide a simulation of its movement.

Erdman and Gustafson (1977) introduced LINCAGES which, like KINSYN, focused on four task positions for the design of a 4R planar linkage. This software introduced a "guide map" which displayed the characteristics of every four-bar linkage that could be constructed from points on the center-point curve. This software was extended by Chase *et al.*, (1981) to design an additional 3R chain to form a six-bar linkage.

Waldron and Song (1981) introduced the design software RECSYN which again sought 4R closed chains that guide a body through three or four task positions. Their innovation was an analytical formulation that ensured the linkage would not "jam" as it moved between the design positions. In linkage design a jam is equivalent to hitting a singular configuration in a robot which occurs when the determinant of the Jacobian becomes zero. An important feature of this software was the growing reliance on graphical communication of geometric information regarding the characteristics of the available set of designs.

Larochelle *et al.*, (1993) introduced the "Sphinx" software for the design of spherical 4R linkages, which can be viewed as planar 4R linkages that are bent onto the surface of a sphere. A spherical RR chain is obtained when the link offset is $\rho = 0$, Figure 9.11. The fixed and moving axes of spherical RR chains are defined by the unit vectors $\mathbf{G} = (x, y, z)^T$ and $\mathbf{W} = (\lambda, \mu, \nu)^T$. The workspace of relative rotations $[A_{1k}]$ reachable by this chain is defined by the algebraic equation

$$\mathbf{G}^T[A_{1k}]\mathbf{W} = \cos\alpha. \qquad (9.45)$$

This equation is evaluated at five specified task orientations to obtain equations that are essentially identical to (9.44) and solved in the same way (McCarthy 2000).

Following the pattern established by KINSYN and LINCAGES, Sphinx asks the designer to specify four task orientations, and then generates the *center-axis cone*, which is the spherical equivalent of the center-point curve.

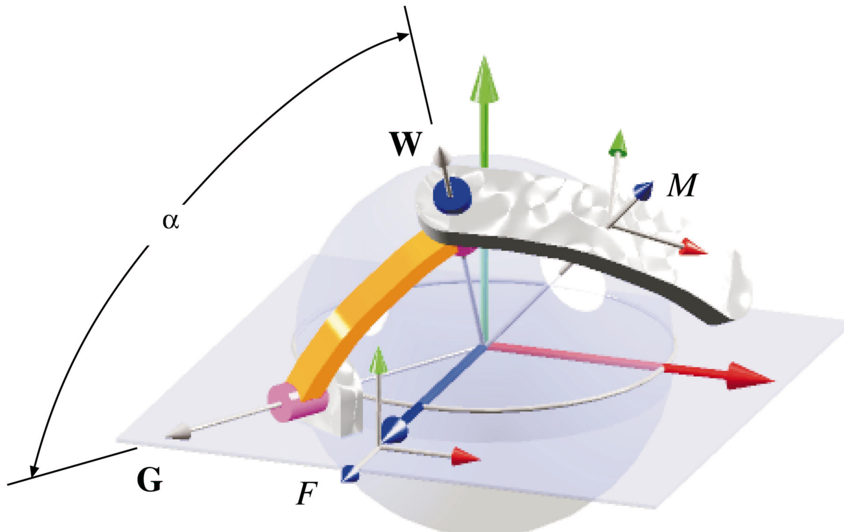


Figure 9.11 A spherical RR chain.

The software also computes a "type map" which classifies via color coding the movement of every 4R chain that can be constructed from pairs of axes on this cone. The typemap also included filters that eliminated designs with known defects. A later version of this software called SphinxPC also included planar 4R linkage design (Ruth and McCarthy 1997). The display and typemap windows of SphinxPC are shown in Figure 9.12.

The three dimensional nature of the interaction needed for spherical 4R linkage design presents severe visualization challenges. The designer finds that specifying a task as a set of spatial orientations is an unfamiliar experience. Furlong *et al.*, (1998) used immersive virtual reality in their software IRIS to enhance this interaction.

Larochelle (1998) introduced the SPADES software which provided interactive design for a truly spatial linkage system, the 4C linkage, for the first time. A CC chain is the generalized robot link that allows both rotation about and sliding along each axis. Let $\mathsf{G} = (\mathbf{G}, \mathbf{P}\times\mathbf{G})^T$ and $\mathsf{W} = (\mathbf{G}, \mathbf{Q}\times\mathbf{W})^T$ be the Plücker coordinates locating the fixed and moving axes in space. The workspace of this chain can be defined as the displacements $[D_{1k}]$ that satisfy the pair of geometric constraints,

$$\mathbf{G}^T[A_{1k}]\mathbf{W} = \cos\alpha,$$
$$(\mathbf{P}\times\mathbf{G})^T[A_{1k}]\mathbf{W} + \mathbf{G}^T[D_{1k}](\mathbf{Q}\times\mathbf{W}) = -\rho\sin\alpha. \quad (9.46)$$

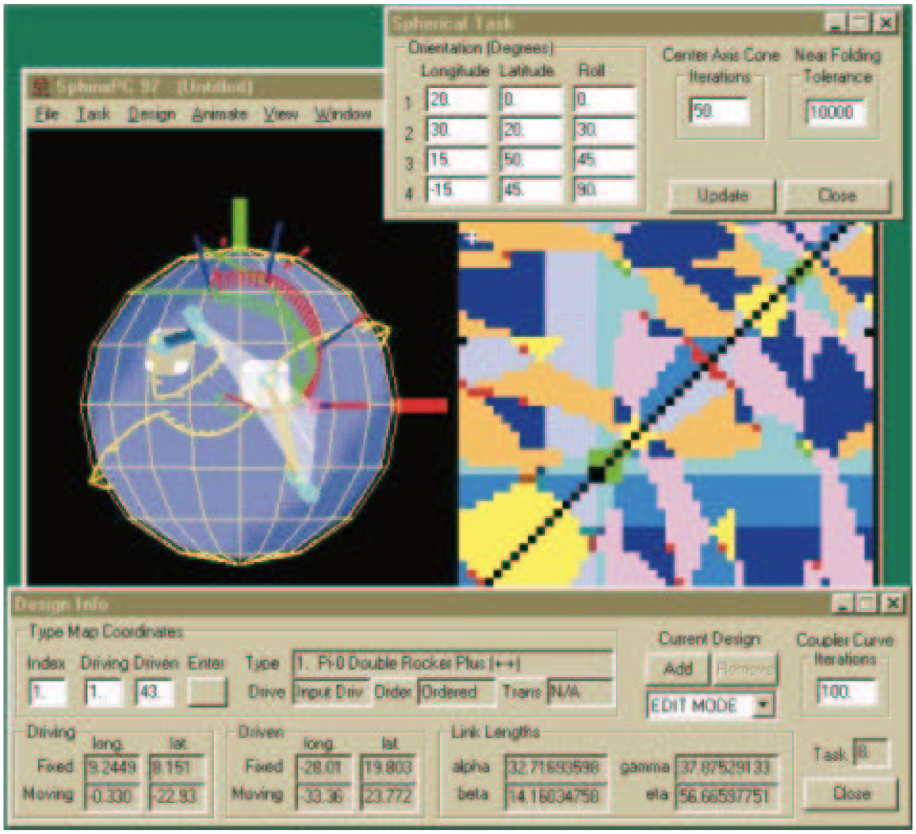


Figure 9.12 The desktop of the spherical linkage design software SphinxPC.

These equations constrain the link parameters α and ρ to be constant for every position of the moving frame. There are 10 design parameters consisting of four parameters for each of the two axes, the link offset ρ and twist angle α.

By evaluating the two constraint equations at each of five task positions, we obtain 10 equations in 10 unknowns. Five of these are identical to those used for the synthesis of spherical RR chains and can be solved to determine the directions $\mathbf{G}$ and $\mathbf{W}$. The remaining five equations are linear in the components of $\mathbf{P} \times \mathbf{G}$ and $\mathbf{Q} \times \mathbf{W}$ and are easily solved.

SPADES generates the *center-axis congruence* which is the set of spatial CC chains that reach four spatial task positions. It then assembles pairs of these chains into two degree-of-freedom 4C linkages. This software demonstrates the significant visualization challenge that exists in the specification of spatial task frames and evaluation of candidate designs.

These linkage design algorithms ensure that the workspace of each linkage includes the specified taskspace. However, in each case the designer is

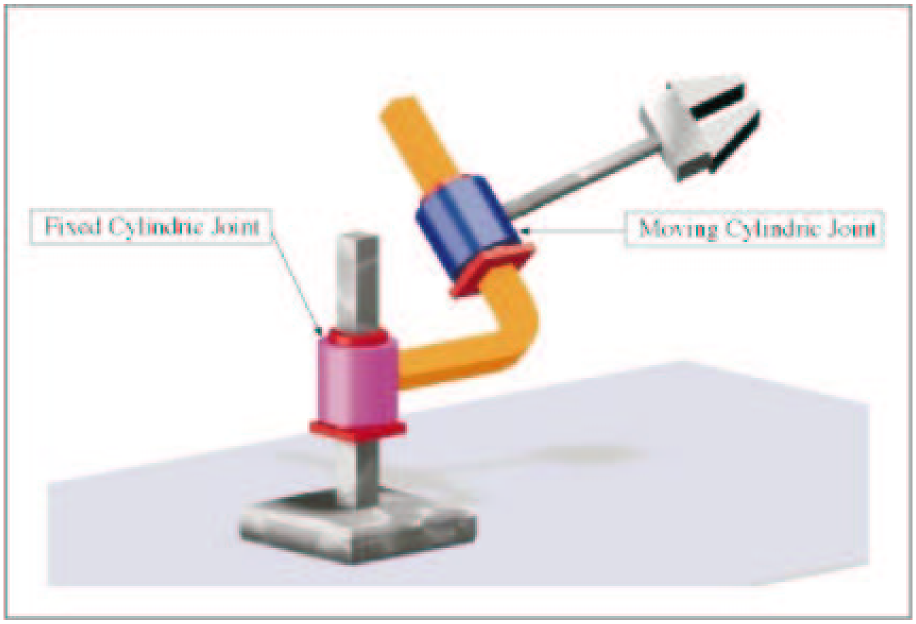


Figure 9.13 The CC open chain robot.

expected to search performance measures and examine simulations for many candidate designs in order to verify the quality of movement between the individual task positions.

6.4 The TS Chain

Another example of the challenge inherent in the kinematic synthesis of spatial chains is found in the design of the TS chain, Figure 9.14. This chain has the workspace defined by the algebraic equation,

$$([D_{1k}]\mathbf{Q} - \mathbf{P})^T([D_{1k}]\mathbf{Q} - \mathbf{P}) = \rho^2. \tag{9.47}$$

There are seven design parameters, the six coordinates of $\mathbf{P} = (x, y, z)^T$ and $\mathbf{Q} = (\lambda, \mu, \nu)^T$ that define the centers of the T and S joints, respectively, and the length ρ. Therefore, we can evaluate this equation at seven spatial positions. The result, however, is that we can compute as many as 20 TS chains (Innocenti 1995). If these are assembled into the single-degree-of-freedom 5TS linkage, Figure 9.15, then over 15,000 designs must be analyzed which is an extreme computational burden (Liao and McCarthy 1999). Prototype software shows that it is remarkably difficult for a designer to specify seven spatial positions and obtain a useful design.

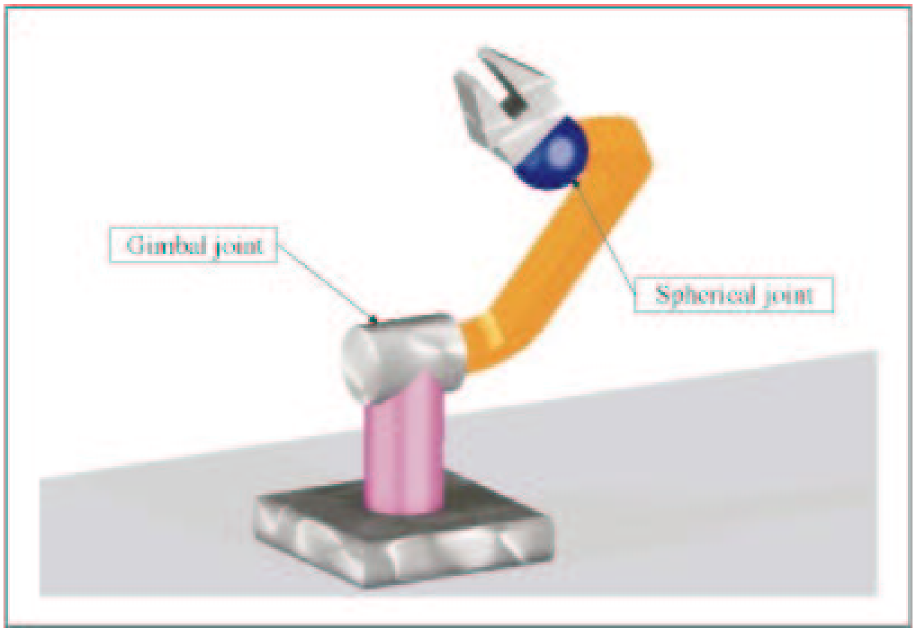


Figure 9.14 The TS open chain robot.

