Chapter 1

INTRODUCTION

1.0 CAM-FOLLOWER SYSTEMS

Cam-follower systems are everywhere. They are used in a wide variety of devices and machines. It is actually difficult to get through a normal day in an industrialized society without using one or even many of these systems, or using products that were made by cam-driven machines. If you shave, the razor you use was probably made with a cam-driven machine. When you turn the timer knob on the dishwasher or washing machine, you are setting cams that will slowly rotate to activate the wash cycle's events. When you drive to work, cams get you there. Perhaps the most common application for cams is valve actuation in internal combustion (IC) engines—the typical IC engine has two or more cams per cylinder. Many sewing machines use cams to obtain patterned stitching. If you go to a health club, you use cams to actuate many of the weight training machines. Figure 1-1 shows some examples of common cams.

This book will present practical information as well as the mathematical foundation needed to properly design and manufacture cams for a variety of applications, principally those that involve high speed operation and the need for accuracy, precision, and repeatability. Cams of this variety are used extensively in vehicles and in machinery of all types. Automated production machinery, such as screw machines, spring winders, and assembly machines, all rely heavily on cam-follower systems for their operation. The cam-follower is an extremely useful mechanical device, without which the machine designer's tasks would be much more difficult to accomplish.*

1.1 FUNDAMENTALS

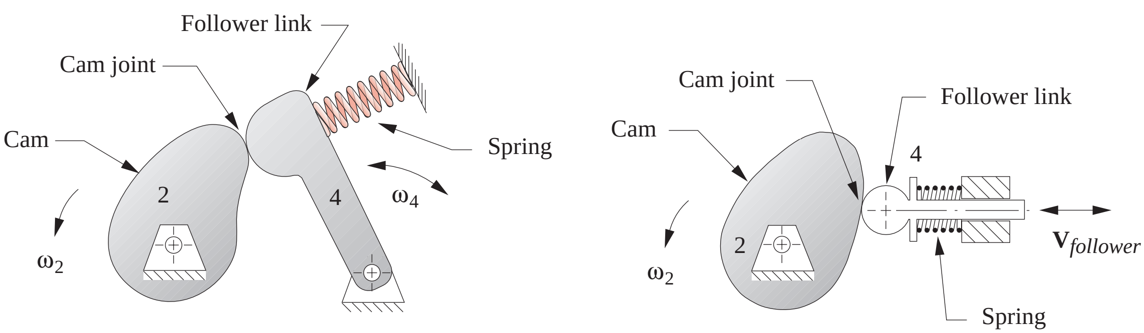
Figure 1-2 shows two simple examples of cams and followers. The cam is a specially shaped piece of metal or other material arranged to move the follower in a controlled fashion. The follower's motion may be either rotation or translation. Figure 1-2a shows a rotating cam driving an oscillating (rotating or swinging) follower. Figure 1-2b shows a rotating cam driving a translating (sliding) follower. In these examples, a spring is used to maintain contact between cam and follower. This is referred to as a *force closed* cam joint, meaning that an external force is needed to keep them together. Just as you cannot push on a rope, you cannot pull on a force-closed cam joint.

* As Neklutin said,[4] *Cams can be rightfully considered as a universally useful mechanism. They have decided advantages over all other mechanisms where a stroke starts from a dwell and ends at a dwell, especially for intermittent motion.*

FIGURE 1-1

Examples of cams (Courtesy of Matrix Tool & Machine, Inc, Mentor, OH)

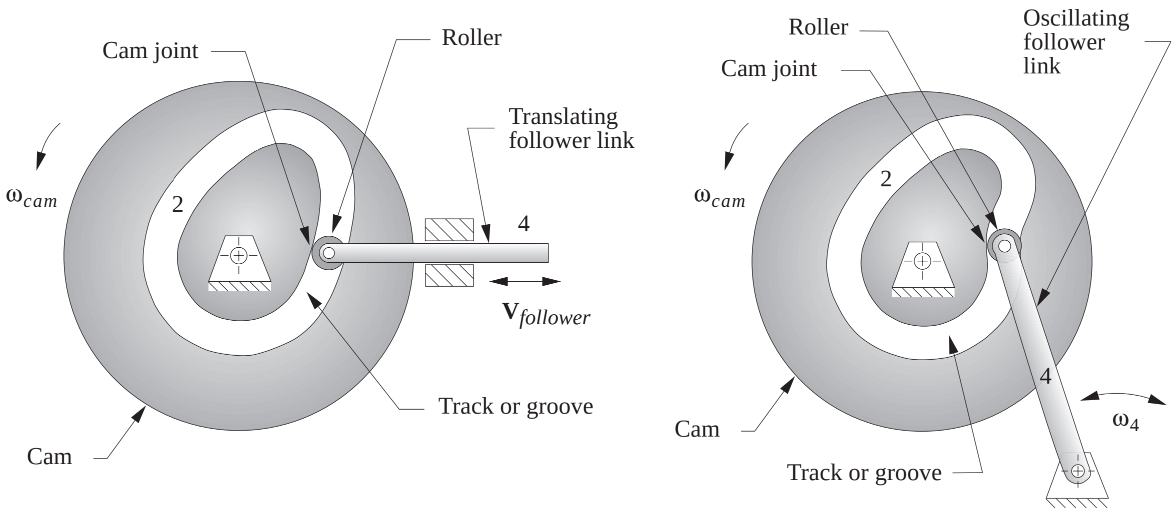
Figure 1-3 shows an alternate arrangement to connect the follower to the cam that does not need a spring. A track or groove in the cam traps the roller follower and now can both push and "pull"—actually it just pushes in both directions. This is called a *form closed* joint as the cam is "formed" around the follower, capturing it by geometry. This type of cam, when used for valve actuation in engines, is also known as **desmodromic**,



