ADHESIVES AND SEALANTS

By strict definition, an adhesive is any substance that fastens or bonds materials to be joined (adherends) by means of surface attachment. The bond durability depends on the strength of the adhesive to the substrate (adhesion) and the strength within the adhesive (cohesion). Besides bonding a joint, an adhesive may serve as a seal against foreign matter. When an adhesive performs both bonding and sealing functions, it is usually referred to as an adhesive sealant. Joining materials with adhesives offers significant benefits compared with mechanical methods of uniting two materials.

Among these benefits are that an adhesive distributes a load over an area rather than concentrating it at a point, resulting in a more even distribution of stresses. The adhesive bonded joint is therefore more resistant to flexural and vibrational stresses than, for example, a bolted, riveted, or welded joint. Another benefit is that an adhesive forms a seal as well as a bond. This seal prevents the corrosion that may occur with dissimilar metals, such as aluminum and magnesium, or mechanically fastened joints, by providing a dielectric insulation between the substrates. An adhesive also joins irregularly shaped surfaces more easily than does a mechanical fastener. Other benefits include negligible weight addition and virtually no change to part dimensions or geometry.

Most adhesives are available in liquids, gels, pastes, and tape forms. The growing variety of adhesives available can make the selection of the proper adhesive or sealant a challenging experience. In addition to the technical requirements of the adhesive, time and costs are also important considerations. Proper choice of an adhesive is based on knowledge of the suitability of the adhesive or sealant for the particular substrates. Appropriate surface preparation, curing parameters, and matching the strength and durability characteristics of the adhesive to its intended use are essential. The performance of an adhesive-bonded joint depends on a wide range of these factors, many of them quite complex. Adhesive suppliers can usually offer essential expertise in the area of appropriate selection.

Adhesives can be classified as structural or nonstructural. In general, an adhesive can be considered structural when it is capable of supporting heavy loads; nonstructural when it cannot support such loads. Many adhesives and sealants, under various brand names, may be available for a particular bonding application. It is always advisable to check the adhesive manufacturers' information before making an adhesive sealant selection. Also, testing under end-use conditions is always suggested to help ensure bonded or sealed joints meet or exceed expected performance requirements.

Though not meant to be all-inclusive, the following information correlates the features of some successful adhesive compositions available in the marketplace.

Bonding Adhesives

Reactive-type bonding adhesives are applied as liquids and react (cure) to solids under appropriate conditions. The cured adhesive is either a thermosetting or thermoplastic polymer. These adhesives are supplied as two-component no-mix, two-component mix, and one-component no-mix types, which are discussed in the following paragraphs.

Two-Component No-Mix Adhesives

Types of Adhesives.—*Anaerobic (Urethane Methacrylate Ester) Structural Adhesives:* Anaerobic structural adhesives are mixtures of acrylic esters that remain liquid when exposed to air but harden when confined between metal substrates. These adhesives can be used for large numbers of industrial purposes where high reliability of bond joints is required. Benefits include: no mixing is required (no pot-life or waste problems), flexible/durable bonds are made that withstand thermal cycling, have excellent resistance to solvents and severe environments, and rapid cure at room temperatures (eliminating expensive ovens). The adhesives are easily dispensed with automatic equipment. An activator is usually required to be present on one surface to initiate the cure for these adhesives. Applications for these adhesives include bonding of metals, magnets (ferrites), glass, thermosetting plastics, ceramics, and stone.

Acrylic Adhesives: Acrylic adhesives are composed of a polyurethane polymer backbone with acrylate end groups. They can be formulated to cure through heat or the use of an activator applied to the substrate surface, but many industrial acrylic adhesives are cured by light. Light-cured adhesives are used in applications where the bond geometry allows light to reach the adhesive and the production rate is high enough to justify the capital expense of a light source. Benefits include: no mixing is required (no pot-life or waste problems); formulations cure (solidify) with activator, heat, or light; the adhesive will bond to a variety of substrates, including metal and most thermoplastics; and tough and durable bonds are produced with a typical resistance to the effects of temperatures up to 180°C. Typical applications include automobile body parts (steel stiffeners), assemblies subjected to paint-baking cycles, speaker magnets to pole plates, and bonding of motor magnets, sheet steel, and many other structural applications. Other applications include bonding glass, sheet metal, magnets (ferrite), thermosetting and thermoplastic plastics, wood, ceramics, and stone.

Two-Component Mix Adhesives

Types of Adhesives.—*Epoxy Adhesives:* Two-component epoxy adhesives are well-established adhesives that offer many benefits in manufacturing. The reactive components of these adhesives are separated prior to use, so they usually have a good shelf life without refrigeration. Polymerization begins upon mixing, and a thermoset polymer is formed. Epoxy adhesives cure to form thermosetting polymers made up of a base side with the polymer resin and a second part containing the catalyst. The main benefit of these systems is that the depth of cure is unlimited. As a result, large volume can be filled for work such as potting, without the cure being limited by the need for access to an external influence such as moisture or light to activate the curing process.

For consistent adhesive performance, it is important that the mix ratio remain constant to eliminate variations in adhesive performance. Epoxies can be handled automatically, but the equipment involves initial and maintenance costs. Alternatively, adhesive components can be mixed by hand. However, this approach involves labor costs and the potential for human error. The major disadvantage of epoxies is that they tend to be very rigid and consequently have low peel strength. This lack of peel strength is less of a problem when bonding metal to metal than it is when bonding flexible substrates such as plastics.

Applications of epoxy adhesives include bonding, potting, and coating of metals, bonding of glass, rigid plastics, ceramics, wood, and stone.

Polyurethane Adhesives: Like epoxies, polyurethane adhesives are available as two-part systems or as one-component frozen premixes. They are also available as one-part moisture-cured systems. Polyurethane adhesives can provide a wide variety of physical properties. Their flexibility is greater than that of most epoxies. Coupled with the high cohesive strength, this flexibility provides a tough polymer able to achieve better peel strength and lower flexural modulus than most epoxy systems. This superior peel resistance allows use of polyurethanes in applications that require high flexibility. Polyurethanes bond very well to a variety of substrates, though a primer may be needed to prepare the substrate surface. These primers are moisture-reactive and require several hours to react sufficiently for the parts to be used. Such a time requirement may cause a production bottleneck if the bond-strength requirements are such that a primer is needed.

Applications for polyurethane adhesives include bonding of metals, glass, rubber, thermosetting and thermoplastic plastics, and wood.

One-Component No-Mix Adhesives

Types of Adhesives.—*Light-Curable Adhesives:* Light-curing systems use a unique curing mechanism. The adhesives contain photoinitiators that absorb light energy and dissociate to form radicals. These radicals then initiate the polymerization of the polymers, oligomers, and monomers in the adhesive. The photoinitiator acts as a chemical solar cell, converting the light energy into chemical energy for the curing process. Typically, these systems are formulated for use with ultraviolet light sources. However, newer products have been formulated for use with visible light sources.

One of the biggest benefits that light-curing adhesives offer to the manufacturer is the elimination of the work time to work-in-progress trade-off, which is embodied in most adhesive systems. With light-curing systems, the user can take as much time as needed to position the part without fear of the adhesive curing. Upon exposure to the appropriate light source, the adhesive then can be fully cured in less than 1 minute, minimizing the costs associated with work in progress. Adhesives that utilize light as the curing mechanism are often one-part systems with good shelf life, which makes them even more attractive for manufacturing use.

Applications for light-curable adhesives include bonding of glass, and glass to metal, tacking of wires, surface coating, thin-film encapsulation, clear substrate bonding, and potting of components,

Cyanoacrylate Adhesives (Instant Adhesives): Cyanoacrylates or instant adhesives are often called Superglue™. Cyanoacrylates are one-part adhesives that cure rapidly, as a result of the presence of surface moisture, to form high-strength bonds, when confined between two substrates. Cyanoacrylates have excellent adhesion to many substrates, including most plastics and they achieve fixture strength in seconds and full strength within 24 hours. These qualities make cyanoacrylates suitable for use in automated production environments. They are available in viscosities ranging from water-thin liquids to thixotropic gels.