Recent research has focussed on developing higher resolution tasks and providing methods for fitting a low dimensional workspace to a general taskspace.

6.5 Task Specification

The workspace of a linkage is a subset of the group of all spatial displacements, denoted SE(3), reachable by a workpiece, or end-effector, of the system. The taskspace is a discrete set of points in SE(3) that located key-frames, or precision positions, that the linkage must reach. Research in motion interpolation has provided techniques to generate smooth trajectories through a set of key-frames using Bezier-style methods.

Bezier interpolation is used in computer drawing systems to generate curves through specified points (Farin 1997). Shoemake (1985) shows that this technique can be used to interpolate rotation key frames specified by quaternion coordinates (Hamilton 1860). Ge and Ravani (1994) generalized Shoemake's results using dual quaternions to interpolate spatial displacements, and Ge and Kang (1995) refined this approach to ensure smooth transitions at each key frame. The result is an efficient method to specify task trajectories in the manifold SE(3) (Figure 9.16); see McCarthy (1990) for a discussion of quaternions and dual quaternions as Clifford Algebras which represent the groups of rotations SO(3) and spatial displacements SE(3), respectively.

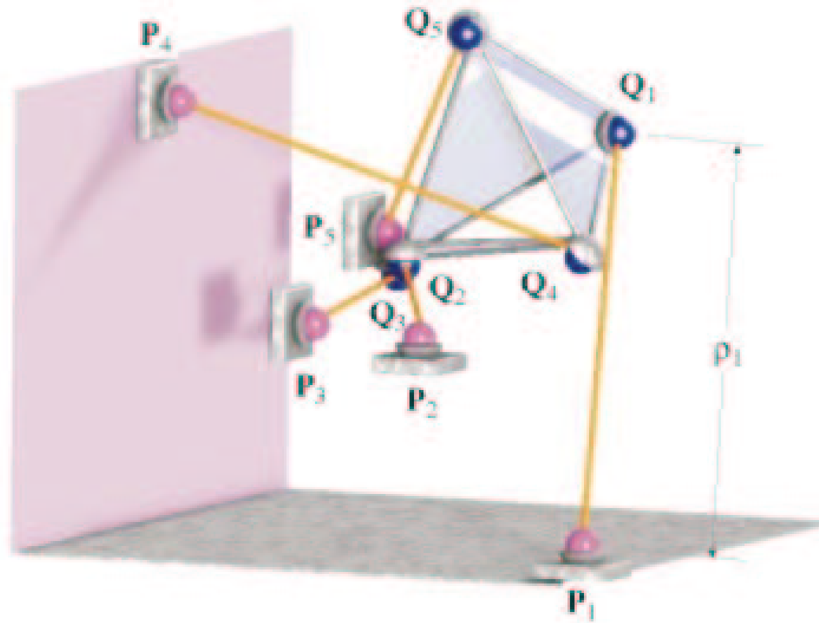


Figure 9.15 The 5-TS platform linkage.

Ahlers and McCarthy (2000) show that the Clifford Algebra of 4×4 rotations provides an efficient method for specifying spatial trajectories using double quaternion equations. In McCarthy and Ahlers (2000) they design spatial CC chains for discrete subsets of a double quaternion trajectory, and select the design that best fits the overall trajectory. This procedure combines an analytical solution for the fixed and moving axes with an optimization procedure that bounds the error associated with coordinate transformations in both the fixed and moving frames. This strategy was applied to the design of spatial RR robots in Perez and McCarthy (2000), Figure 9.17.

7. Mechanism design

So far we have restricted our attention to isolated mechanisms consisting of lower-pair joints permanently attached together. In general, rigid-body systems consist of parts whose contacts are more complex and change with time. These systems include linkages moving in workspaces with obstacles, mechanisms consisting of cams, gears, geneva wheels and other higher pairs, and general assemblies of rigid bodies (Tsai 1993, Norton 1993). The key issue for kinematic analysis and synthesis of these systems is *contact analysis*.

Contact analysis determines the positions and orientations at which the parts of a system touch and the ways that the touching parts interact. The interactions

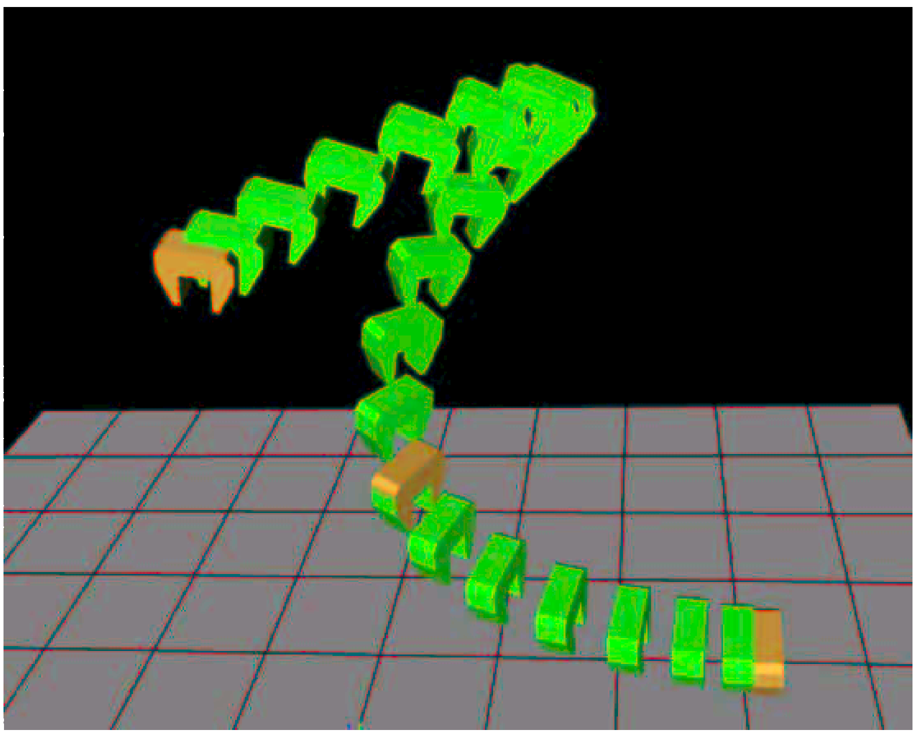


Figure 9.16 Spatial tasks can be specified using Bezier motion interpolation.

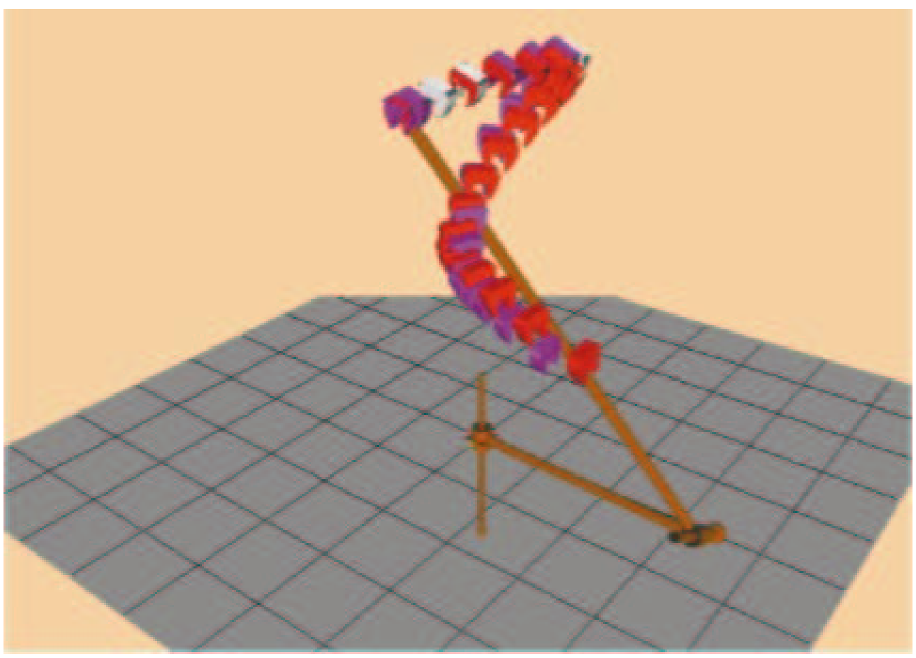


Figure 9.17 The workspace of a spatial RR robot is a manifold can be fit to a desired trajectory by adjusting its design parameters.

consist of constraints on the part motions that prevent them from overlapping. The constraints are expressed as algebraic equations that relate the part coordinates. For example, a ball rolling down a 45 degree slope obeys the constraint $x - y = r$, where x and y are the coordinates of the ball's center point and r is the ball radius. The constraints are a function of the shapes of the touching part features (vertices, edges, and faces), hence they change when one pair of features breaks contact and another makes contact.

To illustrate contact analysis and its role in design, consider the ratchet mechanism in Figure 9.18. The mechanism has four moving parts and a fixed frame. The driver, link, and ratchet are attached to the frame by revolute joints. The pawl is attached to the link by a revolute joint and is attached to a spring (not shown) that applies a counterclockwise torque around the joint. A motor rotates the driver with constant angular velocity, causing the link pin to move left and right. This causes the link to oscillate around its rotation point, which moves the pawl left and right. The leftward motion pushes a ratchet tooth, which rotates the ratchet counterclockwise. The rightward motion frees the pawl tip from the tooth, which allows the spring to rotate the pawl to engage the next tooth.

Kinematic analysis and synthesis of the ratchet mechanism requires contact analysis. We need to determine if the link oscillates far enough, if the pawl pushes the ratchet teeth far enough, if the system can jam, and so on. In the driver/link pair, the link pin interacts with the inner and outer driver profiles, creating a positive drive with small play. The ratchet/pawl pair is much harder to design because the part shapes and contact sequences are complex and because the pawl can translate horizontally, translate vertically, and rotate, whereas the other parts just rotate. We also need to validate intended interactions among all the parts, such as the indirect relation between the driver and the ratchet, and must rule out interference, such as the pawl hitting the frame.

Contact analysis is best understood in the framework of *configuration space*. Configuration space is a geometric representation of rigid body interaction that is widely used in robot motion planning (Latombe 1991). The configuration space of a mechanical system describes all possible part interactions. It encodes quantitative information, such as part motion paths, and qualitative information, such as system failure modes. It provides a framework within which diverse design tasks can be performed, as we explain next.

7.1 Configuration space

We study contact analysis within the configuration space representation of rigid-body interaction. The configuration space of a system of rigid parts is a parameter space whose points specify the spatial configurations (positions and orientations) of the parts. The parameters usually represent part translations

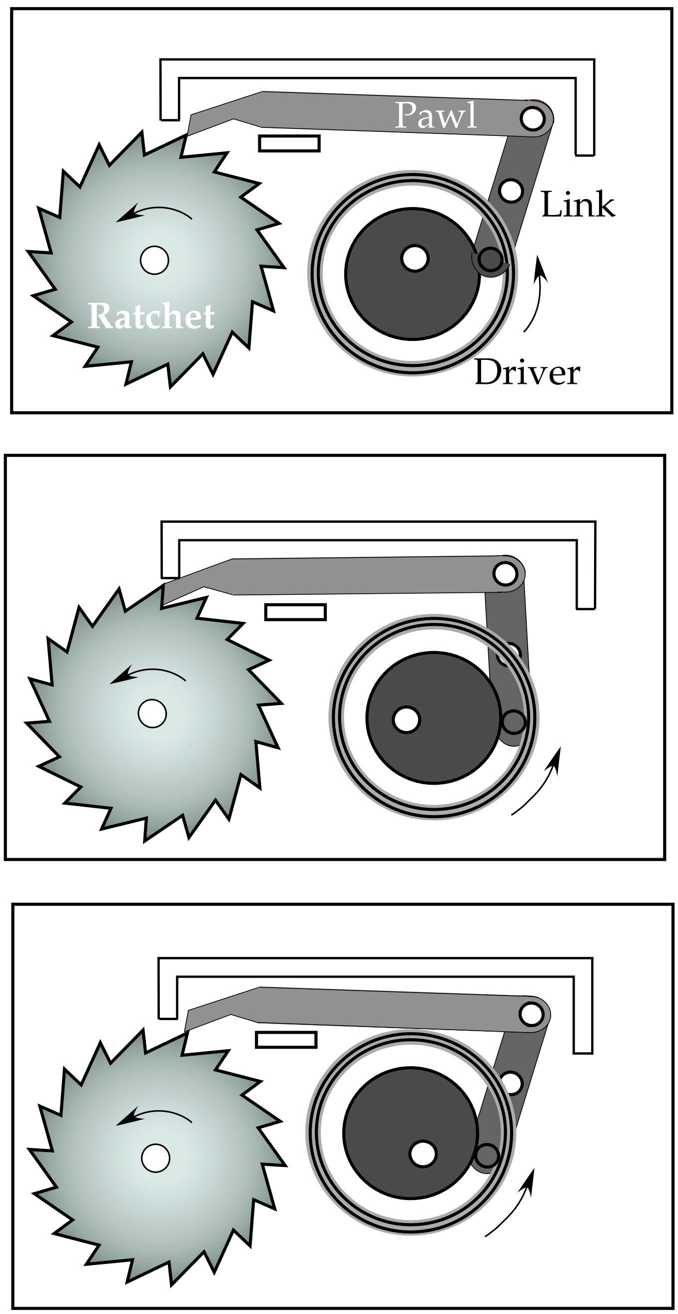


Figure 9.18 Ratchet mechanism: (a) pawl advancing ratchet; (b) pawl fully advanced; (c) pawl retracting. White circles indicate revolute joints.