(*a*) An oscillating cam-follower (*b*) A translating cam-follower

FIGURE 1-2

Force-closed cam-follower systems



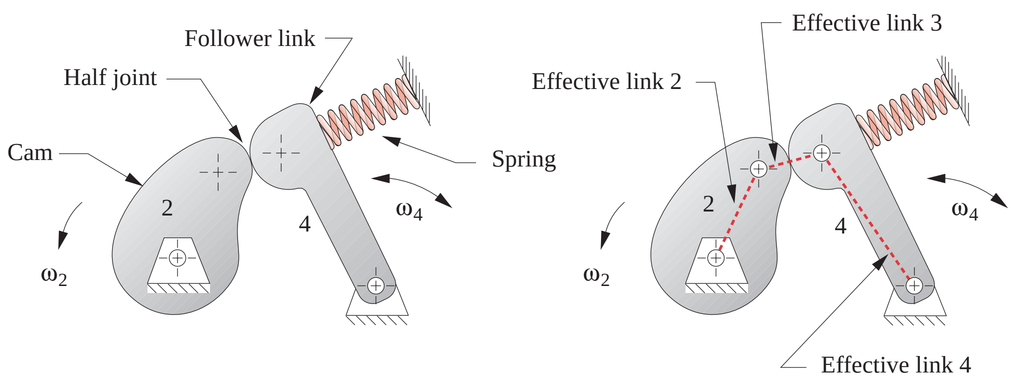
(a) Form-closed cam with translating follower link (b) Form-closed cam with oscillating follower link

FIGURE 1-3

Form-closed cam-follower systems

from the French word *desmodromique* meaning *to force to follow a contour*.[1] Both form- and force-closed cams are used extensively in machinery. Their pros and cons will be explored throughout the book.

Kinematically, cam-follower systems are fourbar linkages with one degree of freedom.[2] Figure 1-4a shows the cam-follower of Figure 1-2a; Figure 1-4b then shows the effective linkage that is (for one instantaneous position) kinematically equivalent to the



(a) An oscillating cam-follower (b) Its effective fourbar linkage equivalent

FIGURE 1-4

Effective linkage of a cam-follower

cam-follower shown in Figure 1-4a. The cam and oscillating arm follower of Figure 1-4a is equivalent to a particular pin-jointed *fourbar crank-rocker* linkage that will change at each position. It is shown in Figure 1-4b for only one position. The cam and translating follower of Figure 1-2b is also equivalent to a particular pin-jointed *fourbar slider-crank* linkage that changes its geometry at each position.

This "equivalent linkage" then has different geometry (link lengths) for each cam-follower position. The lengths of the effective links are determined by the instantaneous locations of the centers of curvature of the cam and follower as shown in Figure 1-4. This is the principal advantage of the cam-follower compared to a "pure" linkage, i.e., it is, in effect, a variable-length linkage that provides a greater degree of motion control than would one with fixed link lengths. It is this difference that makes the cam-follower such a flexible and useful *function generator*. We can specify virtually any output function we desire and create a curved surface on the cam to generate a good approximation to that function in the motion of the follower. The velocities and accelerations of the cam-follower system can be found by analyzing the behavior of the effective linkage for any position. A proof of this can be found in McPhate.[2]

However, in engineering, all advantages come with concomitant disadvantages and the cam-follower system has many such trade-offs. For example, compared to linkages, cams are more compact and easier to design for a specific output function, but they are much more difficult and expensive to make than a linkage. Both positive and negative aspects of cam-follower systems will be explored in the ensuing chapters. For more information on linkage design and kinematics, see Norton.[3]

1.2 TERMINOLOGY

Cam-follower systems can be classified in several ways: by *type of follower motion*, by *type of joint closure*, by *type of follower*, by *type of cam*, by *type of motion constraints*, or by *type of motion program*.

Type of Follower Motion

Figure 1-2a (p. 2) shows a system with an oscillating (rotating or swinging) follower. All three terms are used interchangeably. Figure 1-2b (p. 2) shows a translating (sliding) follower. The choice between these two types of cam-follower is usually determined by the type of output motion desired. If true rectilinear translation is required, then the translating follower is needed. If a pure rotation output is needed, then the oscillator is the obvious choice. There are advantages to each of these approaches separate from their motion characteristics. These will be discussed in Chapters 7 and 18.