Because cyanoacrylates are a relatively mature adhesive family, a wide variety of specialty formulations is now available to help the user address difficult assembly problems. One of the best examples is the availability of polyolefin primers, which allow users to obtain high bond strengths on difficult-to-bond plastics such as polyethylene and polypropylene. One common drawback of cyanoacrylates is that they form a very rigid polymer matrix, resulting in very low peel strengths. To address this problem, formulations have been developed that are rubber-toughened. Although the rubber toughening improves the peel strength of the system to some extent, peel strength remains a weak point for this system, and, therefore, cyanoacrylates are poor candidates for joint designs that require high peel resistance. In manufacturing environments with low relative humidity, the cure of the cyanoacrylate can be significantly retarded.

This problem can be addressed in one of two ways. One approach is to use accelerators that deposit active species on the surface to initiate the cure of the product. The other approach is to use specialty cyanoacrylate formulations that have been engineered to be surface-insensitive. These formulations can cure rapidly even on dry or slightly acidic surfaces.

Applications for cyanoacrylate adhesives include bonding of thermoplastic and thermosetting plastics, rubber, metals, wood, and leather, also strain relief of wires.

Hot-Melt Adhesives: Hot-melt adhesives are widely used in assembly applications. In general, hot-melt adhesives permit fixturing speeds that are much faster than can be achieved with water- or solvent-based adhesives. Usually supplied in solid form, hot-melt adhesives liquify when exposed to elevated temperatures. After application, they cool quickly, solidifying and forming a bond between two mating substrates. Hot-melt adhesives have been used successfully for a wide variety of adherends and can greatly reduce both the need for clamping and the length of time for curing. Some drawbacks with hot-melt adhesives are their tendency to string during dispensing and relatively low-temperature resistance.

Applications for hot-melt adhesives are bonding of fabrics, wood, paper, plastics, and cardboard.

Rubber-Based Solvent Cements: Rubber-based solvent cements are adhesives made by combining one or more rubbers or elastomers in a solvent. These solutions are further modified with additives to improve the tack or stickiness, the degree of peel strength, flexibility, and the viscosity or body. Rubber-based adhesives are used in a wide variety of applications such as contact adhesive for plastics laminates like counter tops, cabinets, desks, and tables. Solvent-based rubber cements have also been the mainstay of the shoe and leather industry for many years.

Applications for rubber-based solvent cements include bonding of plastics laminates, wood, paper, carpeting, fabrics, and leather.

Moisture-Cured Polyurethane Adhesives: Like heat-curing systems, moisture-cured polyurethanes have the advantage of a very simple curing process. These adhesives start to cure when moisture from the atmosphere diffuses into the adhesive and initiates the polymerization process. In general, these systems will cure when the relative humidity is above 25 per cent, and the rate of cure will increase as the relative humidity increases.

The dependence of these systems on the permeation of moisture through the polymer is the source of their most significant process limitations. As a result of this dependence, depth of cure is limited to between 0.25 and 0.5 in. (6.35 and 12.7 mm). Typical cure times are in the range of 12 to 72 hours. The biggest use for these systems is for windshield bonding in automobile bodies.

Applications for moisture-cured polyurethane adhesives include bonding of metals, glass, rubber, thermosetting and thermoplastic plastics, and wood.

Retaining Compounds

The term *retaining compounds* is used to describe adhesives used in circumferential assemblies joined by inserting one part into the other. In general, retaining compounds are anaerobic adhesives composed of mixtures of acrylic esters that remain liquid when exposed to air but harden when confined between cylindrical machine components. A typical example is a bearing held in an electric motor housing with a retaining compound. The first retaining compounds were launched in 1963, and the reaction among users of bearings was very strong because these retaining compounds enabled buyers of new bearings to salvage worn housings and minimize their scrap rate.

The use of retaining compounds has many benefits, including elimination of bulk needed for high friction forces, ability to produce more accurate assemblies and to augment or replace press fits, increased strength in heavy press fits, and reduction of machining costs. Use of these compounds also helps in dissipating heat through assembly, and eliminating distortion when installing drill bushings, fretting corrosion and backlash in keys and splines, and bearing seizure during operation.

The major advantages of retaining compounds for structural assemblies are that they require less severe machining tolerances and no securing of parts. Components are assembled quickly and cleanly, and they transmit high forces and torques, including dynamic forces. Retaining compounds also seal, insulate, and prevent micromovements so that neither fretting corrosion nor stress corrosion occurs. The adhesive joint can be taken apart easily after heating above 450°F (230°C) for a specified time.

Applications for retaining compounds include mounting of bearings in housings or on shafts, avoiding distortion of precision tooling and machines, mounting of rotors on shafts, inserting drill jig bushings, retaining cylinder linings, holding oil filter tubes in castings, retaining engine-core plugs, restoring accuracy to worn machine tools, and eliminating keys and set screws.

Threadlocking

The term *threadlocker* is used to describe adhesives used in threaded assemblies for locking the threaded fasteners by filling the spaces between the nut and bolt threads with a hard, dense material that prevents loosening. In general, thread-lockers are anaerobic adhesives comprising mixtures of acrylic esters that remain liquid when exposed to air but harden when confined between threaded components. A typical example is a mounting bolt on a motor or a pump. Threadlocker strengths range from very low strength (removable) to high strength (permanent).

It is important that the total length of the thread is coated and that there is no restriction to the curing of the threadlocker material. (Certain oils or cleaning systems can impede or even completely prevent the adhesive from curing by anaerobic reaction.) The liquid threadlocker may be applied by hand or with special dispensing devices. Proper coating (wetting) of a thread is dependent on the size of the thread, the viscosity of the adhesive, and the geometry of the parts. With blind-hole threads, it is essential that the adhesive be applied all the way to the bottom of the threaded hole. The quantity must be such that after assembly, the displaced adhesive fills the whole length of the thread.

Some threadlocking products cured by anaerobic reaction have a positive influence on the coefficient of friction in the thread. The values are comparable with those of oiled bolts. Prestress and installation torque therefore can be defined exactly. This property allows threadlocking products cured by anaerobic reaction to be integrated into automated production lines using existing assembly equipment. The use of thread-lockers has many benefits including ability to lock and seal all popular bolt and nut sizes with all industrial finishes, and to replace mechanical locking devices. The adhesive can seal against most industrial fluids and will lubricate threads so that the proper clamp load is obtained. The materials also provide vibration-resistant joints that require handtool dismantling for servicing, prevent rusting of threads, and cure (solidify) without cracking or shrinking.

The range of applications includes such uses as locking and sealing nuts on hydraulic pistons, screws on vacuum cleaner bell housings, track bolts on bulldozers, hydraulic-line fittings, screws on typewriters, oil-pressure switch assembly, screws on carburetors, rocker nuts, machinery driving keys, and on construction equipment.

Sealants

The primary role of a sealant composition is the prevention of leakage from or access by dust, fluids, and other materials to assembly structures. Acceptable leak rates can range from a slight drip to bubbletight to molecular diffusion through the base materials. Equipment users in the industrial market want trouble-free operation, but it is not always practical to specify zero leak rates. Factors influencing acceptable leak rates are toxicity, product or environmental contamination, combustibility, economics, and personnel considerations. All types of fluid seals perform the same basic function: they seal the process fluid (gas, liquid, or vapor) and keep it where it belongs. A general term for these assembly approaches is gasketing. Many products are being manufactured that are capable of sealing a variety of substrates.

Types of Sealants.—*Anaerobic Formed-in-Place Gasketing Materials:* Mechanical assemblies that require the joining of metal-to-metal flange surfaces have long been designed with prefabricated, precut materials required to seal the imperfect surfaces of the assembly. Numerous gasket materials that have been used to seal these assemblies include paper, cork, asbestos, wood, metals, dressings, and even plastics. Fluid seals are divided into static and dynamic systems, depending on whether or not the parts move in relationship to each other. Flanges are classed as static systems, although they may be moved relative to each other by vibration, temperature, and/or pressure changes, shocks, and impacts.

The term *anaerobic formed-in-place gasketing* is used to describe sealants that are used in flanged assemblies to compensate for surface imperfections of metal-to-metal components by filling the space between the substrates with a flexible, nonrunning material. In general, anaerobic formed-in-place gaskets are sealants made up of mixtures of acrylic esters that remain liquid when exposed to air but harden when confined between components. A typical example is sealing two halves of a split crankcase.

The use of anaerobic formed-in-place gaskets has many benefits, including the ability to seal all surface imperfections, allow true metal-to-metal contact, eliminate compression set and fastener loosening, and add structural strength to assemblies. These gaskets also help improve torque transmission between bolted flange joints, eliminate bolt retorquing needed with conventional gaskets, permit use of smaller fasteners and lighter flanges, and provide for easy disassembly and cleaning.