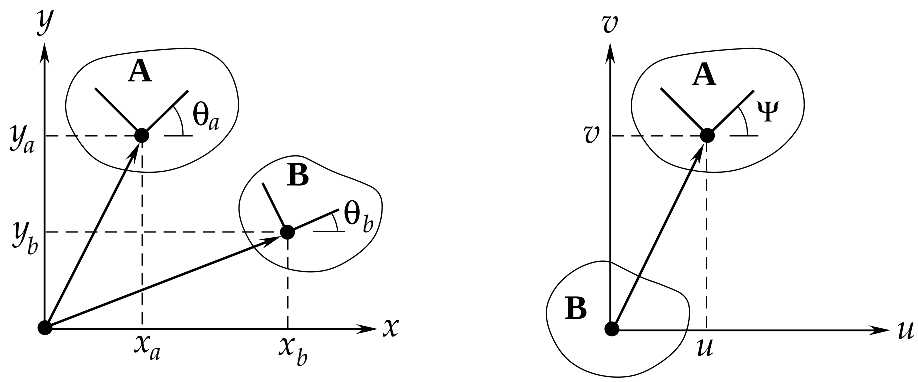


Figure 9.19 Pairwise configuration parameters coordinates: (a) absolute coordinates, (b) relative coordinates.

and rotations, but can be arbitrary generalized coordinates. The configuration space dimension equals the number of independent part motions, called degrees of freedom of the system.

We begin by studying a mechanical system consisting of a pair of planar parts. We attach reference frames to the parts and define the configuration of a part to be the position and orientation of its reference frame with respect to a fixed global frame. Figure 9.19a shows planar parts A and B, their reference frames, and their configurations (x_a, y_a, θ_a) and (x_b, y_b, θ_b). The configuration space of the pair is the Cartesian product, $(x_a, y_a, \theta_a, x_b, y_b, \theta_b)$, of the part configurations. The configuration space coordinates represent three degrees of freedom of each part.

An alternative representation is to describe the relative position and orientation of part A with respect to B, which is fixed and whose reference frame is at the origin of the axes, as illustrated in Figure 9.19b. In this case, three relative parameters (u, v, ψ) uniquely describe the relative position of A with respect to B. The relation between the absolute and relative coordinate systems is:

$$\begin{aligned} u &= (x_a - x_b)\cos\theta_b + (y_a - y_b)\sin\theta_b \\ v &= (y_a - y_b)\cos\theta_b - (x_a - x_b)\sin\theta_b \\ \psi &= \theta_a - \theta_b. \end{aligned} \tag{9.48}$$

The configuration space dimension for planar pairs is six for absolute coordinates, and three for relative coordinates. For spatial pairs, it is twelve for absolute coordinates and six for relative coordinates. Other useful parameterizations include quaternions for spatial rotations (Bottema and Roth 1979) and Clifford Algebras parameterization of planar and spatial displacements that yield algebraic surfaces that represent geometric constraints (McCarthy 1990, Collins and McCarthy 1998, Ge *et al.*, 1998).

Contact analysis is simplified by considering only the varying configuration parameters. In mechanisms, parts frequently have less than six degrees of freedom. The fixed degrees of freedom correspond to constant configuration parameter values. Mathematically, this corresponds to projecting the higher-dimensional configuration space into a lower dimensional space with identical properties. For example, in the ratchet mechanism, the ratchet and the driver are mounted on fixed axes, so only their orientation varies. A two-dimensional configuration space, showing the dependence between the two orientation parameters fully describes the contacts between them.

Configuration space partitions into three disjoint sets that characterize part interaction: blocked space where the parts overlap, free space where they do not touch, and contact space where they touch. Contact space is the common boundary of free and blocked spaces. Free and blocked space are open sets whose dimension is identical to that of the configuration space, whereas the dimension of contact space is one lower. Intuitively, free and blocked space are open because disjoint or overlapping parts remain so under all small motions, whereas contact space is closed because touching parts separate or overlap under some small motions.

We illustrate these concepts with a simple example: a block that moves in a fixed frame. In Figure 9.20a, the frame is fixed at the global origin, so we can ignore its coordinates from the configuration space and consider only the block coordinates (u, v, ψ) relative to the block origin. Assume first that the block translates in the displayed orientation without rotating. This yields a two-dimensional configuration space whose parameters are the horizontal and vertical parameters u and v (Figure 9.20b). The gray region is blocked space, the white region is free space, and the black lines are contact space. The dot in free space marks the displayed position of the block. Free space divides into a central rectangle where the block is inside the frame, an outer region where it is outside, and a narrow connecting rectangle where it is partly inside. The contact constraints (lines in this case) bounding these regions represent contacts between the vertices and edges of the block and the frame. Typical configurations on each region are shown in Figure 9.20b on the left. A collision-free motion of the block corresponds to a continuous path in free and contact space. Changing the orientation of the block yields configuration spaces with different topologies, as shown in Figures 9.20b, c, d, and e). Note that the free space consists now of two disconnected inside and outside regions because the block does not fit through the frame opening and thus cannot exit the inner region as before.

Consider now the same example, but with the block orientation varying. The configuration space becomes three dimensional with rotation coordinate ψ varying from $-\pi$ to π. Contact space is now two-dimensional, and is formed by contact patches, as shown in Figure 9.21. Typical configurations for three

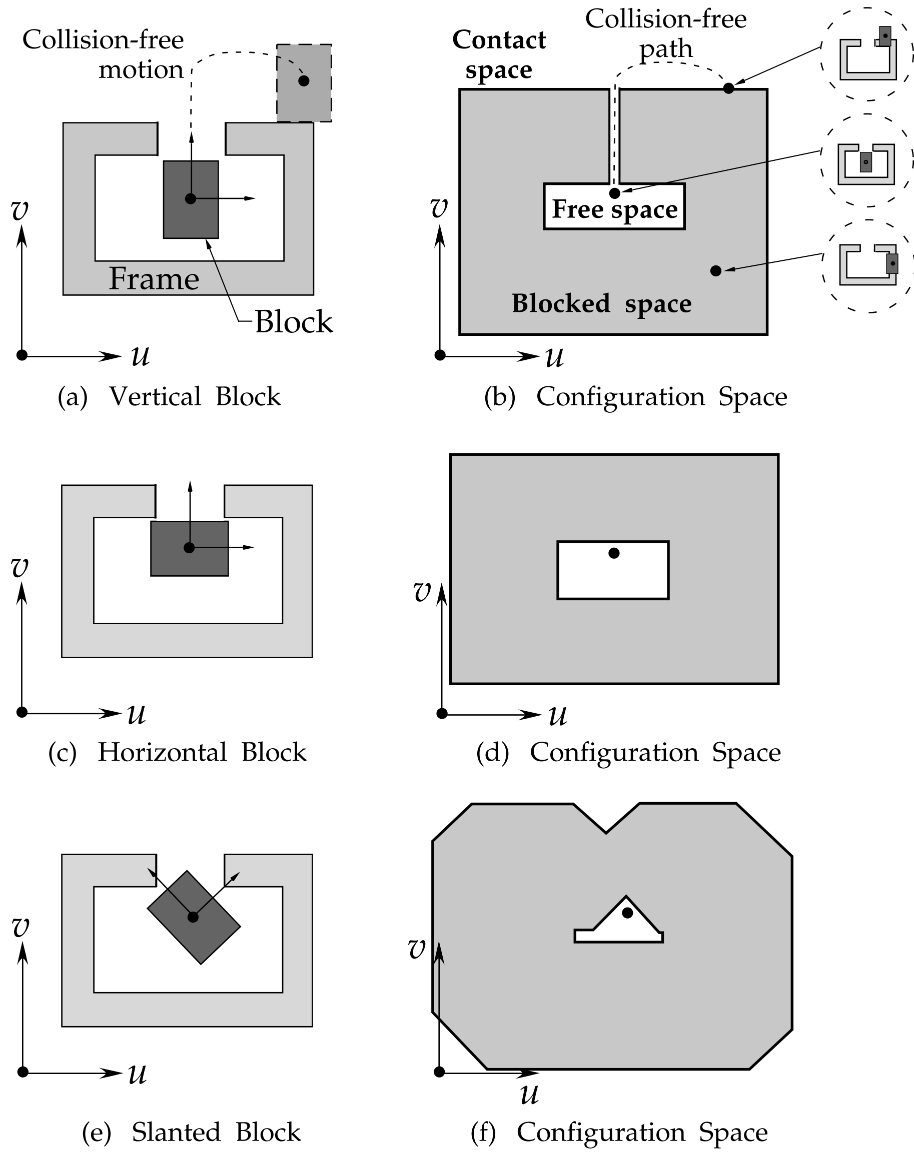


Figure 9.20 A translating block moving around a fixed frame at fixed orientations and their corresponding configuration spaces: vertical orientation (top), horizontal (middle) and slanted (bottom).

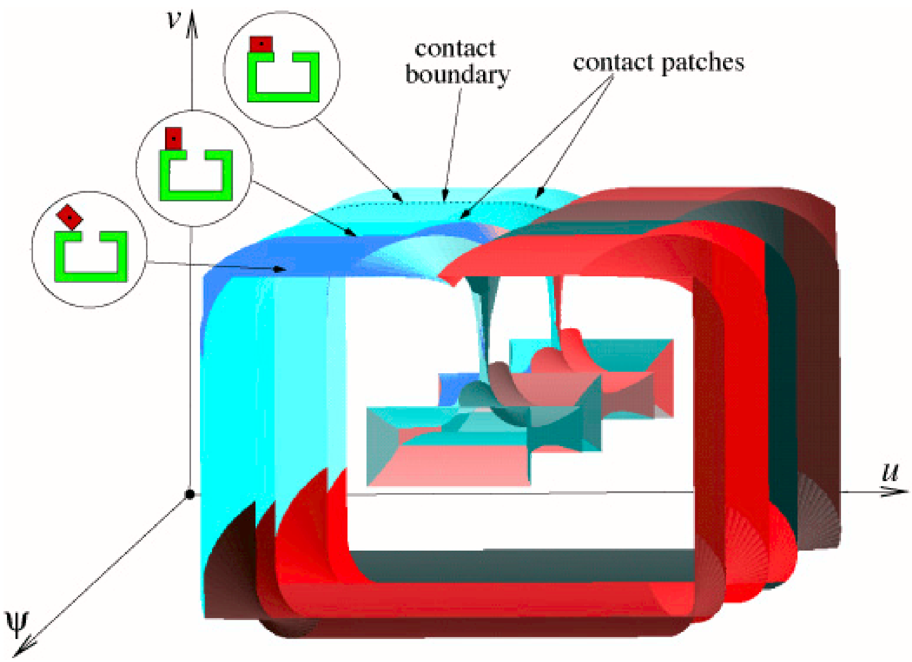


Figure 9.21 Contact space for block with three degrees of freedom—each shade of gray denotes a contact patch.

patches are shown on the left. To understand this space, consider it as a stack of planar slices along the rotation axis. Each slice is the configuration space of a block that translates at a fixed orientation, such as the three examples in Figure 9.20b, d, and e. The full space is the union of the slices. The free space consists of an outer region, two inner regions, and two connecting channels near $\psi = \pm\pi/2$ where the block is nearly vertical. The outer envelope is the union of the outer regions, the inner envelope is the union of the inner rectangles, and the channels are the union of the connecting regions. Blocked space is the region between the envelopes and outside the channels. Contact boundaries, where contact patches intersect, correspond to simultaneous feature contacts.