Type of Joint Closure

Portions of this chapter were adapted from R. L. Norton, *Design of Machinery* 2ed. McGraw-Hill, 2001, with permission.

Force and form closure were introduced earlier. **Force closure**, as shown in Figure 1-2, *requires an external force to be applied to the joint* in order to keep the two links, cam and follower, physically in contact. This force is usually provided by a spring or sometimes by an air cylinder. This force, defined as positive in a direction that closes the joint, cannot be allowed to become negative. If it does, the links lose contact because a *force-closed joint can only push, not pull*. **Form closure**, as shown in Figure 1-3, *closes the joint by geometry*. No external force is required. There are really two cam surfaces in this arrangement, one surface on each side of the follower. Each surface pushes, in its turn, to drive the follower in both directions.

Figure 1-3a and b shows track or groove cams that capture a single follower in the groove and both push and pull on the follower link. Figure 1-5 shows another variety of form-closed cam-follower arrangement, called **conjugate cams**. There are two cams fixed on a common shaft that are mathematical conjugates of one another. Two roller followers, attached to a common arm, are each pushed in opposite directions by one conjugate cam. The conjugate nature of the two cam surfaces provides that the distance across the two cam surfaces between rollers remains essentially constant. When form-closed (or conjugate) cams are used in automobile or motorcycle engine valve trains, they are called **desmodromic** cams.[1] The advantages and disadvantages to both force- and form-closed arrangements will be discussed in later chapters.

Type of Follower

Follower, in this context, refers only to that part of the follower link which contacts the cam. Figure 1-6 shows three common arrangements, **flat-faced**, **mushroom** (curved), and **roller**. The roller follower has the advantage of lower (rolling) friction than the sliding contact of the other two, but can be more expensive. **Flat-faced followers** can package smaller than roller followers for some cam designs; they are often favored for that reason as well as cost for some automotive valve trains. Many modern automotive engine valve trains now use roller followers for their lower friction.

Roller followers are commonly used in production machinery where their ease of replacement and availability from bearing manufacturers' stock in any quantities are ad-