Applications in which formed-in-place gasketing can be used to produce leakproof joints include pipe flanges, split crankcases, pumps, compressors, power takeoff covers, and axle covers. These types of gaskets may also be used for repairing damaged conventional gaskets and for coating soft gaskets.

Silicone Rubber Formed-in-Place Gasketing: Another type of formed-in-place gasket uses room-temperature vulcanizing (RTV) silicone rubbers. These materials are one-component sealants that cure on exposure to atmospheric moisture. They have excellent properties for vehicle use such as flexibility, low volatility, good adhesion, and high resistance to most automotive fluids. The materials will also withstand temperatures up to 600°F (320°C) for intermittent operation.

RTV silicones are best suited for fairly thick section (gap) gasketing applications where flange flexing is greatest. In the form of a very thin film, for a rigid metal-to-metal seal, the cured elastomer may abrade and eventually fail under continual flange movement. The RTV silicone rubber does not unitize the assembly, and it requires relatively clean, oil-free surfaces for sufficient adhesion and leakproof seals.

Because of the silicone's basic polymeric structure, RTV silicone elastomers have several inherent characteristics that make them useful in a wide variety of applications. These properties include outstanding thermal stability at temperatures from 400 to 600°F (204 to 320°C), and good low-temperature flexibility at −85 to −165°F (−65 to −115°C). The material forms an instant seal, as is required of all liquid gaskets, and will fill large gaps up to 0.250 in. (6.35 mm) for stamped metal parts and flanges. The rubber also has good stability in ultraviolet light and excellent weathering resistance.

Applications for formed-in-place RTV silicones in the automotive field are valve, camshaft and rocker covers, manual transmission (gearbox) flanges, oil pans, sealing panels, rear axle housings, timing chain covers, and window plates. The materials are also used on oven doors and flues.

Tapered Pipe-thread Sealing

Thread sealants are used to prevent leakage of gases and liquids from pipe joints. All joints of this type are considered to be dynamic because of vibration, changing pressures, or changing temperatures.

Several types of sealants are used on pipe threads including noncuring pipe dopes, which are one of the oldest methods of sealing the spiral leak paths of threaded joints. In general, pipe dopes are pastes made from oils and various fillers. They lubricate joints and jam threads but provide no locking advantage. They also squeeze out under pressure, and have poor solvent resistance. Noncuring pipe dopes are not suitable for use on straight threads.

Another alternative is solvent-drying pipe dopes, which are an older method of sealing tapered threaded joints. These types of sealant offer the advantages of providing lubrication and orifice jamming and they also extrude less easily than noncuring pipe dopes. One disadvantage is that they shrink during cure as the solvents evaporate and fittings must be retorqued to minimize voids. These materials generally lock the threaded joint together by friction. A third type of sealer is the trapped elastomer supplied in the form of a thin tape incorporating polytetrafluorethylene (PTFE). This tape gives a good initial seal and resists chemical attack, and is one of the only materials used for sealing systems that will seal against oxygen gas.

Some other advantages of PTFE are that it acts as a lubricant, allows for high torquing, and has a good resistance to various solvents. Some disadvantages are that it may not provide a true seal between the two threaded surfaces, and it lubricates in the off direction, so it may allow fittings to loosen. In dynamic joints, tape may allow creep, resulting in leakage over time. The lubrication effect may allow overtightening, which can add stress or lead to breakage. Tape also may be banned in some hydraulic systems due to shredding, which may cause clogging of key orifices.

Anaerobic Pipe Sealants.—*Anaerobic Pipe Sealants:* The term *anaerobic pipe sealants* is used to describe anaerobic sealants used in tapered threaded assemblies for sealing and locking threaded joints. Sealing and locking are accomplished by filling the space between the threads with the sealant. In general, these pipe sealants are anaerobic adhesives consisting of mixtures of acrylic esters that remain liquid when exposed to air but harden when confined between threaded components to form an insoluble tough plastics. The strength of anaerobic pipe sealants is between that of elastomers and yielding metal.

Clamp loads need be only tight enough to prevent separation in use. Because they develop strength by curing after they are in place, these sealants are generally forgiving of tolerances, tool marks, and slight misalignment. These sealants are formulated for use on metal substrates. If the materials are used on plastics, an activator or primer should be used to prepare the surfaces.

Among the advantages of these anaerobic sealers are that they lubricate during assembly, they seal regardless of assembly torque, and they make seals that correspond with the burst rating of the pipe. They also provide controlled disassembly torque, do not cure outside the joint, and are easily dispensed on the production line. These sealants also have the lowest cost per sealed fitting. Among the disadvantages are that the materials are not suitable for oxygen service, for use with strong oxidizing agents, or for use at temperatures above 200°C. The sealants also are typically not suitable for diameters over M80 (approximately 3 inches).

The many influences faced by pipe joints during service should be known and understood at the design stage, when sealants are selected. Sealants must be chosen for reliability and long-term quality. Tapered pipe threads must remain leak-free under the severest vibration and chemical attack, also under heat and pressure surges.

Applications of aerobic sealants are found in industrial plant fluid power systems, the textile industry, chemical processing, utilities and power generation facilities, petroleum refining, and in marine, automotive, and industrial equipment. The materials are also used in the pulp and paper industries, in gas compression and distribution, and in waste-treatment facilities.

MOTION CONTROL

The most important factor in the manufacture of accurately machined components is the control of motion, whatever power source is used. For all practical purposes, motion control is accomplished by electrical or electronic circuits, energizing or deenergizing actuators such as electric motors or solenoid valves connected to hydraulic or pneumatic cylinders or motors. The accuracy with which a machine tool slide, for example, may be brought to a required position, time after time, controls the dimensions of the part being machined. This accuracy is governed by the design of the motion control system in use.

There is a large variety of control systems, with power outputs from milliwatts to megawatts, and they are used for many purposes besides motion control. Such a system may control a mechanical positioning unit, which may be linear or rotary, its velocity, acceleration, or combinations of these motion parameters. A control system may also be used to set voltage, tension, and other manufacturing process variables and to actuate various types of solenoid-operated valves. The main factors governing design of control systems are whether they are to be open- or closed-loop; what kinds and amounts of power are available; and the function requirements.

Factors governing selection of control systems are listed in Table 1.

Table 1. Control System Application Factors

Type of System	Nature of required control motion, i.e., position, velocity, acceleration
Accuracy	Controlled output versus input
Mechanical Load	Viscous friction, coulomb friction, starting friction, load inertia
Impact Loads	Hitting mechanical stops and load disturbances
Ratings	Torque or force, and speed
Torque	Peak instantaneous torque
Duty Cycle	Load response, torque level, and duration and effect on thermal response
Ambient Temperature	Relation to duty cycle and internal temperature rise, and to the effect of temperature on the sensor
Speed of Response	Time to reach commanded condition. Usually defined by a response to a stepped command
Frequency Response	Output to input ratio versus frequency, for varying frequency and specified constant input amplitude. Usually expressed in decibels
No-Load Speed	Frequently applies to maximum kinetic energy and to impact on stops; avoiding overspeeding
Backdriving	With power off, can the load drive the motor? Is a fail-safe brake required? Can the load backdrive with power on without damage to the control electronics? (Electric motor acting as a generator)
Power Source	Range of voltage and frequency within which the system must work. Effect of line transients
Environmental Conditions	Range of nonoperating and operating conditions, reliability and serviceability, scheduled maintenance

Open-Loop Systems.—The term open-loop typically describes use of a rheostat or variable resistance to vary the input voltage and thereby adjust the speed of an electric motor, a low-accuracy control method because there is no output sensor to measure the performance. However, use of stepper motors (see Table 2, and page 2473) in open-loop systems can make them very accurate. Shafts of stepper motors are turned through a fixed angle for every electrical pulse transmitted to them. The maximum pulse rate can be high, and the shaft can be coupled with step-down gear drives to form inexpensive, precise drive units with wide speed ranges. Although average speed with stepper motors is exact, speed modulation can occur at low pulse rates and drives can incur serious resonance problems.