Whatever its dimension, the configuration space of a pair is a complete representation of the part contacts. Any contact question is answerable by a configuration space query. For example, testing if parts overlap, do not touch, or are in contact in a given configuration corresponds to testing if the configuration point is in blocked, free, or contact space. Contacts between pairs of features correspond to contact constraints (curve segments in two dimensions and surface patches in three). The constraints' geometry encode the motion constraints. Their boundaries encode the contact change conditions. Part motions correspond to paths in configuration space. A path is legal if it lies in free and contact space, but illegal if it intersects blocked space. Contacts occur at configurations where the path crosses from free to contact space, break where it crosses from contact to free space, and change where it crosses between neighboring contact constraints.

The configuration space representation generalizes from pairs of parts to systems with more than two parts. A mechanical system of n parts has a $6n$-dimensional configuration space whose points specify the n part configurations. A system configuration is free when no parts touch, is blocked when two parts overlap, and is in contact when two parts touch and no parts overlap. The mechanical system configuration space can be obtained by combining the configuration spaces of its pairs (Sacks and Joskowicz 1991), since the system is a collection of kinematic pairs (Reuleaux 1875). System configuration spaces allow us to analyze multi-part interactions but are difficult to compute.

7.2 Configuration spaces of the ratchet mechanism

We illustrate how pairwise configuration spaces are used in kinematic analysis and synthesis on the ratchet mechanism of Figure 9.18. There are four interacting pairs: driver/link, link/pawl, pawl/ratchet, and pawl/frame. The link/pawl pair is a revolute joint, and thus has a simple relationship: the pawl is constrained to rotate around the pin axis. The driver/link configuration space

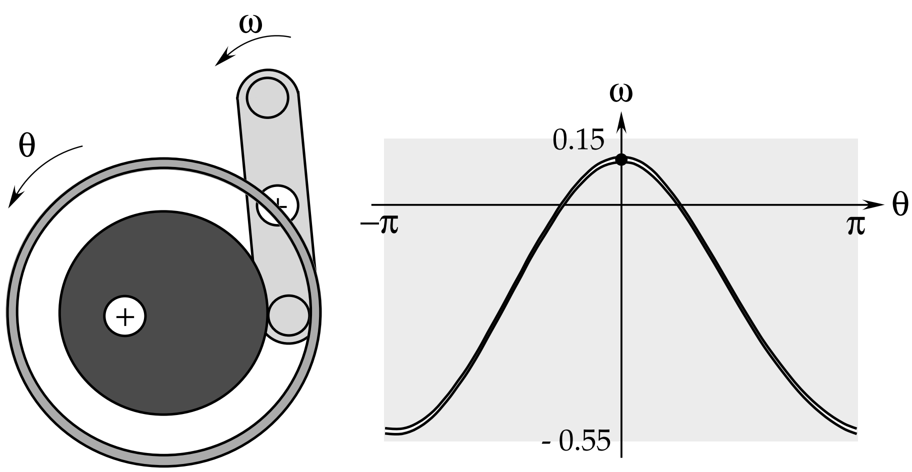


Figure 9.22 Driver/link pair and its configuration space.

is two-dimensional because both are pinned by revolute joints to the base. The pawl/frame and pawl/ratchet configuration spaces are three-dimensional, since the pawl has three degrees of freedom.

Figure 9.22 shows the configuration space of the driver/link pair. The configuration space coordinates are the driver orientation θ and the link orientation ω. The upper and lower contact curves represent contacts between the cylindrical pin and the outer and inner cam profiles. The free space is the region in between. As the driver rotates from $\theta = -\pi$ to $\theta = 0$, its inner profile pushes the link pin right, which rotates the link counter-clockwise from $\omega = -0.47$ radians to $\omega = 0.105$ radians. As the driver rotates from $\theta = 0$ to $\theta = \pi$, the pin breaks contact with the inner profile and makes contact with the outer one, which pulls it left and rotates the link clockwise. The configuration follows the lower contact curve from $\theta = -\pi$ to 0, travels horizontally through free space until it hits the upper contact curve, and follows it to π. The free play is determined by the distance between the curves. Changes in shape and position parameters change the play and can induce blocking.

Figure 9.23 shows the ratchet/pawl pair, a slice of its three-dimensional configuration space, and the three-dimensional contact space of the ratchet moving relative to the pawl. To best understand the figure, consider first the two-dimensional slice in Figure 9.23b, which shows how the ratchet translates in the displayed orientation. The dot marks the displayed position of the ratchet relative to the pawl. It lies on a contact curve that represents contact between the pawl tip and the side of a ratchet tooth. The right end of the curve is the intersection point with a second contact curve that represents contact between the left corner of the pawl and the next tooth counter-clockwise. The ratchet can maintain this contact while translating right until the second contact occurs and further translation is blocked. The topology of the slice is preserved when the orientation of the pawl changes slightly, as can be seen in the contact space Figure 9.23c. Changes in the orientation of the teeth and the ratchet's center of rotation can change the kinematic function of the pair.

7.3 Configuration space computation

Robotics research confirms the empirically observed difficulty of contact analysis with formal proofs that configuration space computation is worst-case exponential in the number of degrees of freedom (Latombe 1991). Despite this result, contact analysis is manageable in practice because mechanical systems have characteristics that distinguish them from arbitrary collections of parts. Mechanism parts usually only interact with a few neighboring parts, are connected by simple joints, have few degrees of freedom, or consist of symmetric patterns of feature groups, *e.g.*, gear teeth. Typical systems have only one or two true degrees of freedom, not many. The challenge is to exploit these properties to develop specialized algorithms for important classes of mechanical systems and design tasks.

The robotics literature contains many configuration space computation algorithms (Latombe 1991), although most are restricted to pairs of polygons and polyhedra. Special-purpose contact analysis methods for gears and cams, for which the contact sequence is known, are described in Angeles and Lopez-Cajun 1991, Gonzales and Angeles 1993 and Litvin (1994). Other higher pairs can be classified as planar or spatial and as fixed-axes (one degree of freedom per part) or general. Fixed-axes planar pairs are by far the most common, followed by fixed-axes spatial pairs and by planar pairs with three degrees of freedom according to a survey of more than 2,500 mechanisms from an encyclopedia (Sacks and Joskowicz 1991). General spatial higher pairs with six degrees of freedom are rare. Efficient algorithms for computing two-dimensional configuration spaces of planar fixed axes pairs whose shapes are formed by arc and line segments (Sacks and Joskowicz 1995), and for fixed-axes spatial pairs whose shapes are formed by planar, cylindrical, and spherical patches bounded by line segments and circular arcs (Drori *et al.*, 1999) are available. Sacks (Sacks 1998) describes an algorithm for computing three-dimensional configuration spaces of general planar pairs with which the figures of this section were generated. There are no algorithms computing the configuration space of general spatial pairs with six degrees of freedom, although robot motion planning research provides algorithms for a polyhedral robot moving amidst fixed polyhedral obstacles (Donald 1987; Joskowicz and Taylor 1996; Latombe 1991).

Configuration space computation consists of partitioning the configuration space of a pair of interacting parts into free, contact, and blocked space. The

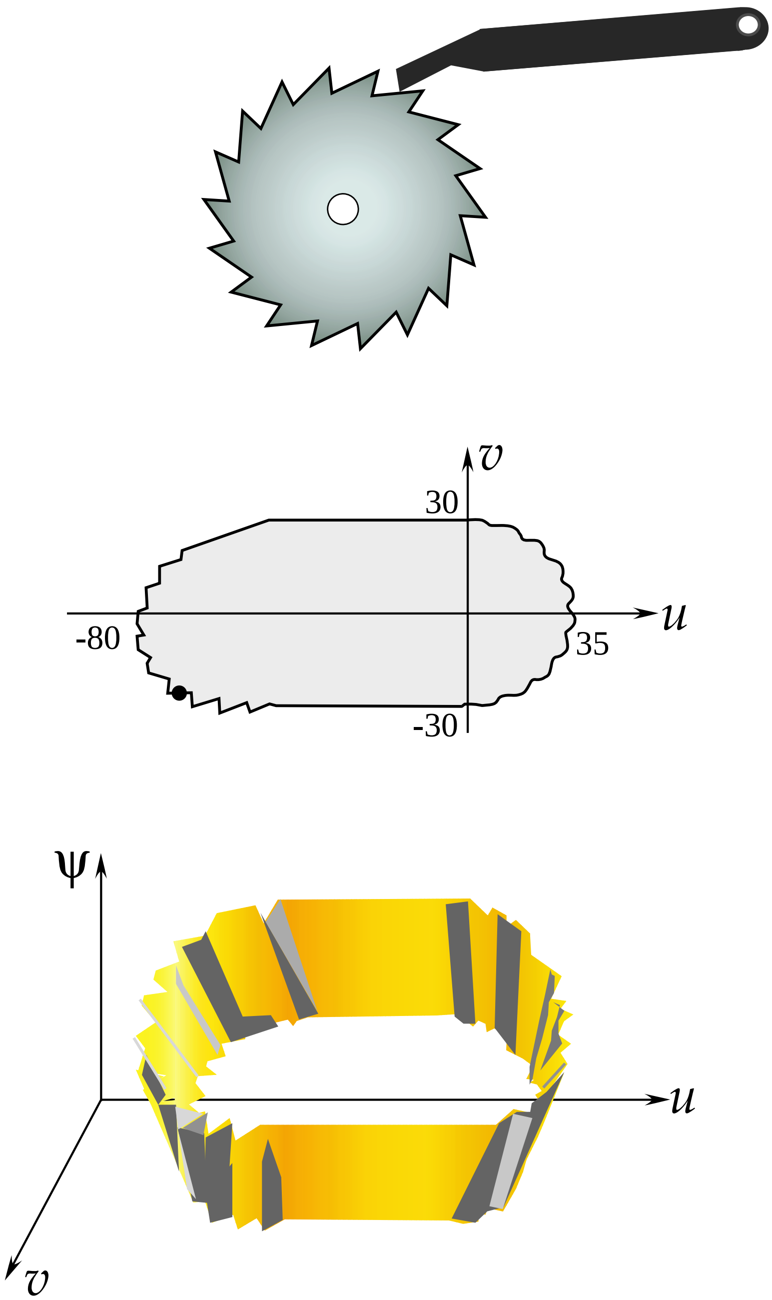


Figure 9.23 (a) Ratchet/pawl pair; (b) configuration space slice at $\psi = 0.277$ radians; (c) contact space; the inner region is blocked space, the outer region free space.

geometric algorithms proceed in two steps: (1) compute the contact constraints for each pair of part features, and (2) compute the partition of the configuration space induced by the contact constraints. We describe each step briefly next.

7.3.1 Contact constraints. The contact constraints are the configurations in which the features would touch if there were no other features to interfere. Contact constraints depend on the types of features in contact and on the part motion. For example, the contact constraints generated by translating line segments are line segments, as illustrated in Figure 9.20b. There is one contact constraint for each combination of part features and motions. For example, there are 16 types of contact constraints for contact constraints for fixed-axes pairs whose parts are polygons, corresponding to all combinations of point and line segment feature pairs and all combinations of rotation and translation motions. The contact constraints are algebraic equalities specifying the contact function and and inequalities specifying the contact boundary conditions. They are curves for two-dimensional configuration spaces (fixed-axes pairs) and patches for three-dimensional configuration spaces (general planar pairs).

Contact constraints can be systematically derived for each combination and stored in parametric form in a table. The parameters are the geometric characteristics of the part features, such as the line segment slope and the arc radius and origin. To obtain the contact constraints of a given feature pair, we substitute the geometric parameters and obtain contact functions and contact inequalities. The derivation of the contact constraints proceeds by formulating the geometric conditions for the features to be in contact, and then substituting the configuration parameters into these expressions to obtain algebraic functions.