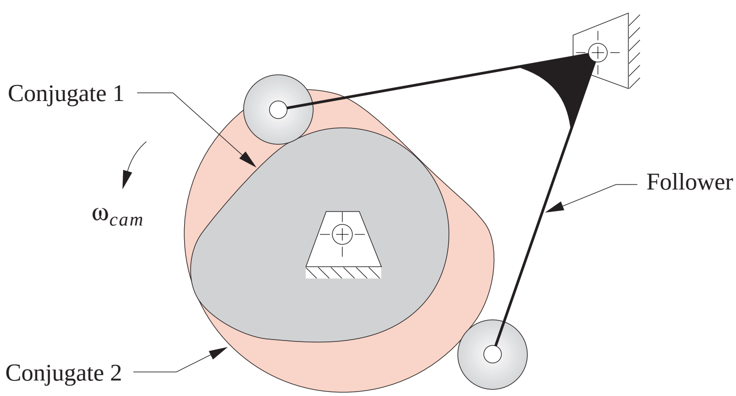


FIGURE 1-5

A conjugate cam pair

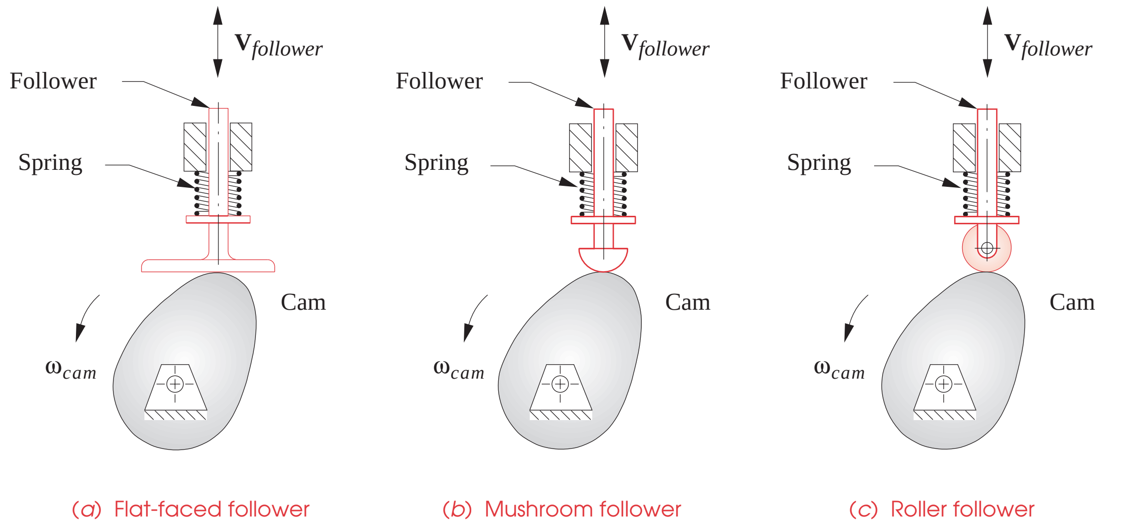


FIGURE 1-6
Three common types of cam followers

vantages. Grooved or track cams require roller followers. Roller followers are essentially ball or roller bearings with customized mounting details. Figure 1-7 shows two common types of commercial roller followers. Flat-faced or **mushroom followers** are usually custom designed and manufactured for each application. For high-volume applications such as automobile engines, the quantities are high enough to warrant a custom-designed follower.

Type of Cam

The direction of the follower's motion relative to the axis of rotation of the cam determines whether it is a **radial** or **axial** cam. All of the cams shown in Figures 1-2 to 1-6

FIGURE 1-7
Commercial roller followers (Courtesy of McGill Manufacturing Co., South Bend, IN)

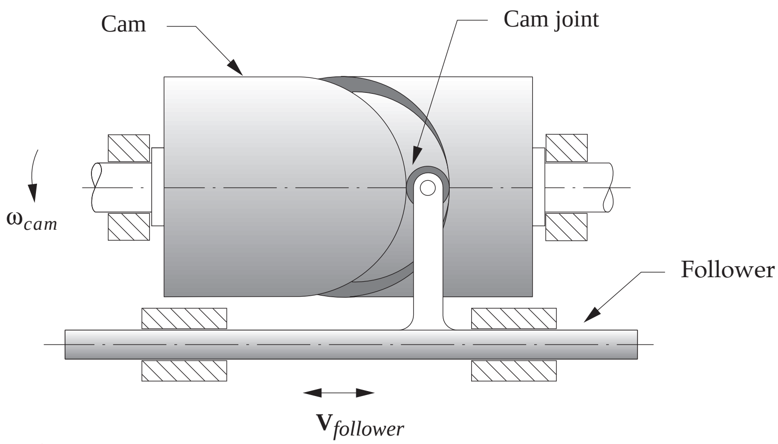


FIGURE 1-8

Axial, cylindrical, or barrel cam with form-closed, translating follower

are radial cams because the follower motion is generally in a radial direction. Open (force-closed) **radial cams** are also called **plate cams**.

Figure 1-8 shows an **axial cam** whose follower moves parallel to the axis of cam rotation. This arrangement is also called a **face** cam if open (force-closed) and a **cylindrical** or **barrel** cam if grooved or ribbed (form-closed).*

Figure 1-9 shows a selection of cams of various types. Clockwise from the lower left, they are: an open axial or face cam (force-closed); an axial grooved (track) cam

FIGURE 1-9

Commercial cams of various types (Courtesy of The Ferguson Co. St. Louis, MO)

* We will refer to this type of cam as a barrel cam throughout the book.

(form-closed) with external gear; an open radial or plate cam (force-closed); a ribbed axial cam (form-closed); and an axial grooved (barrel) cam.

A **three-dimensional cam** or **camoid** (Figure 1-10) is a combination of radial and axial cams. It is a two-degree-of-freedom system. The two inputs are rotation of the cam about its axis and translation of the cam along its axis. The follower motion is a function of both inputs. The follower tracks along a different portion of the cam depending on the axial input.

Type of Motion Constraints

There are two general categories of motion constraint, **critical extreme position** (CEP—also called endpoint specification) and **critical path motion** (CPM). **Critical extreme position** refers to the case in which the design specifications define only the start and finish positions of the follower (i.e., the extreme positions) but do not specify any constraints on the path motion between those extreme positions.* This case is the easier one to design as one has complete freedom to choose the cam functions that control the motion between the extremes. **Critical path motion** is a more constrained problem than CEP because the path motion and/or one or more of its derivatives are defined over all or part of the interval of motion. This requires the generation of a particular function to match the given constraints. It may only be possible to create an approximation of the specified function and still maintain suitable dynamic behavior.

Type of Motion Program

The motion programs **rise-fall** (RF), **rise-fall-dwell** (RFD), and **rise-dwell-fall-dwell** (RDFD)[4] all refer mainly to the CEP case of motion constraint. In effect they define how many dwells are present in the full cycle of motion, either none (RF), one (RFD), or more than one (RDFD). **Dwells,** defined as *no output motion for a specified period of input motion*, are an important feature of cam-follower systems.

FIGURE 1-10

A three-dimensional cam or camoid (Courtesy of Matrix Tool & Machine, Inc, Mentor, OH)

* In some applications it may be desireable to also maximize the area under the curve between the endpoints. An example is the displacement function for an automotive valve motion where area under the curve relates to air flow past the open valve. Such a constraint will affect the choice of follower motion function. Such considerations will be addressed in later chapters after spline functions are introduced.