Table 2. Control Motor Types

AC Motors	Induction motors, simplest, lowest cost, most rugged, can work directly off the ac line or through an inexpensive, efficient, and compact thyristor controller. Useful in fan and other drives where power increases rapidly with speed as well as in simple speed regulation. Ac motors are larger than comparable permanent-magnet motors
Two-Phase Induction Motors	Often used as control motors in small electromechanical control systems. Power outputs range from a few milliwatts to tens of watts
Split-Field Series Motors	Work on both ac and dc. Feature high starting torque, low cost, uniform power output over a wide speed range, and are easily reversed with a single-pole three-position switch. Very easy to use with electric limit switches for controlling angle of travel
Permanent-Magnet Motors	Operate on dc, with high power output and high efficiency. The most powerful units use rare-earth magnets and are more expensive than conventional types. Lower-cost ferrite magnets are much less expensive and require higher gear-reduction ratios, but at their higher rated speeds are very efficient
Brushless DC Motors	Use electrical commutation and may be applied as simple drive motors or as four-quadrant control motors. The absence of brushes for commutation ensures high reliability and low electromagnetic interference
Stepper Motors	Index through a fixed angle for each input pulse so that speed is in exact proportion to pulse rate and the travel angle increases uniformly with the number of pulses. Proper application in systems with backlash and load inertia requires special care
Wound-Field DC Motors	For subfractional to integral horsepower applications where size is not significant. Cost is moderate because permanent magnets are not required. Depending on the windings, output characteristics can be adjusted for specific applications

Open-loop systems are only as accurate as the input versus output requirement can be calibrated, including the effects of changes in line voltage, temperature, and other operating conditions.

Closed-Loop Systems.—Table 3 shows some parameters and characteristics of closed-loop systems, and a simple example of such a system is shown below. A command may be input by a human operator, it may be derived from another piece of system equipment, or it may be generated by a computer. Generally, the command is in the form of an electrical signal. The system response is converted by the output sensor to a compatible, scaled electrical signal that may be compared with the input command, the difference constituting an error signal. It is usually required that the error be small, so it is amplified and applied to an appropriate driving unit. The driver may take many forms, but for motion control it is usually a motor.

The amplified error voltage drives the motor to correct the error. If the input command is constant, the system is a closed-loop regulator.

Closed-loop systems use feedback sensors that measure system output and give instructions to the power drive components, based on the measured values. A typical closed-loop speed control, for instance, uses a tachometer as a feedback sensor and will correct automatically for differences between the tachometer output and the commanded speed. All motion control systems require careful design to achieve good practical performance. Closed-loop systems generally cost more than open-loop systems because of the extra cost of the tachometer or transducer used for output measurement. Faster response components also increase cost.

Table 3. Closed-Loop System Parameters and Characteristics

Step Response	The response of the system to a step change in the input command. The response to a large step, which can saturate the system amplifier, is different from the response to a small nonsaturating step. Initial overshoots may not be permissible in some types of equipment
Frequency Response	System response to a specified small-amplitude sinusoidal command where frequency is varied over the range of interest. The response is in decibels (dB), where dB = 20 $\log_{10}$(output/input). This characteristic determines whether the system is responsive enough to meet requirements
Bandwidth	The effective range of input frequencies within which the control system responds well. The bandwidth is often described by the point where the frequency response is down by three decibels. Bandwidth is usually defined in Hz (cycles/sec) or $\omega = 2\pi \times$ Hz (radians/sec)
Loading	The torque required to drive the load and the load inertia. The amplifier must supply enough power to meet acceleration as well as output power requirements. If the load is nonlinear, its effect on error must be within specifications. Behavior may vary considerably, depending on whether the load aids or opposes motor torque, as in a hoist
Output Stiffness	A measure of the system's response to load disturbances. Dynamic stiffness measures the system's response to a rapidly varying load
Resonant Peaks	Can show up in frequency-response testing as sharp (undamped) resonances. These resonances cannot be tolerated in the normal frequency range of the control system because they can lead to oscillation and vibration
No load, or maximum speed and maximum torque	Can be controlled by voltage or current limiting in the electronic amplifier. A slip clutch can also be used for torque limiting, particularly to avoid impact damage

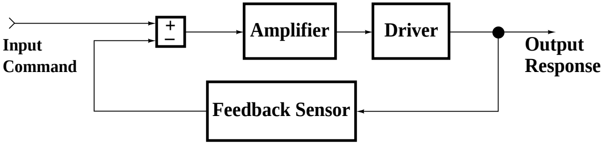


Fig. 1. General Arrangement of a Closed-Loop Control System

Accuracy of closed-loop systems is directly related to the accuracy of the sensor, so that choosing between open-loop and closed-loop controls may mean choosing between low price and consistent, accurate repeatability. In the closed-loop arrangement in Fig. 1, the sensor output is compared with the input command and the difference is amplified and applied to the motor to produce a correction. When the amplifier gain is high (the difference is greatly enlarged), even a small error will generate a correction. However, a high gain can lead to an unstable system due to inherent delays between the electrical inputs and outputs, especially with the motor.

Response accuracy depends not only on the precision of the feedback sensor and the gain of the amplifier, but also on the rate at which the command signal changes. The ability of the control system to follow rapidly changing inputs is naturally limited by the maximum motor speed and acceleration.

Amplified corrections cannot be applied to the motor instantaneously, and the motor does not respond immediately. Overshoots and oscillations can occur and the system must be adjusted or tuned to obtain acceptable performance. This adjustment is called damping the system response. Table 4 lists a variety of methods of damping, some of which require specialized knowledge.

Table 4. Means of Damping System Response

Network Damping	Included in the electrical portion of the closed loop. The networks adjust amplitude and phase to minimize control system feedback oscillations. Notch networks are used to reduce gain at specific frequencies to avoid mechanical resonance oscillations
Tachometer Damping	Feedback proportional to output velocity is added to the error signal for system stabilization
Magnetic Damping	Viscous or inertial dampers on the motor rear shaft extension for closed-loop stabilization. Similar dampers use silicone fluid instead of magnetic means to provide damping
Nonlinear Damping	Used for special characteristics. Inverse error damping provides low damping for large errors, permitting fast slewing toward zero and very stable operation at zero. Other nonlinearities meet specific needs, for example, coulomb friction damping works well in canceling backlash oscillations
Damping Algorithms	With information on output position or velocity, or both, sampled data may be used with appropriate algorithms to set motor voltage for an optimum system response

The best damping methods permit high error amplification and accuracy, combined with the desired degree of stability. Whatever form the output takes, it is converted by the output sensor to an electrical signal of compatible form that can be compared with the input command. The error signal thus generated is amplified before being applied to the driving unit.

Drive Power.—Power for the control system often depends on what is available and may vary from single- and three-phase ac 60 or 400 Hz, through dc and other types. Portable or mobile equipment is usually battery-powered dc or an engine-driven electrical generator. Hydraulic and pneumatic power may also be available. Cost is often the deciding factor in the choice.

Table 5. Special Features of Controllers

Linear or Pulse-Width Modulated	Linear is simpler, PWM is more complex and can generate electromagnetic interference, but is more efficient
Current Limiting	Sets limits to maximum line or motor current. Limits the torque output of permanent magnet motors. Can reduce starting transients and current surges
Voltage Limiting	Sets limits to maximum motor speed. Permits more uniform motor performance over a wide range of line voltages
Energy Absorption	Ability of the controller to absorb energy from a dc motor drive, back-driven by the load
EMI Filtering	Especially important when high electrical gain is required, as in thermocouple circuits, for example
Isolation	Of input and output, sometimes using optoisolators, or transformers, when input and output circuits require a high degree of isolation

Control Function.—The function of the control is usually set by the designer of the equipment and needs careful definition because it is the basis for the overall design. For instance, in positioning a machine tool table, such aspects as speed of movement and permissible variations in speed, accuracy of positioning, repeatability, and overshoot are among dozens of factors that must be considered. Some special features of controllers are listed in Table 5. Complex electromechanical systems require more knowledge of design and debugging than are needed for strictly mechanical systems.

Electromechanical Control Systems.—Wiring is the simplest way to connect components, so electromechanical controls are more versatile than pure hydraulic or pneumatic controls. The key to this versatility is often in the controller, the fundamental characteristic of which is its power output. The power output must be compatible with motor and load requirements. Changes to computer chips or software can usually change system performance to suit the application.

When driving a dc motor, for instance, the controller must supply sufficient power to match load requirements as well as motor operating losses, at minimum line voltage and maximum ambient temperature. The system's wiring must not be greatly sensitive to transient or steady-state electrical interference, and power lines must be separated from control signal lines, or appropriately shielded and isolated to avoid cross-coupling. Main lines to the controller must often include electrical interference filters so that the control system does not affect the power source, which may influence other equipment connected to the same source. For instance, an abruptly applied step command can be smoothed out so that heavy motor inrush currents are avoided. The penalty is a corresponding delay in response.