We illustrate contact constraint derivation for general planar pairs whose shapes are formed by arc and line segments. There are three types of contact constraints, corresponding to the types of features in contacts and their motions: moving arc/fixed line, moving line/fixed arc, and moving arc/fixed arc. Contacts involving points are identical to those for arcs of radius zero. Line/line contacts are subsumed by line/point contacts. Figure 9.24a shows an arc/line contact. The contact condition is that the distance between the center o of the arc and the line lm equals the arc radius r:

$$(\vec{o}_A - \vec{l}) \times (\vec{m} - \vec{l}) \quad = \quad dr$$

where $\times$ denotes the vector cross-product, d is the length of the line segment, and its interior lies to the left when traversed from l to m. Figure 9.24b shows an arc/arc contact. The contact condition is that the distance between the centers equals the sum of the radii:

$$(\vec{o}_A - \vec{p}) \cdot (\vec{o}_B - \vec{q}) \quad = \quad (r+s)^2$$

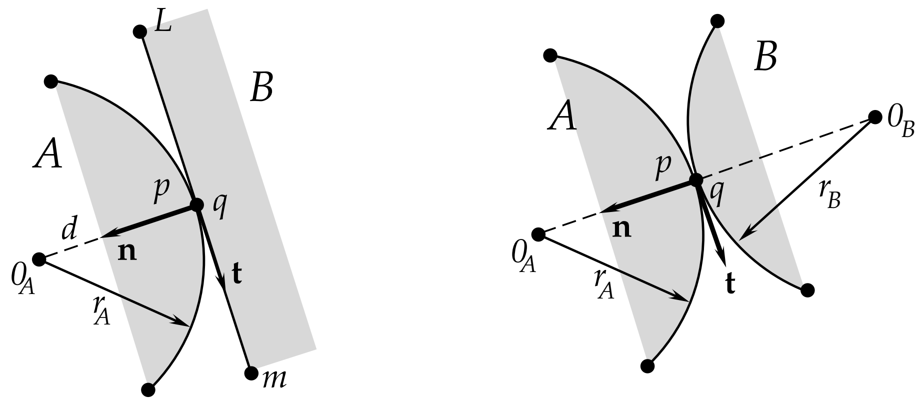


Figure 9.24 Contacts between planar features: (a) circular arc/line segment, (b) two circular arcs. Shading indicates part interior.

where r and s are positive for convex arcs and negative for concave arcs.

We obtain the contact constraint functions from these equations by expressing the vectors in coordinate form. Let $\vec{q} = (q_x, q_y)$ be the coordinates of a point on the fixed part, and let $\vec{p} = (u, v) + R_\psi(p_x, p_y)$ be the coordinates of a moving point, where (p_x, p_y) are the part coordinates and R_ψ a rotation matrix. After substitution, we obtain functions of the form $f(u, v, \psi) = 0$ parameterized by the part features. The moving arc/fixed line function is:

$$\begin{aligned} u(m_y - l_y) - v(m_x - l_x) + (m_y - l_y)(o_x \cos\psi - o_y \sin\psi) - \\ (m_x - l_x)(o_x \sin\psi + o_y \cos\psi) &= dr. \end{aligned}$$

The moving line/fixed arc equation is:

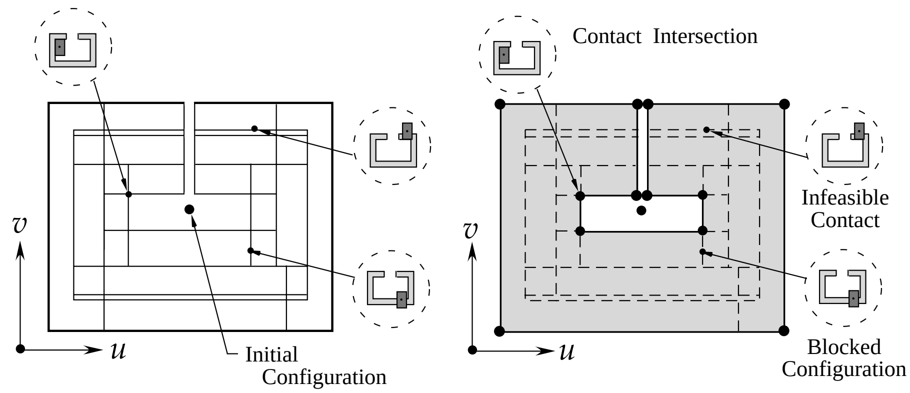
$$\begin{aligned} (u - o_x)(l_x \cos\psi - l_y \sin\psi) - (v - o_y)(l_x \sin\psi + l_y \cos\psi) \\ + l_x(m_y - l_y) - l_y(m_x - l_x) + dr &= 0. \end{aligned}$$

The moving arc/fixed arc equation is:

$$\begin{aligned} u^2 + v^2 + 2(o_x \cos\psi - o_y \sin\psi)u + 2(o_x \sin\psi + o_y \cos\psi)v \\ + o_x^2 + o_y^2 + p_x^2 + p_y^2 - (r + s)^2 &= 0. \end{aligned}$$

Similar equations are obtained for contact ranges. Parametric contact constraints thus derived can be stored in a table and instantiated for particular feature pair geometry. The contact constraints of a pair are obtained by instantiating the parametric constrains for each pair of features.

7.3.2 Configuration space partition. The contact constraints partition configuration space into connected components. The component that



(a) Pairwise contact constraints (b) Induced partition

Figure 9.25 Partition of the fastener configuration space: (a) all pairwise contact constraints; (b) induced partition: solid lines are contact space, dashed lines infeasible contacts.

contains the initial configuration is the reachable free space. Its boundary is a subset of the contact constraints. The other constraints lie in blocked space or form the boundaries of unreachable free space regions. Infeasible contact constraints must be eliminated. This is illustrated in Figure 9.25a. It shows all the pairwise contact constraints for the translating block in Figure 9.20a. Note that the two contact configurations on the left are unrealizable, and thus their contact constraints should be eliminated or restricted. Figure 9.25b shows the contact space (solid lines) partitioning the configuration space. Dashed lines indicate subsumed contacts.

Configuration space partition is computed using computational geometry algorithms. The algorithms compute the intersection between contact constraints and classify free and blocked space accordingly. The algorithms are based on two and three-dimensional geometric operations involving curve and surface intersection and sweeping algorithms (Sacks and Joskowicz, 1995; Sacks 1998; Drori *et al.*, 1999).

7.4 Mechanism design with configuration spaces

Configuration spaces provide the computational basis for a wide variety of mechanism design tasks, including tolerancing, assembly, and shape and motion synthesis. The goal is to study system function under a range of operating conditions, find and correct design flaws, and optimize performance.

7.4.1 Tolerancing. Tolerancing consists of determining the variations in the system function due to manufacturing variation in its parts. Manufac-

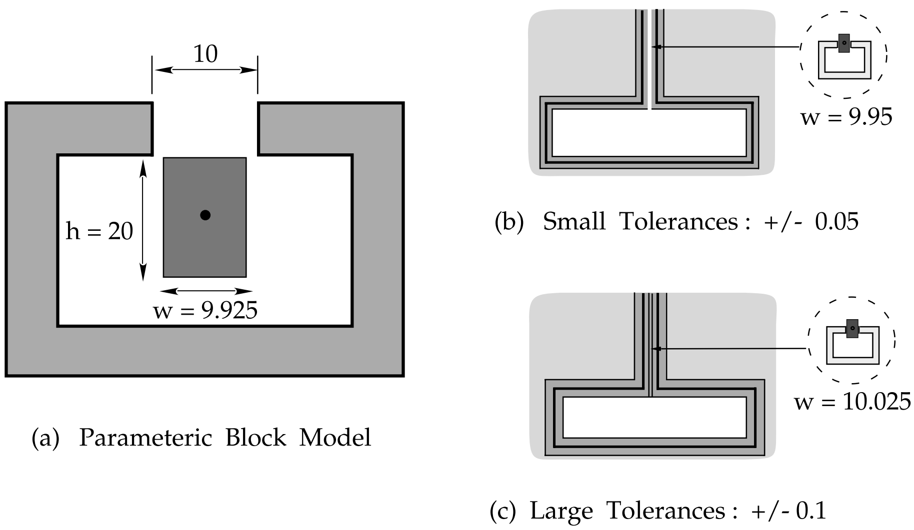


Figure 9.26 Parametric block model and details of the generalized configuration space with different tolerances.

turing variations are often expressed as tolerances, which correspond to intervals around nominal part shape and position values. For example, assume that the moving block in Figure 9.26 has nominal height $h = 20$mm and width $w = 9.925$mm (the frame has a fixed opening of 10mm). A small variation corresponds to a tight tolerance interval, *i.e.*, ± 0.05mm, whereas a larger variation corresponds to a looser interval, *i.e.*, ± 0.1mm. The variations in the system function can be qualitative and quantitative. Qualitative variations occur when part parameter variations can cause unintended contact effects, such as jamming or interference with a frame. Quantitative variations determine the worst-case and average error in the system function. For example, the looser tolerance interval on the block width can prevent the block from exiting the frame (a qualitative change), whereas the smaller tolerance interval bounds the play between the block and the frame (a quantitative measure).

Kinematic variation can be modeled within the configuration space representation (Sacks and Joskowicz 1998b). As the parameters vary around their nominal values, the contact configurations vary around a zone around the nominal contact space. The zones define the set of all configurations for which there is one or more combinations of parameter values within the tolerance interval that generates a contact. The boundaries of the contact zone represent the worst-case variation. A standard first order approximation of the contact zone boundaries can be obtained by summing up the partial derivatives of the parametric contact functions.

Figures 9.26b and 9.26 c illustrate these concepts. They show a detail of the nominal configuration space (solid black lines and gray shading) and the contact zones (blue lines and blue shading) corresponding to the tolerance intervals. Zone boundaries (blue lines) correspond to worst-case variations. Configurations in the zone correspond to specific part instances in contact (inserts on the right). The distance between the inner and outer upper vertical zone boundaries quantify minimum and maximum play. For small tolerances, the topology of the space is preserved, meaning that the function of the system remains unchanged and the play interval is $[0.025, 0.125]$mm. However, for the larger tolerances, the zone boundaries cross, blocking the channel. This means that for some block dimensions within the tolerance interval, the block will not fit through the frame since the inner configuration space region is now disconnected from the outer one.

7.4.2 Assembly. Assembly consists of finding a set of part motions that bring the parts of a mechanism in their working configuration. It is a kind of motion planning problem, in which the starting configuration is the unassembled parts, the assembly sequence is a set of collision-free assembly part motions for parts, and the goal configuration is the assembled mechanism in its working configuration. Assembly planning is an integral part of mechanism design, as it influences part shapes and positions. Because it deals with part contacts, the configuration space framework described in this chapter is an appropriate framework to study the problem. Note that the assembly part motions are usually different from the motions that the parts have when functioning.

The main difficulty is that assembly motions are usually spatial, so parts can have up to six degrees of freedom, which precludes pairwise configuration space computation. Many approaches have been developed (Homem de Mello and Lee 1991), some based on local geometric structures called blocking graphs (Wilson *et al.*, 1994). Latombe *et al.*, (1997), Blind *et al.*, 2000, and Xiao and Xuerang 2000 describe an algorithm for assembly planning with toleranced parts based on configuration space.

7.4.3 Shape synthesis. One aspect of mechanism synthesis is part shape synthesis and optimization. The goal is to create and modify part shapes to satisfy design requirements. Special-purpose approaches have been developed for cam (Angeles and Lopez-Cajun 1991, Gonzales and Angeles 1993) and for gear (Litvin 1994) pairs. For other pairs, configuration space based approaches have been proposed.

Shape synthesis can be seen as the inverse process of analysis: given a parametric part model and a design specification represented as a configuration space, the goal is to invert the mapping from parameter values to configuration spaces so that the values yield the desired configuration space (Joskowicz and Sacks 1994). For non-parametric part models, part boundaries can be modified using heuristic methods. (Joskowicz and Addanki 1988; Faltings and Sun, 1996). Similar techniques have been applied to feeding track design (Caine 1994).

A related task is fixture synthesis, where the goal is to design fixture shapes that eliminate the movement of parts to be machined. Eliminating movement corresponds to reducing the volume of the configuration space surrounding the machining configuration to zero (Rimon and Blake 1996; Brost and Goldberg 1996; Rimon and Burdick 1998).

7.4.4 Motion synthesis. Another aspect of mechanism synthesis is motion synthesis. Motion synthesis is the task of designing a mechanism topology that transforms a set of input motions into desired output motions. The mechanism topology is built as a chain of basic kinematic pair building blocks, such as joints, cams, gears, and other higher pairs. Hoover and Rinderle (1989) developed a heuristic method based on functional integration for speed ratio and geometric requirements of gear transmission mechanisms. Kota and Chiou (1992) represent kinematic pair function with matrices which are chained to obtain the desired output. Chakrabarti and Bligh (1996) successively refine motion type requirements based on kinematic pair behavior.