A cam-follower is the mechanism of choice whenever a dwell is required as it is very easy to create precise dwells in cam-follower mechanisms. Pure linkages can, at best, only provide an approximate dwell. Single- or double-dwell linkages tend to be quite large for their output motion and are somewhat difficult to design. In general, cam-follower systems tend to be more compact than linkages for the same output motion. For the RF case (no dwell), a pin-jointed fourbar crank-rocker linkage is often a superior solution and should be considered before designing a cam-follower solution. See Norton.[3]

A cam-follower is an obvious choice for the RFD and RDFD cases. However, each of these two cases has its own set of constraints on the behavior of the cam functions at the interfaces between the segments that control the rise, fall, and dwell(s). The kinematic constraints that drive the choice of cam profile functions are different for the RFD and RDFD cases. Adding more dwells than two does not change the character of the kinematic constraints from that of the RDFD case. So, a multi-dwell cam is kinematically similar to a double-dwell cam, but both are different than a single-dwell cam in terms of the type of motion program needed. In general, we must match the **boundary conditions** (BC) of the follower displacement functions and their derivatives at all interfaces between the segments of the cam. This topic will be discussed in the next chapter.

1.3 APPLICATIONS

Figure 1-11 shows a cam-follower system used in an automotive valve actuation application. This is an overhead camshaft engine. The camshaft operates against an oscillating follower arm that, in turn, opens the valve. The cam joint is force-closed by the valve spring. The camshaft is typically driven from the engine's crankshaft by gears, chain, or

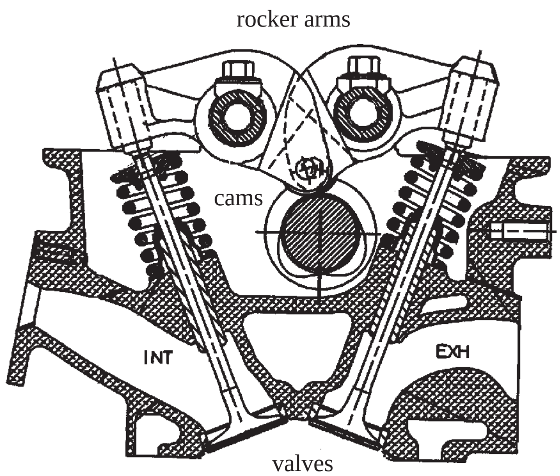


FIGURE 1-11

An overhead camshaft automotive valve train. (Courtesy of DaimlerChrysler Corp., Auburn Hills, MI) toothed belt at a 1:2 reduction. Maximum camshaft speeds in these applications can range from about 2 500 rpm in large automobile engines to over 10 000 rpm in motorcycle racing engines.

Figure 1-12 shows a typical automated assembly machine cam-follower train. Two cams are shown, each of which drives a linkage that actuates tooling in one of several assembly stations along a conveyor line. The tooling will insert a part, crimp a fastener, or do some other operation on the product that is being automatically assembled as it is carried along on the conveyor. A machine of this type may have several dozen of these cam-follower trains arrayed along one or more large camshafts that run the length of the machine (ten meters or more). They may have a mix of force- and form-closed followers. Those shown in the figure are force-closed with air cylinders acting as springs. Maximum camshaft speeds in these applications typically range from about 100 to 1000 rpm.

1.4 TIMING DIAGRAMS

When a machine such as an internal combustion (IC) engine, or an assembly machine that requires several operations, is being designed, one of the first tasks is to define its timing diagram. The timing diagram specifies the relative phasing of all the related events within the machine's cycle. For an IC engine, the timing diagram defines the duration of each cylinder's valve openings and their phasing relative to some rotational reference (or fiduciary) such as top dead center (TDC). For an assembly machine, it defines the start and end points of all motions relevant to the assembly task and relates their phasing to a rotational reference. The cam designs will all flow from this timing diagram. Figure 1-13 shows a timing diagram for a subset of the tooling motions on a typical automated assembly machine.

1.5 CAM DESIGN SOFTWARE

Cam design requires specialized software to be properly done. This book contains a CD-ROM that contains a demonstration version of the program DYNACAM for Windows. Most of the examples in the book were calculated with this program. Instructions for its use are in Appendix A. Data files for all the worked examples in the book are also on the CD-ROM. These can be opened in program DYNACAM to see more details of their solution. Readers are encouraged to use this program to reinforce their understanding of the concepts presented in the book.