Use of current limiting units in a controller will not only set limits to line currents, but will also limit motor torque. Electronic torque limiting can frequently avoid the need for mechanical torque limiting. An example of the latter is using a slip clutch to avoid damage due to overtravel, the impact of which usually includes the kinetic energy of the moving machine elements. In many geared systems, most of the kinetic energy is in the motor. Voltage limiting is less useful than current limiting but may be needed to isolate the motor from voltage transients on the power line, to prevent overspeeding, as well as to protect electronic components.

Mechanical Stiffness.—When output motion must respond to a rapidly changing input command, the control system must have a wide bandwidth. Where the load mass (in linear motion systems) or the polar moment of inertia (in rotary systems) is high, there is a possibility of resonant oscillations. For the most stable and reliable systems, with a defined load, a high system mechanical stiffness is preferred. To attain this stiffness requires strengthening shafts, preloading bearings, and minimizing free play or backlash. In the best-performing systems, motor and load are coupled without intervening compliant members. Even tightly bolted couplings can introduce compliant oscillations resulting from extremely minute slippages caused by the load motions.

Backlash is a factor in the effective compliance of any coupling but has little effect on the resonant frequency because little energy is exchanged as the load is moved through the backlash region. However, even in the absence of significant torsional resonance, a high-gain control system can “buzz” in the backlash region. Friction is often sufficient to eliminate this small-amplitude, high-frequency component.

The difficulty with direct-drive control systems lies in matching motor to load. Most electric motors deliver rated power at higher speeds than are required by the driven load, so that load power must be delivered by the direct-drive motor operating at a slow and relatively inefficient speed. Shaft power at low speed involves a correspondingly high torque, which requires a large motor and a high-power controller. Motor copper loss (heating) is high in delivering the high motor torque. However, direct-drive motors provide maximum load velocity and acceleration, and can position massive loads within seconds of arc (rotational) or tenths of thousandths of an inch (linear) under dynamic conditions.

Where performance requirements are moderate, the required load torque can be traded off against speed by using a speed-changing transmission, typically, a gear train. The transmission effectively matches the best operating region of the motor to the required operating region of the load, and both motor and controller can be much smaller than would be needed for a comparable direct drive.

Torsional Vibration.—Control system instabilities can result from insufficient stiffness between the motor and the inertia of the driven load. The behavior of such a system is similar to that of a torsional pendulum, easily excited by commanded motions of the control system. If frictional losses are moderate to low, sustained oscillations will occur. In spite of the complex dynamics of the closed-loop system, the resonant frequency, as for a torsional pendulum, is given to a high degree of accuracy by the formula:

$$f_n = \frac{1}{2\pi} \times \sqrt{\frac{K}{J_L}}$$

where f_n is in hertz, K is torsional stiffness in in.-lb/rad, and J_L is load inertia in in.-lb-sec^2/rad. If this resonant frequency falls within the bandwidth of the control system, self-sustained oscillations are likely to occur. These oscillations are often overlooked by control systems analysts because they do not appear in simple control systems, and they are very difficult to correct.

Friction inherently reduces the oscillation by dissipating the energy in the system inertia. If there is backlash between motor and load, coulomb friction (opposing motion but independent of speed) is especially effective in damping out the oscillation. However, the required friction for satisfactory damping can be excessive, introducing positioning error and adding to motor (and controller) power requirements. Friction also varies with operating conditions and time.

The most common method of eliminating torsional oscillation is to introduce a filter in the error channel of the control system to shape the gain characteristic as a function of frequency. If the torsional resonance is within the required system bandwidth, little can be done except stiffening the mechanical system and increasing the resonant frequency. If the filter reduces the gain within the required bandwidth, it will reduce performance. This method will work only if the natural resonance is above the minimum required performance bandwidth.

The simplest shaping network is the notch network (Table 4, network damping), which, in effect, is a band-rejection filter that sharply reduces gain at the notch frequency. By locating the notch frequency so as to balance out the torsional resonance peak, the oscillation can be eliminated. Where there are several modes of oscillation, several filter networks can be connected in series.

Electric Motors.—Electric motors for control systems must suit the application. Motors used in open-loop systems (excluding stepper motors) need not respond quickly to input command changes. Where the command is set by a human, response times of hundreds of milliseconds to several seconds may be acceptable. Slow response does not lead to the instabilities that time delays can introduce into closed-loop systems.

Closed-loop systems need motors with fast response, of which the best are permanent-magnet dc units, used where wide bandwidth, efficient operation, and high power output are required. Table 2 lists some types of control motors and their characteristics. An important feature of high-performance, permanent-magnet motors using high-energy, rare-earth magnets is that their maximum torque output capacity can be 10 to 20 or more times higher than their rated torque. In intermittent or low-duty-cycle applications, very high torque loads can be driven by a given motor. However, when rare-earth magnets (samarium cobalt or neodymium) are not used, peak torque capability may be limited by the possibility of demagnetization. Rare-earth magnets are relatively expensive, so it is important to verify peak torque capabilities for lower-cost motors that may use weaker Alnico or ferrite magnets.

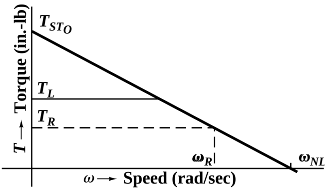


Fig. 2. Idealized Control Motor Characteristics for a Consistent Set of Units

Duty-cycle calculations are an aspect of thermal analysis that are well understood and are not covered here. Motor manufacturers usually supply information on thermal characteristics including thermal time constants and temperature rise per watt of internal power dissipation.

Characteristics of permanent-magnet motors are defined with fair accuracy by relatively few parameters. The most important characteristics are: D_M motor damping in lb-in.-sec/rad; J_M motor inertia in lb-in.-sec_2/rad; and R winding resistance in ohms. Fig. 1 shows other control motor characteristics, T_{STo} stall torque with no current limiting; T_L maximum torque with current limiting; ω_{NL} no-load speed; ω_R rated speed. Other derived motor parameters include V rated voltage in volts; $I_{STo} = V/R$ current in amperes at stall with no current limiting; I_L ampere limit, adjusted in amplifier; I_R rated current; $K_T = T_{STo}/I_{STo}$ torque constant in in.-lb/ampere; $K_E = V/\omega_{NL}$ voltage constant in volt/rad/sec; $K_M = K_T/\sqrt{R}$, torque per square root of winding resistance; $D_M = T_{STo}/\omega_{NL}$ motor damping in in.-lb/rad/sec; and $T_M = J_M/D_M$ motor mechanical time constant in seconds.

Stepper Motors.—In a stepper motor, power is applied to a wound stator, causing the brushless rotor to change position to correspond with the internal magnetic field. The rotor maintains its position relative to the internal magnetic field at all times. In its most common mode of operation, the stepper motor is energized by an electronic controller whose current output to the motor windings defines the position of the internally generated magnetic field. Applying a command pulse to the controller will change the motor currents to reposition the rotor. A series of pulses, accompanied by a direction command, will cause rotation in uniformly spaced steps in the specified direction.

If the pulses are applied at a sufficiently high frequency, the rotor will be carried along with the system's inertia and will rotate relatively uniformly but with a modulated velocity. At the other extreme, the response to a single pulse will be a step followed by an overshoot and a decaying oscillation. Where the application cannot permit the oscillation, damping can be included in the controller.

Stepper motors are often preferred because positions of the rotor are known from the number of pulses and the step size. An initial index point is required as an output position reference, and care is required in the electronic circuits to avoid introducing random pulses that will cause false positions. As a minimum, the output index point on an appropriate shaft can verify the step count during operation.

Gearing.—In a closed-loop system, gearing may be used to couple a high-speed, low-torque motor to a lower-speed, higher-torque load. The gearing must meet requirements for accuracy, strength, and reliability to suit the application. In addition, the closed loop requires minimum backlash at the point where the feedback sensor is coupled. In a velocity-controlled system, the feedback sensor is a tachometer that is usually coupled directly to the rotor shaft. Backlash between motor and tachometer, as well as torsional compliance, must be minimized for stable operation of a high-performance system. Units combining motor and tachometer on a single shaft can usually be purchased as an assembly.