Configuration space provides a representation of mechanism function that has been used for mechanism retrieval (Joskowicz 1990; Murakami and Nakajima 1997) and motion synthesis (Subramanian and Wang 1995; Li *et al.*, 1999). The basic building blocks are kinematic pairs, whose function is described with a qualitative representation of their configuration space. The synthesis algorithm generates design solutions by forming chains of kinematic pairs that match the desired mechanism function. The advantages of these methods are that they allow for qualitative function descriptions akin to those used in conceptual design (Stahovich *et al.*, 1998), and that they allow for the design of multiple-state mechanisms. To date, they are limited to simple planar mechanisms.

8. State of the Art

The kinematic synthesis of machines ranging from the lever to the robot has provided remarkable capabilities for redirecting power to our ends. Furthermore, it seems clear that the future holds more opportunities for mechanical systems that are tailored to enhance and augment our individual capabilities whether at the human or micro-scale.

Research in the kinematic synthesis of mechanisms and robots has demonstrated the central importance of workspace and mechanical advantage as design criteria. The workspace, as defined by the kinematics equations, can be used for design, tolerancing, and assembly planning. The Jacobian of these equations yields the speed ratios of the system, which, via the principle of virtual work, define its mechanical advantage. And, optimization techniques have proven effective for fitting the workspace and mechanical advantage of serial and parallel chain systems to design specifications.

Unfortunately, computer tools for kinematic synthesis seem to exist only as special purpose algorithms developed by individual researchers. Commercial systems, such as LINCAGES 2000 (Univ. of Minnesota), SyMech (www.symech.com), and Watt (heron-technologies.com), focus solely on planar four- and six-bar linkages. Kinematic synthesis in engineering applications seems to consist of iterated analysis, in which little or no attempt is made to use workspace and mechanical advantage criteria to generate designs.

A systematic procedure for the design of both serial and parallel linkages and robots that can match workspace and mechanical advantage is needed, especially one that allows comparison of different machine topologies. Consider the following enumeration of spatial open chains of various degrees-of-freedom,

- 2 dof chains: 2R, RP, C, T.
- 3 dof chains: 3R, RRP, PPR, CR, CP, TR, RP, S.
- 4 dof chains: 4R, 3PR, 3RP, CRR, CPR, CC, TRR, TPR, TC, TT, RS, PS.
- 5 dof chains: 5R, 3PRR, RRS, PPS, CPC, CRC, CS, TRC, TPC, 2TR, 2TP, TS.

Only a few of these topologies have been explored for use in the design of spatial linkage systems. Clearly computer automation of the synthesis of these chains and their assembly into parallel systems can open the door to a wealth of new devices. This is particularly true when asymmetry exists in the task, or the device must work around obstacles and people.

Configuration space analysis provides new capabilities for automation of the design and manufacture of mechanical systems. This is an extension of the concept of workspace to include all bodies not just the end-effector (Joskowicz and Sacks 1999). In this formulation the free space boundary for each part is obtained by analyzing the contact available with every other part. The set of positions and orientations of a body can be parameterized in many useful ways. Quaternions have been found to provide a convenient representation for spatial rotations (Bottema and Roth 1979). Clifford Algebras have been used to parameterize planar and spatial displacements in order to obtain algebraic surfaces that represent geometric constraints (McCarthy 1990, Collins and McCarthy 1998, Ge *et al.*, 1998).

The generalization of Bezier curves to spatial motion provides a convenient way to specify continuous taskspaces. However, the extension of this technique to surfaces, solids, hypersolids, or hypersurfaces in SE(3) does not exist. This means that we cannot directly specify the workspace of a robotic system. Currently, we rely on general symmetry requirements.

Methodologies for the design of cam systems, gear trains, and linkages have yet to be integrated into computer aided engineering tools. While sculptured surfaces are central to the operation of human and animal joints, they have yet to be exploited for robotics applications (Lenarčič *et al.*, 2000, Parenti-Castelli and DiGregorio 2000).

Micro-electro-mechanical systems provide unique opportunities for multiple small systems constructed using layered manufacturing technologies typical of electronic devices. The small size of these systems challenges the basic assumptions of virtual work and their design has benefited from the synthesis theory for compliant linkages (Ananthasuresh and Kota 1995). The design of micro-robotic systems requires advances in the design and construction of joints and actuators (Will 2000).

A speculative direction for kinematic synthesis research involves the analysis of proteins and rational drug design (Wang *et al.*, 1998, LaValle *et al.*, 2000). A protein can be viewed a serial chain of links, called amino acids, each of which are similar except for molecular radicals that extend to the side. A protein chain may contain 1000 amino acids which folds into a complex spatial configuration. This configuration can shift between two positions in order to perform a task, exactly like the machines that we have been considering in this chapter.

9. Conclusion

This chapter surveys the theoretical foundation and current implementations of kinematic synthesis for the design of machines. The presentation is organized to illustrate the importance of the concepts of workspace and mechanical advantage in the synthesis process. While the workspace of a robot arm is a familiar entity, it is not generally recognized that concepts ranging from the geometric constraints used in linkage design, through configuration spaces used to define tolerances are simply different representations of workspace. Similarly, while mechanical advantage is clearly important to the design of a lever and wedge, it is not as obvious that the Jacobian conditions used in the design of robotic systems are, in fact, specifications on mechanical advantage.

The result of this study are the conclusions that (i) workspace and mechanical advantage are effective specifications for the synthesis of a broad range of mechanical devices; (ii) there already exist a large number of specialized algorithms that demonstrate the effectiveness of kinematic synthesis; and (iii) there are many opportunities for the application of new devices available through kinematic synthesis. What is needed is a systematic development of computer tools for kinematic synthesis which integrates the efforts of a community of researchers from mechanical design, robotics, and computer science.

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J. Michael McCarthy received his Ph.D. in Mechanical Engineering at Stanford University in 1979. He has worked for FMC Corp. taught at Loyola Marymount University, and the University of Pennsylvania before joining the faculty at the University of California, Irvine, in 1986. He has published over 100 technical articles in refereed journals , conference proceedings and book chapters. He edited *The Kinematics of Robot Manipulators*, and authored *An Introduction to Theoretical Kinematics*, both published by MIT Press, and recently published *Geometric Design of Linkages* with Springer-Verlag. His research team produced the interactive graphics software *SphinxPC* for planar and spherical linkage design, and developed award-winning assistive devices for the wheel-chair bound. His current research is in the kinematic synthesis of spatial mechanisms and robots.

Leo Joskowicz is an Associate Professor at the School of Computer Science and Engineering at the Hebrew University of Jerusalem, Israel, where he conducts research in computer-aided mechanical design, computer-assisted surgery, and robotics since 1995. He obtained his Ph.D. in Computer Science at the Courant Institute of Mathematical Sciences, New York University, in 1988. From 1988 to 1995, he was at the IBM T.J. Watson Research Center, Yorktown Heights, New York, where he conducted research in intelligent computer-aided design and computer-aided orthopaedic surgery. From 1994 to 1995, he was the project leader of a U.S. government funded joint project on computer-integrated revision total hip replacement surgery. In 1996, he founded the Computer-Aided Surgery and Medical Image Processing Laboratory, which he heads. Prof. Joskowicz is a member of the editorial boards of *Artificial Intelligence in Engineering*, *Annals of Mathematics and Artificial Intelligence*, the *Journal of Computer-Aided Surgery*, and *Medical Image Analysis*. He has published extensively in these areas and has served on numerous related program committees.

Chapter 1

VITRUVIUS REDUX

Formalized Design Synthesis in Architecture

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Abstract

Keywords: Architecture Formalized Design Synthesis

William J. Mitchell is a Professor of Architecture and Media Arts and Sciences and the Dean of the School of Architecture and Planning at the Massachusetts Institute of Technology in Cambridge, Massachusetts.

Mitchell teaches courses and conducts research in design theory and computer applications in architecture and urban design and holds a joint appointment with the Program in Media Arts and Sciences. Recent classes have included a workshop in software design and a seminar on digital communities (both with Prof. Mitchell Kapor) and design studios taught with Prof. Andrew Scott. He also serves as Architectural Advisor to the President of MIT. His most recent book is the edited volume High Technology and Low-Income Communities, with Donald A. Schön and Bish Sanyal (MIT Press, 1999). And his E-Topia: Our Town Tomorrow, which explores the new forms and functions of cities in the digital electronic era, will be published by the MIT Press in Fall 1999. His book, City of Bits: Space, Place, and the Infobahn, examines the architectural and urbanistic implications of the digital telecommunications revolution, the ongoing miniaturization of electronics, the commodification of bits, and the growing domination of software over materialized form, discussing the emergent but still invisible cities of the twenty-first century. It was released simultaneously in hardcover and on-line by The MIT Press in 1995. Mitchell's earlier works on computers and design have included overviews of the state of the field (most recently Digital Design Media, with Malcolm McCullough, 2nd ed., 1995) and theoretical analysis including The Logic of Architecture: Design, Computation, and Cognition (1990). His other major works include The Reconfigured Eye: Visual Truth in the Post-Photographic Era (1992) and, with Charles Moore and William Turnbull, The Poetics of Gardens (1988). Before coming to MIT, he was the G. Ware and Edythe M. Travelstead Professor of Architecture and Director of the Master in Design Studies Program at the Harvard Graduate School of Design. He previously served as Head of the Architecture/Urban Design Program at UCLA's Graduate School of Architecture and Urban Planning, and he has also taught at Yale, Carnegie-Mellon, and Cambridge Universities. In Spring 1999 he held a visiting appointment at the University of Virginia as Thomas Jefferson Professor. He holds a BArch from the University of Melbourne, MED from Yale University, and MA from Cambridge. He is a Fellow of the Royal Australian Institute of Architects, a Fellow of the American Academy of Arts and Sciences, and a recipient of honorary doctorates from the University of Melbourne and the New Jersey Institute of Technology. In 1997 he was awarded the annual Appreciation Prize of the Architectural Institute of Japan for his "achievements in the development of architectural design theory in the information age as well as worldwide promotion of CAD education.

Chapter 2

HOW TO CALCULATE WITH SHAPES

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> *An interesting question for a theory of semantic information is whether there is any equivalent for the engineer's concept of noise. For example, if a statement can have more than one interpretation and if one meaning is understood by the hearer and another is intended by the speaker, then there is a kind of semantic noise in the communication even though the physical signals might have been transmitted perfectly.*
>
> —George A. Miller

Abstract

Keywords: Shape grammars, Engineering grammars

George Stiny is a Professor of Design and Computation at the Massachusetts Institute of Technology in Cambridge, Massachusetts. Dr. Stiny joined the Department of Architecture in 1996 after fifteen years on the faculty of the University of California, Los Angeles. Educated at MIT and at UCLA, where he received a Ph.D. in Engineering. He has also taught at the University of Sydney, the Royal College of Art (London), and the Open University. Dr. Stiny's particular contribution to the field has been the invention and refinement of the idea of shape grammars, and his work stands as a critique of the vast majority of existing computational methods in design. He is currently working on a book on shape to be published by the MIT Press and is the author of *Pictorial and Formal Aspects of Shape and Shape Grammars*, and of *Algorithmic Aesthetics: Computer Models for Criticism and Design in the Arts* with J. Gips.

Chapter 3

ENGINEERING SHAPE GRAMMARS

Where Have We Been and Where are We Going?

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Abstract The state of the art in engineering applications of shape grammars is discussed, motivating underlying decisions that must be made during the design of an engineering shape grammar. Open research questions point to the future of the technology in engineering domains.

Keywords: Shape grammars, Engineering grammars

Jonathan Cagan is a Professor of Mechanical Engineering at Carnegie Mellon University, with appointments in the School of Design, Biomedical and Health Engineering and Computer Science. His research, teaching, and consulting are in the area of design theory, methodology, automation, and practice. Dr. Cagan has served as Associate Technical Editor of the ASME Journal of Mechanical Design in the area of Design Theory and Methodology (DTM); he serves on the editorial board of Research in Engineering Design; he has served as the Chair of the ASME DTM committee and of its annual conference. Dr. Cagan held the Ladd Development Chair and is a recent recipient of the Dowd Fellowship in the College of Engineering at CMU. He received his B.S. in 1983 and M.S. in 1985 from the University of Rochester, and his Ph.D. in 1990 from the University of California at Berkeley, all in Mechanical Engineering. Dr. Cagan is a Fellow of the ASME. Prof. Cagan is the recipient of the National Science Foundation's NYI Award and the Society of Automotive Engineer's Ralph R. Teetor Award for Education. He is a member of the Phi Beta Kappa, Tau Beta Pi, and Sigma Xi National Honor Societies, and the ASME, IDSA, AAAI, SAE, and ASEE Professional Societies. Dr. Cagan is a registered Professional Engineer.