1.6 UNITS

There are several systems of units used in engineering. The most common in the United States are the **U.S. foot-pound-second (fps) system**, the **U.S. inch-pound-second (ips) system**, and the **System International (SI)**. All systems are created from the choice of three of the quantities in the general expression of Newton's second law:

$$F = \frac{m\,l}{t^2} \tag{1.1a}$$

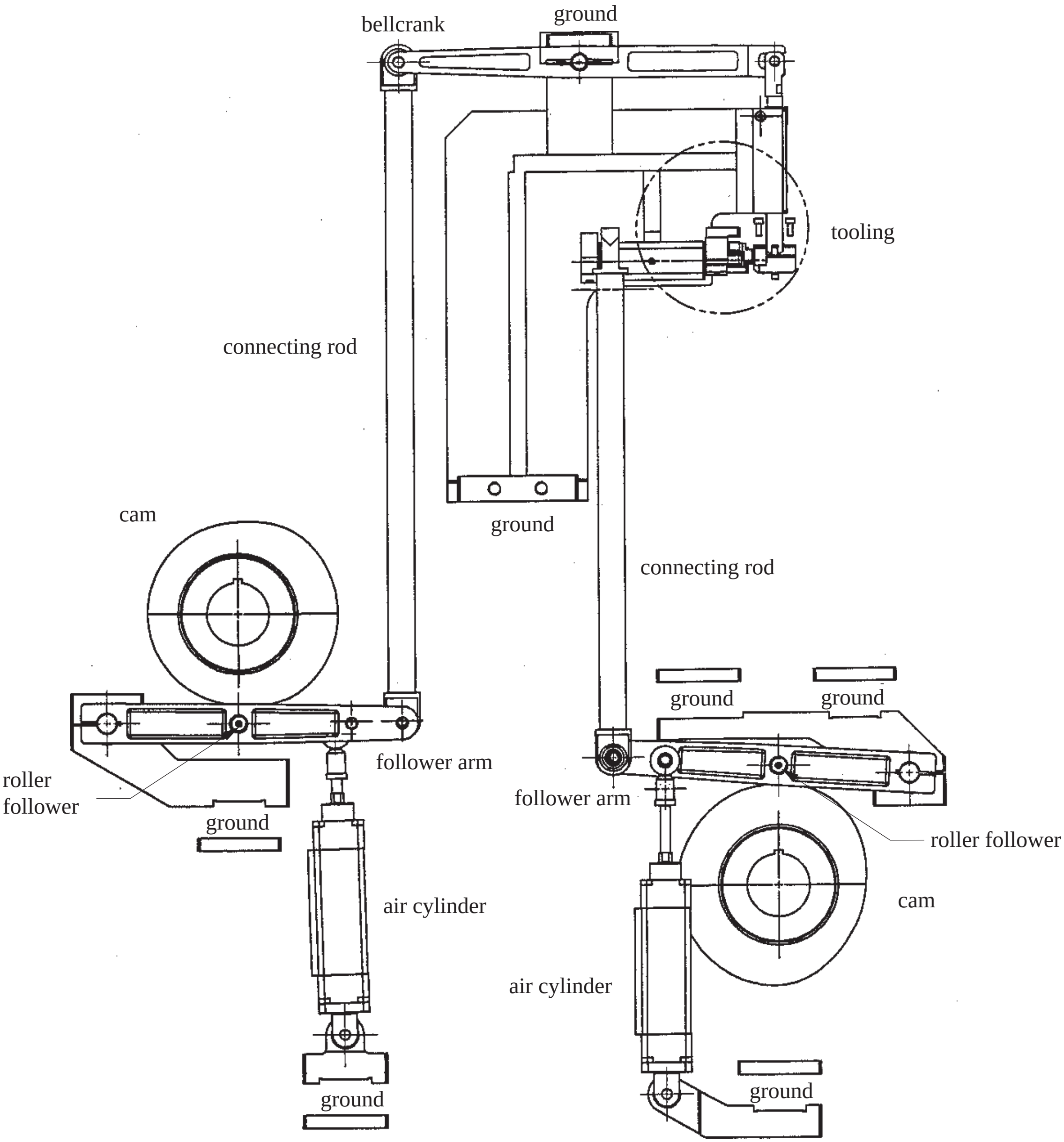


FIGURE 1-12

Cam-follower mechanisms for one station of an automated assembly machine (Courtesy of the Gillette Co., Boston, MA)

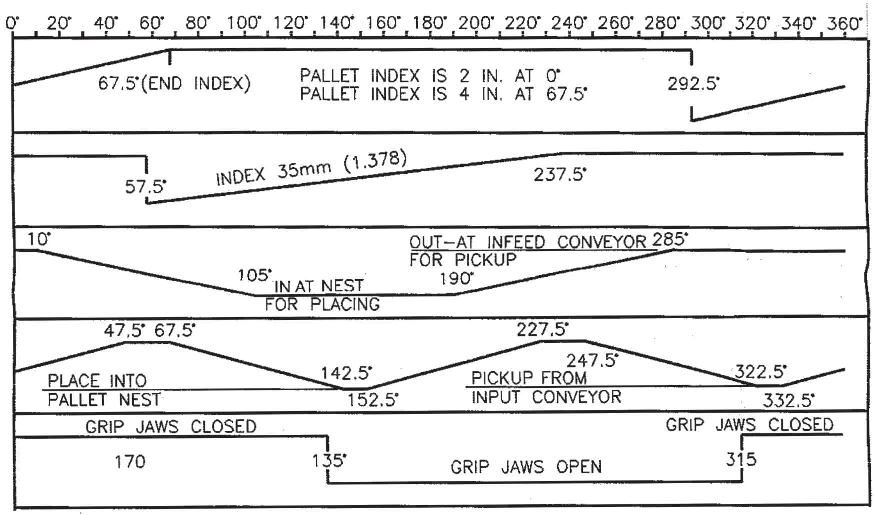


FIGURE 1-13

Partial timing diagram for an automated assembly machine (Courtesy of the Gillette Co., Boston, MA)