By contrast, a positioning system may use a position feedback sensor that is closely coupled to the shaft being positioned. As with the velocity system, backlash between the motor and feedback sensor must be minimized for closed-loop stability. Antibacklash gearing is frequently used between the gearing and the position feedback sensor. When the position feedback sensor is a limited rotation device, it may be coupled to a gear that turns faster than the output gear to allow use of its full range. Although this step-up gearing enhances it, accuracy is ultimately limited by the errors in the intermediate gearing between the position sensor and the output.

When an appreciable load inertia is being driven, it is important that the mechanical stiffness between the position sensor coupling point and the load be high enough to avoid natural torsional resonances in the passband.

Feedback Transducers.—Controlled variables are measured by feedback transducers and are the key to accuracy in operation of closed-loop systems. When the accuracy of a carefully designed control system approaches the accuracy of the feedback transducer, the need for precision in the other system components is reduced.

Transducers may measure the quantity being controlled in digital or analog form, and are available for many different parameters such as pressure and temperature, as well as distance traveled or degrees of rotation. Machine designers generally need to measure and control linear or rotary motion, velocity, position, and sometimes acceleration. Although some transducers are nonlinear, a linear relationship between the measured variable and the (usually electrical) output is most common.

Output characteristics of an analog linear-position transducer are shown in Fig. 2. By dividing errors into components, accuracy can be increased by external adjustments, and slope error and zero offsets are easily trimmed in. Nonlinearity is controlled by the manufacturer. In Fig. 2 are seen the discrete error components that can be distinguished because of the ease with which they can be canceled out individually by external adjustments. The most common compensation is for zero-position alignment, so that when the machine has been set to the start position for a sequence, the transducer can be positioned to read zero output. Alternatively, with all components in fixed positions, a small voltage can be inserted in series with the transducer output for a very accurate alignment of mechanical and electrical zeros. This method helps in canceling long-term drift, particularly in the mechanical elements.

The second most common adjustment of a position transducer is of its output gradient, that is, transducer output volts per degree. Depending on the type of analog transducer, it is usually possible to add a small adjustment to the electrical input, to introduce a proportional change in output gradient. As with the zero-position adjustment, the gradient may be set very accurately initially and during periodic maintenance. The remaining errors shown in Fig. 2, such as intrinsic nonlinearity or nonconformity, result from limitations in design and manufacture of the transducer.

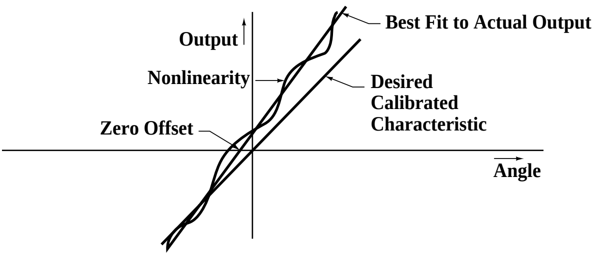


Fig. 3. Output Characteristics of a General Linear Position Transducer

Greater accuracy can be achieved in computer-controlled systems by using the computer to cancel out transducer errors. The system's mechanical values and corresponding transducer values are stored in a lookup table in the computer and referred to as necessary. Accuracies approaching the inherent repeatability and stability of the system can thus be secured. If necessary, recalibration can be performed at frequent intervals.

Analog Transducers.—The simplest analog position transducer is the resistance potentiometer, the resistance element in which is usually a deposited-film rather than a wire-wound type. Very stable resistance elements based on conductive plastics, with resolution to a few microinches and operating lives in the 100 million rotations, are available, capable of working in severe environments with high vibrations and shock and at temperatures of 150 to 200°C. Accuracies of a few hundredths, and stability of thousandths, of a per cent, can be obtained from these units by trimming the plastics resistance element as a function of angle.

Performance of resistance potentiometers deteriorates when they operate at high speeds, and prolonged operation at speeds above 10 rpm causes excessive wear and increasing output noise. An alternative to the resistance potentiometer is the variable differential transformer, which uses electrical coupling between ac magnetic elements to measure angular or linear motion without sliding contacts. These units have unlimited resolution with accuracy comparable to the best resistance potentiometers but are more expensive and require compatible electronic circuits.

A variable differential transformer needs ac energization, so an ac source is required. A precision demodulator is frequently used to change the ac output to dc. Sometimes the ac output is balanced against an ac command signal whose input is derived from the same ac source. In dealing with ac signals, phase-angle matching and an accurate amplitude-scale factor are required for proper operation. Temperature compensation also may be required, primarily due to changes in resistance of the copper windings. Transducer manufacturers will supply full sets of compatible electronic controls.

Synchros and Resolvers.—Synchros and resolvers are transducers that are widely used for sensing of angles at accuracies down to 10 to 20 arc-seconds. More typically, and at much lower cost, their accuracies are 1 to 2 arc-minutes. Cost is further reduced when accuracies of 0.1 degree or higher are acceptable.

Synchros used as angle-position transducers are made as brush types with slip rings and in brushless types. These units can rotate continuously at high speeds, the operating life of brushless designs being limited only by the bearing life. Synchros have symmetrical three-wire stator windings that facilitate transmission of angle data over long distances (thousands of feet). Such a system is also highly immune to noise and coupled signals. Practically the only trimming required for very long line systems is matching the line-to-line capacitances.

Because synchros can rotate continuously, they can be used in multispeed arrangements, where, for example, full-scale system travel may be represented by 36 or 64 full rotations. When reduced by gearing to a single, full-scale turn, a synchro's electrical inaccuracy is the typical 0.1° error divided by 36 or 64 or whatever gear ratio is used. This error is insignificant compared with the error of the gearing coupling the high-speed synchro and the single speed (1 rotation for full scale) output shaft. The accuracy is dependable and stable, using standard synchros and gearing.

Hydraulic and Pneumatic Systems

In Fig. 1 is shown a schematic of a hydraulic cylinder and the relationships between force and area that govern all hydraulic systems. Hydraulic actuators that drive the load may be cylinders or motors, depending on whether linear or rotary motion is required. The load must be defined by its torque–speed characteristics and inertia, and a suitable hydraulic actuator selected before the remaining system components can be chosen. Fluid under pressure and suitable valves are needed to control motion. Both single- and double-acting hydraulic cylinders are available, and the latter type is seen in Fig. 1.

Pressure can be traded off against velocity, if desired, by placing a different effective area at each side of the piston. The same pressure on a smaller area will move the piston at a higher speed but lower force for a given rate of fluid delivery. The cylinder shown in Fig. 1 can drive loads in either direction. The simple formulas of plane geometry relate cylinder areas, force, fluid flow, and rate of movement. Other configurations can develop equal forces and speeds in both directions.

The rotary equivalent of the cylinder is the hydraulic motor, which is defined by the fluid displacement required to turn the output shaft through one revolution, by the output torque, and by the load requirements of torque and speed. Output torque is proportional to fluid pressure, which can be as high as safety permits. Output speed is defined by the number of gallons per minute supplied to the motor. As an example, if 231 cu. in. = 1 gallon, an input of 6 gallons/min (gpm) with a 5-cu. in. displacement gives a mean speed of $6 \times 231/5 = 277$ rpm. The motor torque must be defined by lb-in. per 100 $lb_f/in.^2$ (typically) from which the required pressure can be determined. Various motor types are available.

Hydraulic Pumps.—The most-used hydraulic pump is the positive-displacement type, which delivers a fixed amount of fluid for every cycle. These pumps are also called hydrostatic because they deliver energy by static pressure rather than by the kinetic energy of a moving fluid. Positive-displacement pumps are rated by the gpm delivered at a stated speed and by the maximum pressure, which are the key parameters defining the power capacity of the hydraulic actuator. Delivered gpm are reduced under load due to leakage, and the reduction is described by the volumetric efficiency, which is the ratio of actual to theoretical output.

Hydraulic Fluids.—The hydraulic fluid is the basic means of transmitting power, and it also provides lubrication and cooling when passed through a heat exchanger. The fluid must be minimally compressible to avoid springiness and delay in response. The total system inertia reacts with fluid compliance to generate a resonant frequency, much as inertia and mechanical compliance react in an electromechanical system. Compliance must be low enough that resonances do not occur in the active bandwidth of the servomechanism, and that unacceptable transients do not occur under shock loads. Seal friction and fluid viscosity tend to damp out resonant vibrations. Shock-absorbing limit stops or cushions are usually located at the travel limits to minimize transient impact forces.