Chapter 4

CREATING STRUCTURAL CONFIGURATIONS

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Abstract

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Panos Papalambros is the Donald C. Graham Professor of Engineering and a Professor of Mechanical Engineering at the University of Michigan. He teaches in the field of design.

Born in Patras, Greece, he attended the National Technical University of Athens (Ethnikon Metsovion Polytechnion) and earned a diploma in Mechanical and Electrical Engineering in 1974. Moving to California he attended Stanford University and earned his M.S. degree (Mechanical Engineering) in 1976 and Ph.D. degree (Design Division, Mechanical Engineering) in 1979. At Michigan he has served as a faculty member since 1979.

During his tenure at Michigan he served as department chair (1992-98) and became the founding director of several laboratories and centers: Design Laboratory (1990-92); Ford Durability Simulation Center (1992-93); Automotive Research Center (1994-2000, now serving as executive director); the UM/GM Satellite Research Laboratory (1998-). His research interests include design methods and systems design optimization, with applications to product design, automotive systems, and product development. With D. J. Wilde, he co-authored the textbook *Principles of Optimal Design: Modeling and Computation* (1988, 2000 2nd edition).

A member of ASME, INFORMS, MPS, SIAM, SME, SAE, ISSMO, AIAA and ASEE, he serves on the editorial boards of the journals Artificial Intelligence in Engineering Design and Manufacturing, Engineering Design, Engineering Optimization, Integrated Computer-Aided Engineering, Structural and Multidisciplinary Optimization, and the International Journal of Engineering Simulation (a web-based journal). He has also served the ASME Journal of Mechanical Design, the Journal of Global Optimization, and the Japan Society of Mechanical Engineers International Journal.

A Fellow of ASME, he is the recipient of the ASME Design Automation Award (1998) and the Machine Design Award (1999).

He maintains strong ties with the University of Patras where he spends much of his summer time, often gazing at the view of the Patraikos Bay on the coast of Rion or just sailing down the coast.

Kristina Shea was appointed Lecturer in Engineering Design at the University of Cambridge in 1999 where she is involved in teaching undergraduate design courses. She is affiliated with the Engineering Design Centre and a fellow of Jesus College. She received her B.S. with University Honors in 1993 from the Mechanical Engineering Department at Carnegie Mellon. She continued on at Carnegie Mellon working in the area of computational synthesis to complete her M.S. (1995) and Ph.D. (1997). Her development of a generative structural design system, called eifForm, has led to collaborative teaching projects with the Department of Architecture at M.I.T. (USA). After completing her Ph.D., she held a postdoctoral research appointment in the Department of Civil Engineering at the Swiss Federal Institute of Technology in Lausanne (EPFL) where she helped start a new research direction in Intelligent Structures and continues to collaborate as an academic guest. While at EPFL she also co-developed a course on Topics in Computer-Aided Engineering. Her primary research interest is the use of computation to expand design possibilities. She is especially interested in creating tools that support performance driven design exploration in the early phases of engineering and architectural design. Her special interests include parametric synthesis, grammatical methods, form/function relations, optimization, performance metrics, artificial intelligence in engineering, software engineering and the integration of advanced computational tools in the design process. She is a member of ASME and EG-SEA-AI.

Chapter 5

MICROSYSTEM DESIGN SYNTHESIS

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It is typical of many kinds of design problems that the inner system consists of components whose fundamental laws of behavior are well known. The difficulty of the design problem often resides in predicting how an assemblage of such components will behave.

—Herbert A. Simon, *The Sciences of the Artificial*, 1969, MIT Press

Abstract Electro-mechanical devices and systems, constructed at length scales of microns to millimeters, fabricated with processes borrowed from micro-electronics, have become commercially successful in a variety of applications. This chapter addresses the issues relating to the automatic design synthesis of microsystems. Several research groups have made noteworthy progress in this area, including two approaches using composition of parametrically scalable primitive elements, and one approach using geometry synthesized by a stochastic exploration procedure. The current status of microsystem design and design research is reviewed, and an example of automated microsystem synthesis is presented. A discussion of future needs and trends in this area, illustrating the crucial role automated and structured design synthesis capabilities will play in the future developments of microsystmes, concludes the chapter.

Keywords: Microsystems, MEMS, Automated Synthesis, Stochastic Exploration, Genetic Algorithms

Erik K. Antonsson is Executive Officer (Dept. Chair) and Professor of Mechanical Engineering at the California Institute of Technology in Pasadena, CA, U.S.A., where he organized the Engineering Design Research Laboratory and has conducted research and taught since 1984.

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In 1983 he joined the Mechanical Engineering faculty at the University of Utah, as an Assistant Professor. In 1984 he became the Technical Director of the Pediatric Mobility and Gait Laboratory, and an Assistant in Bioengineering (Orthopaedic Surgery), at the Massachusetts General Hospital. He also simultaneously joined the faculty of the Harvard University Medical School as an Assistant Professor of Orthopaedics (Bioengineering).

He was an NSF Presidential Young Investigator from 1986 to 1992, and won the 1995 Richard P. Feynman Prize for Excellence in Teaching.

Dr. Antonsson is a Fellow of the ASME, and a member of the IEEE, SME, ACM, ASEE, IFSA, and NAFIPS.

He teaches courses in engineering design, computer aided engineering design, machine design, mechanical systems, and kinematics. His research interests include formal methods for engineering design, design synthesis, representing and manipulating imprecision in preliminary engineering design, rapid assessment of early designs (RAED), structured design synthesis of micro-electro-mechanical systems (MEMS), and digital micropropulsion microthrusters.

Dr. Antonsson is currently on the editorial board of the International Journals: *Research in Engineering Design*, and *Fuzzy Sets and Systems*, and from 1989 to 1993 served as an Associate Technical Editor of the ASME Journal of Mechanical Design, (formerly the *Journal of Mechanisms, Transmissions and Automation in Design*), with responsibility for the Design Research and the Design Theory and Methodology area. He serves as a member of the Engineering and Applied Science Division Advisory Group, and as Chairman of the Engineering Computing Facility at Caltech. He was a member of the Caltech Faculty Committee on Patents and Relations with Industry from 1992 to 1999, and since 1990 has been a member of the CALSTART Technical Advisory Committee. He has published over 95 scholarly papers in the engineering design research literature, and holds 4 U.S. Patents. He is a Registered Professional Engineer in California, and serves as an engineering design consultant to industry, research laboratories (including NASA's Jet Propulsion Laboratory and the 10 meter W. M. Keck Telescope), and the Intellectual Property bar.

Chapter 6

FUNCTION-BASED SYNTHESIS METHODS IN ENGINEERING DESIGN

State-of-the-Art, Methods Analysis, and Visions for the Future

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Abstract Concept generation is at the heart of engineering design. This chapter considers an emerging tool set for generating concepts: function-based synthesis methods. To understand this tool set, the prominent methods in the field are reviewed and summarized, with a subset being investigated with more technical rigor. In addition to this review and investigation, the methods are analyzed against three models: a method architecture, design process, and research model. This analysis extracts the fundamental features of the methods, provides a basis for comparison, and elicits future research opportunities, directions, and industrial applications. Through this analysis, a clear picture emerges of the function-based synthesis field: Many fundamental research results have been realized, and it is just a matter of time before we have tools to assist product development teams in generating dynamic systems, kinematic structures, and the skeletal backbones of consumer products.

Keywords: Function, Synthesis

Kristin L. Wood is currently a Professor of Mechanical Engineering, Mechanical Systems and Design Division at The University of Texas at Austin. Dr. Wood completed his M.S. and Ph.D. degrees in Mechanical Engineering (Division of Engineering and Applied Science) at the California Institute of Technology, where he was an AT&T Bell Laboratories Ph.D. Scholar. He received his Bachelor of Science in Engineering Science (Magna cum Laude, minor in mathematics) from Colorado State University, May 1985. Dr. Wood joined the faculty at the University of Texas in September 1989 and established a computational and experimental laboratory for research in engineering design and manufacturing, in addition to a teaching laboratory for prototyping, reverse engineering measurements, and testing. He was a National Science Foundation Young Investigator and is currently the June and Gene Gillis Endowed Faculty Fellow in Manufacturing. Dr. Wood has received three ASME Best Research Paper Awards, an ASEE Best Paper Award, the Keck Foundation Award for Excellence in Engineering Education, the ASEE Fred Merryfield Design Award, the NSPE AT&T Award for Excellence in Engineering Education, the ASME Curriculum Innovation Award, and the Engineering Foundation Faculty Excellence Award. He is currently supervising a number of graduate students on projects related to product design, development, and evolution. Such projects include structured modeling and control, new manufacturing processes, design for manufacturing and tolerance methods, feature-based design, micro-automation and micro-deformable surfaces for journal and thrust bearings, reverse engineering, design for the environment, rapid prototyping, and design teaching methods for kindergarten through graduate levels. Dr. Wood also annually teaches a number of outreach short courses in Design Technology and Engineering for America's Children, he was the conference and committee chair for the annual ASME Design Theory and Methodology Conference, he co-authored a textbook on product design, and he is currently an Associate Editor for the ASME Journal of Mechanical Design.

The current and near-future objective of Dr. Wood's career is to develop design strategies, representations, and languages that will result in more comprehensive design tools and design teaching aids at the college, pre-college, and industrial levels.

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Chapter 7

ARTIFICIAL INTELLIGENCE FOR DESIGN

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Abstract This chapter examines the application of Artificial Intelligence (AI) to engineering design. AI is the study of knowledge representations and inference mechanisms necessary for reasoning and problem solving. This chapter focuses on those AI techniques that are the most useful for design synthesis: search, knowledge-based systems, machine learning, and qualitative physical reasoning. The theory behind these techniques is described and their strengths and weaknesses in the context of synthesis are discussed. Examples of their application to synthesis are provided. These are drawn primarily from mechanical engineering, however the techniques are suitable for a wide variety of design problems. The chapter concludes with a discussion of a few important areas for future research.

Keywords: Artificial Intelligence, Design, Synthesis

Thomas F. Stahovich is an Assistant Professor in the Mechanical Engineering Department at Carnegie Mellon University, where he is the director of the Smart Tools Lab. He received a B.S. in Mechanical Engineering from the University of California at Berkeley in 1988 and an S.M. and Ph.D. in Mechanical Engineering from the Massachusetts Institute of Technology in 1990 and 1995. His research interests focus on creating intelligent software tools for engineering design. Current projects include: sketch interpretation techniques to enable sketch-based design and analysis tools, techniques for capturing and reusing design knowledge, techniques for automatically documenting designs, and techniques for managing design modifications in large-scale engineered systems. Professor Stahovich is a member of Pi Tau Sigma, Tau Beta Pi, and Sigma Xi National Honor Societies. He is also a member of the American Association of Artificial Intelligence and the American Society of Mechanical Engineers.

Chapter 8

EVOLUTIONARY AND ADAPTIVE SYNTHESIS METHODS

Cin-Young Lee, Lin Ma, and Erik K. Antonsson

The simplest scheme of evolution is one that depends on two processes; a generator and a test. The task of the generator is to produce variety, new forms that have not existed previously, whereas the task of the test is to cull out the newly generated forms so that only those that are well fitted to the environment will survive.

—Herbert A. Simon, *The Sciences of the Artificial*, 1969, MIT Press

Abstract Synthesis of novel engineering designs often requires experimental exploration with a wide range of different configurations. Evolutionary and adaptive exploration methods have successfully synthesized novel design configurations in several engineering application areas, including VLSI, pattern packing, mechanical structures and mechanisms. An adaptive search method alters its selection mechanism and/or search operators in response to the structure of the performance landscape. These methods stochastically refine individual candidate solutions in a population, evaluate the fitness or performance of these new candidates, and keep only those with good fitness values for the next iteration. An overview of evolutionary and adaptive search methods is presented, in the context of their application to engineering design synthesis, including several examples and a discussion of future research trends in this area.

Keywords: Evolutionary Computation, Genetic Algorithms, Simulated Annealing, Tabu Search, Multiagent Systems

Cin-Young Lee is currently pursuing a Ph.D. in Mechanical Engineering at the California Institute of Technology where he received a M.S. degree in the same field. Prior to arriving at the California Institute of Technology, he obtained a dual B.S. degree in Mechanical Engineering and Materials Science from the University of California at Berkeley. His interests lie primarily in the field of soft computing which, in contrast to conventional (hard) computing, attempts to deal with imprecision, uncertainty, and partial truth through the cooperative use of fuzzy logic, neural networks, and probablistic reasoning (which subsumes evolutionary computation, belief networks, chaos theory, and parts of learning theory).