where F is force, m is mass, l is length, and t is time. The units for any three of these variables can be chosen and the other is then derived in terms of the chosen units. The three chosen units are called *base units*, and the remaining one is then a *derived unit*.

Most of the confusion that surrounds the conversion of computations between either one of the U.S. systems and the SI system is due to the fact that the SI system uses a different set of base units than the U.S. systems. Both U.S. systems choose ***force***, *length*, and *time* as the base units. Mass is then a derived unit in the U.S. systems, which are referred to as *gravitational systems* because the value of mass is dependent on the local gravitational constant. The SI system chooses ***mass***, *length*, and *time* as the base units and force is the derived unit. SI is then referred to as an *absolute system* since the mass is a base unit whose value is not dependent on local gravity.

The **U.S. foot-pound-second (fps)** system requires that all lengths be measured in feet (ft), forces in pounds (lb), and time in seconds (sec). Mass is then derived from Newton's law as

$$m = \frac{Ft^2}{l} \tag{1.1b}$$

and the units are:

Pounds seconds squared per **foot** (lb-sec^2/ft) = **slugs**

The **U.S. inch-pound-second (ips)** system requires that all lengths be measured in inches (in), forces in pounds (lb), and time in seconds (sec). Mass is still derived from Newton's law, equation 1.1b, but the units are now:

Pounds seconds squared per **inch** (lb-sec^2/in) = **blobs**

This mass unit is not slugs! It is worth twelve slugs or one blob.*

Weight is defined as the force exerted on an object by gravity. Probably the most common units error is to mix up these two unit systems (**fps** and **ips**) when converting weight units (which are pounds force) to mass units. Note that the gravitational acceleration constant (g) on earth at sea level is approximately 32.2 **feet** per second squared which is equivalent to 386 **inches** per second squared. The relationship between mass and weight is:

Mass = weight / gravitational acceleration

$$m = \frac{W}{g} \tag{1.2}$$

It should be obvious that, if you measure all your lengths in **inches** and then use g = 32.2 **feet**/sec^2 to compute mass, you will have an error of a *factor of 12* in your results. This is a serious error, large enough to crash the airplane you designed. Even worse off is one who neglects to convert weight to mass *at all* in his calculations. He will have an error of either 32.2 or 386 in his results. This is enough to sink the ship!

To add even further to the confusion about units is the common use of the unit of **pounds mass** (lb_m). This unit is often used in fluid dynamics and thermodynamics; it comes about through the use of a slightly different form of Newton's equation:

$$F = \frac{ma}{g_c} \tag{1.3}$$

where m = mass in lb_m, a = acceleration and g_c = the gravitational constant.

The value of the **mass** of an object measured in **pounds mass** (lb_m) is *numerically equal* to its **weight** in **pounds force** (lb_f). However, the student *must remember to divide* the value of m in lb_m by g_c when substituting into this form of Newton's equation. Thus the lb_m will be divided either by 32.2 or by 386 when calculating the dynamic force. The result will be the same as when the mass is expressed in either slugs or blobs in the $F = ma$ form of the equation. Remember that in round numbers at sea level on earth:

1 lb_m = 1 lb_f 1 slug = 32.2 lb_f 1 blob = 386 lb_f

The **SI** system requires that lengths be measured in meters (m), mass in kilograms (kg), and time in seconds (sec). This is sometimes also referred to as the **mks** system. Force is derived from Newton's law, equation 1.1b and the units are:

kilogram-meters per second^2 (kg-m/sec^2) = newtons

Thus in the SI system, there are distinct names for mass and force which helps alleviate confusion. When converting between SI and U.S. systems, be alert to the fact that mass converts from kilograms (kg) to either slugs (sl) or blobs (bl), and force converts

* It is unfortunate that the mass unit in the **ips** system has never officially been given a name such as the term ***slug*** used for mass in the **fps** system. The author boldly suggests (with tongue only slightly in cheek) that this unit of mass in the **ips** system be called a *blob* (bl) to distinguish it more clearly from the ***slug*** (sl), and to help avoid some of the common units errors listed above.

Twelve slugs = one blob.