F = force (lb) = pressure × area

P_1 and P_2 = line pressure on either side of piston in $lb_f/in.^2$

d_1 and d_2 = diameters of piston rod and piston in in.

$$F_1 = \frac{\pi}{4}(d_2^2 - d_1^2) \times P_1 = 0.7854P_1(d_2^2 - d_1^2)$$

$$F_2 = \frac{\pi}{4} d_2^2 P_2 = 0.7854 P_2 d_2^2$$

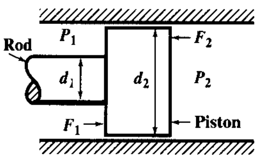


Fig. 1. Elementary Hydraulic Force/Area Formulas

Hydraulic fluids with special additives for lubrication minimize wear between moving parts. An auxiliary function is prevention of corrosion and pitting. Hydraulic fluids must also be compatible with gaskets, seals, and other nonmetallic materials.

Viscosity is another critical parameter of hydraulic fluids as high viscosity means high resistance to fluid flow with a corresponding power loss and heating of the fluid, pressure drop in the hydraulic lines, difficulty in removing bubbles, and sometimes overdamped operation. Unfortunately, viscosity falls very rapidly with increasing temperature, which can lead to reduction of the lubrication properties and excessive wear as well as increasing leakage. For hydraulic actuators operating at very low temperatures, the fluid pour point is important. Below this temperature, the hydraulic fluid will not flow. Design guidelines similar to those used with linear or rotating bearings are applicable in these conditions. Fire-resistant fluids are available for use in certain conditions such as in die casting, where furnaces containing molten metal are often located near hydraulic systems.

A problem with hydraulic systems that is absent in electromechanical systems is that of dirt, air bubbles, and contaminants in the fluid. Enclosed systems are designed to keep out contaminants, but the main problem is with the reservoir or fluid storage unit. A suitable sealer must be used in the reservoir to prevent corrosion and a filter should be used during filling. Atmospheric pressure is required on the fluid surface in the reservoir except where a pressurized reservoir is used. Additional components include coarse and fine filters to remove contaminants and these filters may be rated to remove micron sized particles (1 micron = 0.00004 in.).

Very fine filters are sometimes used in high-pressure lines, where dirt might interfere with the operation of sensitive valves. Where a high-performance pump is used, a fine filter is a requirement. Usually, only coarse filters are used on fluid inlet lines because fine filters might introduce excessive pressure drop.

Aside from the reservoir used for hydraulic fluid storage, line connections, fittings, and couplings are needed. Expansion of these components under pressure increases the mechanical compliance of the system, reducing the frequencies of any resonances and possibly interfering with the response of wide-band systems.

Formulas relating fluid flow and mechanical power follow. These formulas supplement the general force, torque, speed, and power formulas of mechanical systems.

$$F = P \times A$$

$$A = 0.7854 \times d^2$$

$$hp = 0.000583q \times \text{pressure in lb}_f/\text{in.}^2$$

1 gallon of fluid flow/min at 1 $\text{lb}_f/\text{in.}^2$ pressure = 0.000582 hp.

For rotary outputs,

$$hp = \text{torque} \times \text{rpm}/63{,}025$$

where torque is in lb-in. (Theoretical hp output must be multiplied by the efficiency of the hydraulic circuits to determine actual output.)

In the preceding equations,

P = pressure in $\text{lb}_f/\text{in.}^2$
A = piston area in in.^2
F = force in lb
q = fluid flow in gallons/min
d = piston diameter in inches

Hydraulic and Pneumatic Control Systems.—Control systems for hydraulic and pneumatic circuits are more mature than those for electromechanical systems because they have been developed over many more years. Hydraulic components are available at moderate prices from many sources. Although their design is complex, application and servicing of these systems are usually more straightforward than with electromechanical systems.

Electromechanical and hydraulic/pneumatic systems may be analyzed by similar means. The mathematical requirements for accuracy and stability are analogous, as are most performance features, although nonlinearities are caused by different physical attributes. Nonlinear friction, backlash, and voltage and current limiting are common to both types of system, but hydraulic/pneumatic systems also have the behavior characteristics of fluid-driven systems such as thermal effects and fluid flow dynamics including turbulence, leakage caused by imperfect seals, and contamination.

Both control types require overhead equipment that does not affect performance but adds to overall cost and complexity. For instance, electromechanical systems require electrical power sources and power control components, voltage regulators, fuses and circuit breakers, relays and switches, connectors, wiring and related devices. Hydraulic/pneumatic systems require fluid stored under pressure, motor-driven pumps or compressors, valves, pressure regulators/limiters, piping and fasteners, as well as hydraulic/pneumatic motors and cylinders. Frequently, the optimum system is selected on the basis of overhead equipment already available.

Electromechanical systems are generally slower and heavier than hydraulic systems and less suited to controlling heavy loads. The bandwidths of hydraulic control systems can respond to input signals of well over 100 Hz as easily as an electromechanical system can respond to, say, 10 to 20 Hz. Hydraulic systems can drive very high torque loads without intermediate transmissions such as the gear trains often used with electromechanical systems. Also, hydraulic/pneumatic systems using servo valves and piston/cylinder arrangements are inherently suited to linear motion operation, whereas electromechanical controls based on conventional electrical machines are more naturally suited to driving rotational loads.

Until recently, electromechanical systems were limited to system bandwidths of about 10 Hz, with power outputs of a few hundred watts. However, their capabilities have now been sharply extended through the use of rare-earth motor magnets having much higher energies than earlier designs. Similarly, semiconductor power components deliver much higher output power at lower prices than earlier equipment. Electromechanical control systems are now suited to applications of more than 100 hp with bandwidths up to 40 Hz and sometimes up to 100 Hz.

Although much depends on the specific design, the edge in reliability, even for high-power, fast-response needs, is shifting toward electromechanical systems. Basically, there are more things that can go wrong in hydraulic/pneumatic systems, as indicated by the shift to more electrical systems in aircraft.

Hydraulic Control Systems.—Using essentially incompressible fluid, hydraulic systems are suited to a wide range of applications, whereas pneumatic power is generally limited to simpler uses. In Fig. 2 are shown the essential features of a simple linear hydraulic control system and a comparable system for driving a rotating load.

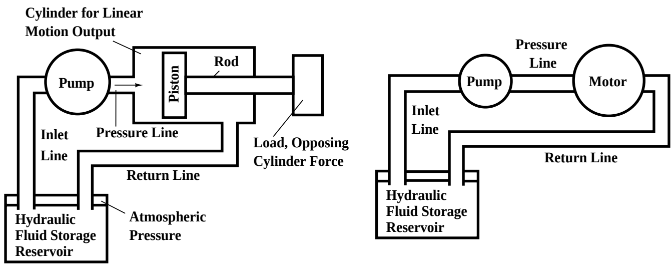


Fig. 2. (left) A Simple Linear Hydraulic Control System in Which the Load Force Returns the Piston and (right) a Comparable System for Driving a Rotating Load

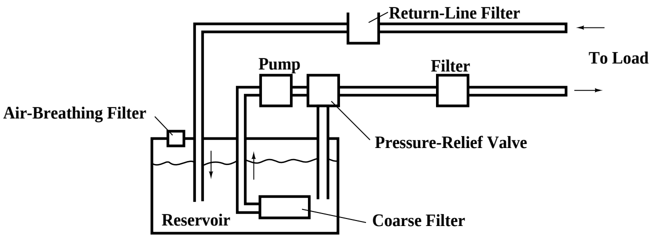


Fig. 3. Some of the Auxiliary Components Used in a Practical Hydraulic System

Hydraulic controls of the type shown have fast response and very high load capacities. In a linear actuator, for example, each $lb_f/in.^2$ of system pressure acts against the area of the piston to generate the force applied. Hydraulic pressures of up to 3000 $lb_f/in.^2$ are readily obtained from hydraulic pumps, so that cylinders can exert forces of hundreds of tons without the need for speed-reducing transmission systems to increase the force. The hydraulic fluid distributes heat, so it helps cool the system.

Systems similar to those in Fig. 2 can be operated in open- or closed-loop modes. Open-loop operation can be controlled by programming units that initiate each step by operating relays, limit switches, solenoid valves, and other components to generate the forces over the required travel ranges. Auxiliary components are used to ensure safe operation and make such systems flexible and reliable, as shown in Fig. 3.