Lin Ma is currently pursuing a Ph.D. in Mechanical Engineering at the California Institute of Technology where he received a M.S. degree in Mechanical Engineering in 1997. He earned a B.S. degree in Mechanics & Engineering Science from Beijing University, China, in 1996. His research interest is in the field of design and synthesis of Micro Electro Mechanical Systems (MEMS). His interests also include utilizing stochastic methods to solve design and optimization problems.

Erik K. Antonsson's biography can be found at the end of Chapter 5.

Chapter 9

KINEMATIC SYNTHESIS

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Leo Joskowicz is an Associate Professor at the School of Computer Science and Engineering at the Hebrew University of Jerusalem, Israel, where he conducts research in computer-aided mechanical design, computer-assisted surgery, and robotics since 1995. He obtained his Ph.D. in Computer Science at the Courant Institute of Mathematical Sciences, New York University, in 1988. From 1988 to 1995, he was at the IBM T.J. Watson Research Center, Yorktown Heights, New York, where he conducted research in intelligent computer-aided design and computer-aided orthopaedic surgery. From 1994 to 1995, he was the project leader of a U.S. government funded joint project on computer-integrated revision total hip replacement surgery. In 1996, he founded the Computer-Aided Surgery and Medical Image Processing Laboratory, which he heads. Prof. Joskowicz is a member of the editorial boards of *Artificial Intelligence in Engineering*, *Annals of Mathematics and Artificial Intelligence*, the *Journal of Computer-Aided Surgery*, and *Medical Image Analysis*. He has published extensively in these areas and has served on numerous related program committees.

Chapter 10

SYSTEMATIC CHEMICAL PROCESS SYNTHESIS

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Keywords: Chemical Process Synthesis

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Jeffrey J. Siirola received a Ph.D. in chemical engineering from the University of Wisconsin-Madison in 1970 for early work in the area of computer-aided chemical process invention. The work became part of the basis for the textbook *Process Synthesis* co-authored with Dale F. Rudd and Gary J. Powers.

Dr. Siirola joined Eastman Chemical Company in Kingsport, Tennessee in 1972. His assignments have involved the development and application of process synthesis methods and tools for inventing better industrial chemical processes; computer-aided process design, simulation, optimization, and economic evaluation; resource conservation and recovery; and chemical technology assessment. Dr. Siirola is currently a Technology Fellow responsible for an integrated process chemistry and conceptual process engineering laboratory in the Chemicals Research Division.

Dr. Siirola is a member of the National Academy of Engineering and serves on the NRC Board on Chemical Science and Technology. He is a director and fellow of the American Institute of Chemical Engineers and has been particularly active in the Computing and Systems Technology Division. He serves on the Engineering Accreditation Commission of the Accreditation Board for Engineering and Technology, and is an industrial trustee and former president of the Computer Aids for Chemical Engineering Education (CACHE) Corporation.

Chapter 11

SYNTHESIS OF ANALOG AND MIXED-SIGNAL INTEGRATED ELECTRONIC CIRCUITS

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Abstract This survey reviews recent advances in the state of the art for computer-aided synthesis tools for analog and mixed-signal (*i.e.*, jointly analog *and* digital) integrated circuits (ICs). Analog blocks typically constitute only a small fraction of the components on mixed-signal ICs and emerging system-on-chip (SoC) designs. However, the increasing level of integration available in silicon technology, and the growing requirement for digital systems that communicate with the continuous-valued external world are fueling new demands for practical synthesis techniques for these analog circuits. This chapter describes the motivation and evolution of these tools, and reviews progress to date on analog circuit and layout synthesis. The chapter summarizes the problems for which viable solutions are emerging, and those which are still unsolved.

Keywords: VLSI Synthesis, Analog CAD, Systems-on-a-Chip, SoC

Georges G. E. Gielen received the M.Sc. and Ph.D. degrees in Electrical Engineering from the Katholieke Universiteit Leuven, Belgium, in 1986 and 1990, respectively. From 1986 to 1990, he was appointed as a research assistant by the Belgian National Fund of Scientific Research for carrying out his Ph.D. research in the ESAT-MICAS laboratory of the Katholieke Universiteit Leuven. In 1990, he was appointed as a postdoctoral research assistant and visiting lecturer at the department of Electrical Engineering and Computer Science of the University of California, Berkeley, U.S.A. From 1991 to 1993, he was a postdoctoral research assistant of the Belgian National Fund of Scientific Research at the ESAT-MICAS laboratory of the Katholieke Universiteit Leuven. In 1993, he was appointed as a tenure research associate of the Belgian National Fund of Scientific Research and at the same time as an assistant professor at the Katholieke Universiteit Leuven. In 1995 he promoted to associate professor at the same university. In 2000 he was appointed full-time professor at the Katholieke Universiteit Leuven.

His research interests are in the design of analog and mixed-signal integrated circuits, and especially in analog and mixed-signal CAD tools and design automation (modeling, simulation and symbolic analysis, analog synthesis, analog layout generation, analog and mixed-signal testing). He is coordinator or partner of several (industrial) research projects in this area. He has authored or coauthored two books and more than 100 papers in edited books, international journals and conference proceedings. He regularly is a member of the Program Committees of international conferences (ICCAD, DATE, CICC, *etc.*). He is an Associate Editor of the IEEE Transactions on Circuits and Systems, part II, and is a member of the Editorial Board of the Kluwer international journal on Analog Integrated Circuits and Signal Processing. He is a senior member of IEEE, and a member of the Board of Governors of the IEEE Circuits and Systems (CAS) Society, and the Chairman of the IEEE Benelux CAS Chapter. He was the 1997 Laureate of the Belgian Royal Academy of Sciences, Literature and Arts, in the category of engineering sciences. He also received the 1995 Best Paper award of the John Wiley international journal on Circuit Theory and Applications.

Rob A. Rutenbar received the Ph.D. degree from the University of Michigan in 1984, and subsequently joined the faculty of Carnegie Mellon University. He is currently Professor of Electrical and Computer Engineering, and (by courtesy) of Computer Science. From 1993 to 1998 he was Director of the CMU Center for Electronic Design Automation. He is a cofounder of Neolinear, Inc., and also serves as its Chief Scientist.

His research interests focus on circuit and layout synthesis algorithms for mixed-signal ICs, for high-speed digital systems, and for field programmable gate arrays. In 1987, Dr. Rutenbar received a Presidential Young Investigator Award from the National Science Foundation. He has won Best/Distinguished paper awards from the ACM/IEEE Design Automation Conference (1987) the ACM/IEEE International Conference on CAD (1991), and the SRC TECHCON Conference (1993, 2000). He has served on the program committees for the International Conference on CAD, the Design Automation Conference, and the ACM International Symposium on Physical Design. He was General Chair of the 1996 ICCAD; he has also served on the Editorial Board of IEEE Spectrum. He chaired the Analog Technical Advisory Board for Cadence Design Systems (the world's largest integrated circuits CAD software company) from 1992 through 1996. He is a Fellow of the IEEE and a member of the ACM and Eta Kappa Nu.

Chapter 12

MECHANICAL DESIGN COMPILERS

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Abstract

Keywords: Compiler, Design, Synthesis

Allen C. Ward has had successful careers as a machine designer, soldier, university researcher in design theory and automation, and consultant of development practices. Some (including Jim Womack, author of *The Machine that Changed the World*) regard him as the leading U.S. authority on Toyota's product development systmes. Selected recent clients include: Delphi Energy and Engine Management Systems (as lead consultant in a transformation of their product development process), the National Center for Manufacturing Sciences (as lead consultant on a project to find new development paradigms), Knowledge Technology International, (artificial intelligence systems for mechanical design), Schindler Elevator (innovative elevator designs), Photonics (design and build of production machinery), and the Industrial Technology Institute (design team support software). He is a member of executive committee of the National Academies of Science and Engineering's Board on Army Science and Technology, a leading U.S. expert on the "Lean Product Development", and author of *Learning to See in Product Development*, forthcoming from the Lean Enterprise Institute.

As a professor in the University of Michigan's Mechanical Engineering department from June 1989 to June 1996 his studies of Toyota's development processes earned the Shingo Prize. Both this work and his work in design automation earned "best paper" honors from the ASME Design Theory and Methodology conference.

Chapter 13

SCIENTIFIC DISCOVERY AND INVENTIVE ENGINEERING DESIGN

Cognitive and Computational Similarities

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Abstract Cognitive and computational models of discovery and invention are examined, highlighting their similarities at the process level. The basis for this exploration begins with an examination of historic case studies of famous designers and scientists. The chapter motivates cross-fertilization of the two computational fields.

Keywords: Scientific Discovery, Inventive Design, Cognition, Computation

Jonathan Cagan is a Professor of Mechanical Engineering at Carnegie Mellon University, with appointments in the School of Design, Biomedical and Health Engineering and Computer Science. His research, teaching, and consulting are in the area of design theory, methodology, automation, and practice. Dr. Cagan has served as Associate Technical Editor of the ASME Journal of Mechanical Design in the area of Design Theory and Methodology (DTM); he serves on the editorial board of Research in Engineering Design; he has served as the Chair of the ASME DTM committee and of its annual conference. Dr. Cagan held the Ladd Development Chair and is a recent recipient of the Dowd Fellowship in the College of Engineering at CMU. He received his B.S. in 1983 and M.S. in 1985 from the University of Rochester, and his Ph.D. in 1990 from the University of California at Berkeley, all in Mechanical Engineering. Dr. Cagan is a Fellow of the ASME. Prof. Cagan is the recipient of the National Science Foundation's NYI Award and the Society of Automotive Engineer's Ralph R. Teetor Award for Education. He is a member of the Phi Beta Kappa, Tau Beta Pi, and Sigma Xi National Honor Societies, and the ASME, IDSA, AAAI, SAE, and ASEE Professional Societies. Dr. Cagan is a registered Professional Engineer.

Kenneth Kotovksy is a Professor in the Psychology Department at Carnegie Mellon University. He also directs the undergraduate program in psychology at CMU where he has been on the faculty since 1988. He holds a B.S. from M.I.T. and an M.S. and Ph.D. (Psychology) from CMU. His research is focused on cognition, and in particular the cognitive processes involved in problem solving. He uses empirical and computer simulation methodologies to study problem solving. Some of the issues his work has focused on include factors that influence problem difficulty, the early stages of the acquisition of expertise and how the representation of problems influences the above. He is also particularly interested in the processes involved in design as well as the role played by non-conscious processes in a variety of problem-solving behaviors and situations. He has been awarded the Karl Taylor Compton Prize at M.I.T. and the University Undergraduate advising award at CMU. He is a member of the American Psychological Association, the American Psychological Society, the Cognitive Science Society and the Association for the Scientific Study of Consciousness.

Herbert A. Simon is a the Richard King Mellon Professor of Computer Science and Psychology at Carnegie Mellon University, Pittsburgh, Pennsylvania, U.S.A.

His research has ranged from computer science to psychology, administration, and economics, and philosophy. The thread of continuity through all his work has been his interest in human decision-making and problem-solving processes, and the implications of these processes for social institutions. For more than 40 years, he has been making extensive use of the computer as a tool for both simulating human thinking and augmenting it with artificial intelligence.

Born in 1916 in Milwaukee, Wisconsin, Simon was educated in political science at the University of Chicago (B.A., 1936, Ph.D., 1943). He has held research and faculty positions at the University of California (Berkeley), Illinois Institute of Technology, and since 1949, Carnegie Mellon University, where he is Richard King Mellon University Professor of Computer Science and Psychology, and a member also of the Departments of Philosophy and of Social and Decision Sciences, and the Graduate School of Industrial Administration.

In 1978, he received the Alfred Nobel Memorial Prize in Economic Sciences, and in 1986 the National Medal of Science; in 1969, the Distinguished Scientific Contribution Award of the American Psychological Association, in 1975 the A.M. Turing Award of the Association for

Computing Machinery (with Allen Newell), in 1988, the John von Neumann Theory Prize of ORSA/TIMS, and in 1995, the Research Excellence Award of the International Joint Conference on Artificial Intelligence.

Simon's books include Administrative Behavior, Human Problem Solving, jointly with Allen Newell, The Sciences of the Artificial, Scientific Discovery, with Pat Langley, Gary Bradshaw, and Jan Zytkow, three volumes of his collected economics papers (Models of Bounded Rationality), two volumes of collected psychology papers (Models of Thought), a volume of papers on philosophy of science (Models of Discovery), and his autobiography, Models of My Life.