Blob does not sound any sillier than slug, is easy to remember, implies mass, and has a convenient abbreviation (bl) which is an anagram for the abbreviation for pound (lb). Besides, if you have ever seen a garden slug, you know it looks just like a "*little blob.*"

Table 1-1 Variables and Units
Base Units in Boldface – Abbreviations in ( )

Variable	Symbol	ips unit	fps unit	SI unit
Force	F	**pounds (lb)**	**pounds (lb)**	newtons (N)
Length	l	**inches (in)**	**feet (ft)**	**meters (m)**
Time	t	**seconds (sec)**	**seconds (sec)**	**seconds (sec)**
Mass	m	lb–sec^2/in (bl)	lb–sec^2/ft (sl)	**kilograms (kg)**
Weight	W	pounds (lb)	pounds (lb)	newtons (N)
Velocity	v	in/sec	ft/sec	m/sec
Acceleration	a	in/sec^2	ft/sec^2	m/sec^2
Jerk	j	in/sec^3	ft/sec^3	m/sec^3
Angle	θ	degrees (deg)	degrees (deg)	degrees (deg)
Angle	θ	radians (rad)	radians (rad)	radians (rad)
Angular velocity	ω	rad/sec	rad/sec	rad/sec
Angular acceleration	α	rad/sec^2	rad/sec^2	rad/sec^2
Angular jerk	φ	rad/sec^3	rad/sec^3	rad/sec^3
Torque	T	lb–in	lb–ft	N–m
Mass moment of inertia	I	lb–in–sec^2	lb–ft–sec^2	N–m–sec^2
Energy	E	in–lb	ft–lb	joules
Power	P	in–lb / sec	ft–lb / sec	watts
Volume	V	in^3	ft^3	m^3
Weight density	γ	lb/in^3	lb/ft^3	N/m^3
Mass density	ρ	bl/in^3	sl/ft^3	kg/m^3

from newtons (N) to pounds (lb). The gravitational constant (g) in the SI system is approximately 9.81 m/sec^2.

The principal system of units used in this book will be the U.S. **ips** system. Most machine design in the United States is still done in this system. Table 1-1 shows some of the variables used in this text and their units. Table 1-2 shows conversion factors between the U.S. and SI systems.

The reader is cautioned to always check the units in any equation written for a problem solution. If properly written, an equation should cancel all units across the equal sign. If it does not, then you can be *absolutely sure it is* ***incorrect.*** Unfortunately, a unit balance in an equation does not guarantee that it is correct, as many other errors are possible.

Table 1-2 Selected Units Conversion Factors

Multiply this	by	this	to get	this
acceleration				
in/sec^2	x	0.0254	=	m/sec^2
ft/sec^2	x	12	=	in/sec^2
angles				
radian	x	57.2958	=	deg
area				
in^2	x	645.16	=	mm^2
ft^2	x	144	=	in^2
area moment of inertia				
in^4	x	416 231	=	mm^4
in^4	x	4.162E–07	=	m^4
m^4	x	1.0E+12	=	mm^4
m^4	x	1.0E+08	=	cm^4
ft^4	x	20 736	=	in^4
density				
lb/in^3	x	27.6805	=	g/cc
g/cc	x	0.001	=	g/mm^3
lb/in^3	x	1 728	=	lb/ft^3
kg/m^3	x	1.0E–06	=	g/mm^3
force				
lb	x	4.448	=	N
N	x	1.0E+05	=	dyne
ton (short)	x	2 000	=	lb
length				
in	x	25.4	=	mm
ft	x	12	=	in
mass				
blob	x	386.4	=	lb
slug	x	32.2	=	lb
blob	x	12	=	slug
kg	x	2.205	=	lb
kg	x	9.8083	=	N
kg	x	1 000	=	g
mass moment of inertia				
lb-in-sec^2	x	0.1138	=	N-m-sec^2
moments and energy				
in-lb	x	0.1138	=	N-m
ft-lb	x	12	=	in-lb
N-m	x	8.7873	=	in-lb
N-m	x	0.7323	=	ft-lb
power				
HP	x	550	=	ft-lb/sec
HP	x	33 000	=	ft-lb/min
HP	x	6 600	=	in-lb/sec
HP	x	745.7	=	watts
N-m/sec	x	8.7873	=	in-lb/sec
pressure and stress				
psi	x	6 894.8	=	Pa
psi	x	6.895E-3	=	MPa
psi	x	144	=	psf
kpsi	x	1 000	=	psi
N/m^2	x	1	=	Pa
N/mm^2	x	1	=	MPa
spring rate				
lb/in	x	175.126	=	N/m
lb/ft	x	0.08333	=	lb/in
stress intensity				
MPa-m$^{0.5}$	x	0.909	=	ksi-in$^{0.5}$
velocity				
in/sec	x	0.0254	=	m/sec
ft/sec	x	12	=	in/sec
rad/sec	x	9.5493	=	rpm
volume				
in^3	x	16 387.2	=	mm^3
ft^3	x	1 728	=	in^3
cm^3	x	0.061023	=	in^3
m^3	x	1.0E+9	=	mm^3

1.7 REFERENCES

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