In the simplest mode, whether open- or closed-loop, hydraulic system operation may be discontinuous or proportional. Discontinuous operation, sometimes called bang-bang, or on–off, works well, is widely used in low- to medium-accuracy systems, and is easy to maintain. In this closed-loop mode, accuracy is limited; if the response to error is set too high, the system will oscillate between on–off modes, with average output at about the desired value. This oscillation, however, can be noisy, introduces system transients, and may cause rapid wear of system components.

Another factor to be considered in on–off systems is the shock caused by sudden opening and closing of high-pressure valves, which introduce transient pulses in the fluid flow and can cause high stresses in components. These problems can be addressed by the use of pressure-limiting relief valves and other units.

Proportional Control Systems.—Where the highest accuracy is required, perhaps in two directions, and with aiding or opposing forces or torques, a more sophisticated proportional control, closed-loop system is preferred. As shown in Fig. 4, the amplifier and electric servomotor used in electromechanical closed-loop systems is replaced in the closed-loop hydraulic system by an electronically controlled servo-valve. In its simplest form, the valve uses a linear motor to position the spool that determines the flow path for the hydraulic fluid. In some designs, the linear motor may be driven by a solenoid against a bias spring on the value spool. In other arrangements, the motor may be a bidirectional unit that permits a fluid flow depending on the polarity and amplitude of the voltage supplied to the motor.

Such designs can be used in proportional control systems to achieve smooth operation and minimum nonlinearities, and will give the maximum accuracy required by the best machine tool applications. Where very high power must be controlled, use is often made of a two-stage valve in which the output from the first stage is used to drive the second-stage valve, as shown in Fig. 5.

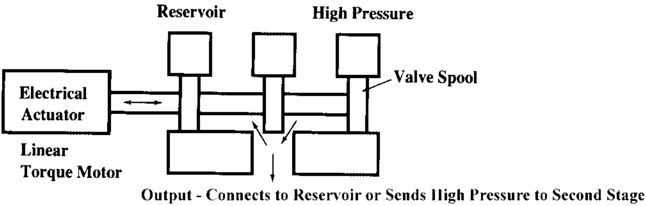


Fig. 4.

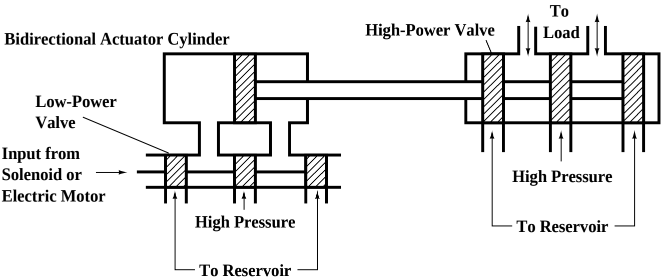


Fig. 5. Two-Stage Valve for Large-Power Control from a Low-Power Input

Electronic Controls.—An error-sensing electronic amplifier drives the solenoid motor of Fig. 5, which provides automatic output correction in a closed-loop system. The input is an ideal place to introduce electrical control features, adding greatly to the versatility of the control system. The electronic amplifier can provide the necessary driving power using pulse-width modulation as required, for minimum heating. The output can respond to signals in the low-microvolt range.

A major decision is whether to use analog or digital control. Although analog units are simple, they are much less versatile than their digital counterparts. Digital systems can be readjusted for total travel, speed, and acceleration by simple reprogramming. Use of appropriate feedback sensors can match accuracy to any production requirement, and a single digital system can be easily adapted to a great variety of similar applications. This adaptability is an important cost-saving feature for moderate-sized production runs. Modern microprocessors can integrate the operation of sets of systems.

Because nonlinearities and small incremental motions are easy to implement, digital systems are capable of very smooth acceleration, which avoids damaging shocks and induced leaks, and enhances reliability so that seals and hose connections last longer. The accuracy of digital control systems depends on transducer availability, and a full range of such devices has been developed and is now available.

Other features of digital controls are their capacity for self-calibration, easy digital readout, and periodic self-compensation. For example, it is easy to incorporate backlash compensation. Inaccuracies can be corrected by using lookup tables that may themselves be updated as necessary. Digital outputs can be used as part of an inspection plan, to indicate need for tool changing, adjustment or sharpening, or for automatic record keeping. Despite continuing improvements in analog systems, digital control of hydraulic systems is favored in large plants.

Pneumatic Systems.—Hydraulic systems transmit power by means of the flow of an essentially incompressible fluid. Pneumatic systems use a highly compressible gas. For this reason, a pneumatic system is slower in responding to loads, especially sudden output loads, than a hydraulic system. Similarly, torque or force requires time and output motion to build up. Response to sudden output loads shows initial overshoot. Much more complex networks or other damping means are required to develop stable response in closed-loop systems. On the other hand, there are no harmful shock waves analogous to the transients that can occur in hydraulic systems, and pneumatic system components last comparatively longer.

Notwithstanding their performance deficiencies, pneumatic systems have numerous desirable features. Pneumatic systems avoid some fire hazards compared with the most preferred hydraulic fluids. Air can be vented to the atmosphere so a flow line only is needed, reducing the complexity, cost, and weight of the overall system. Pneumatic lines, couplings, and fittings are lighter than their hydraulic counterparts, often a significant advantage. The gaseous medium also is lighter than hydraulic fluid, and pneumatic systems are usually easier to clean, assemble, and generally maintain. Fluid viscosity and its temperature variations are virtually negligible with pneumatic systems.

Among drawbacks with pneumatics are that lubrication must be carefully designed in, and more power is needed to achieve a desired pressure when the fluid medium is a compressible gas. Gas under high pressure can cause an explosion if its storage tank is damaged, so storage must have substantial safety margins. Gas compressibility makes pneumatic systems 1 or 2 orders of magnitude slower than hydraulic systems.

The low stiffness of pneumatic systems is another indicator of the long response time. Resonances occur between the compressible gas and equivalent system inertias at lower frequencies. Even the relatively low speed of sound in connecting lines contributes to response delay, adding to the difficulty of closed-loop stabilization. Fortunately, it is possible to construct pneumatic analogs to electrical networks to simplify stabilization at the exact point of the delays. Such pneumatic stabilizing means are commercially available and are important elements of closed-loop pneumatic control systems.

In contrast with hydraulic systems, where speed may be controlled by varying pump output, pneumatic system control is almost exclusively by valves, which control the flow from a pneumatic accumulator or pressure source. The pressure is maintained between limits by an intermittently operated pump. Low-pressure outlet ports must be large enough to accommodate the high volume of the expanded gas. In Fig. 6 is shown a simplified system for closed-loop position control applied to an air cylinder, in which static accuracy is controlled by the position sensor. Proper design requires a good theoretical analysis and attention to practical design if good, stable, closed-loop response is to be achieved.

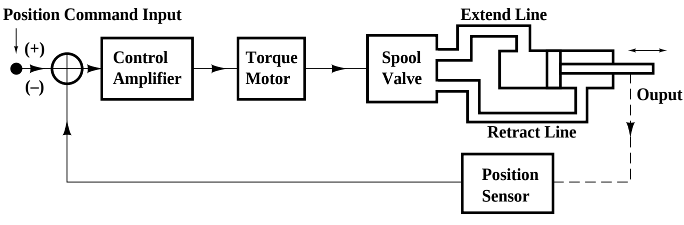


Fig. 6. A Pneumatic Closed-Loop Linear Control System

O-RINGS

An O-ring is a one-piece molded elastomeric seal with a circular cross-section that seals by distortion of its resilient elastic compound. Dimensions of O-rings are given in ANSI/SAE AS568A, Aerospace Size Standard for O-rings. The standard ring sizes have been assigned identifying dash numbers that, in conjunction with the compound (ring material), completely specifies the ring. Although the ring sizes are standardized, ANSI/SAE AS568A does not cover the compounds used in making the rings; thus, different manufacturers will use different designations to identify various ring compounds. For example, 230-8307 represents a standard O-ring of size 230 (2.484 in. ID by 0.139 in. width) made with compound 8307, a general-purpose nitrile compound. O-ring material properties are discussed at the end of this section.

When properly installed in a groove, an O-ring is normally slightly deformed so that the naturally round cross-section is squeezed diametrically out of round prior to the application of pressure. This compression ensures that under static conditions, the ring is in contact with the inner and outer walls enclosing it, with the resiliency of the rubber providing a zero-pressure seal. When pressure is applied, it tends to force the O-ring across the groove, causing the ring to further deform and flow up to the fluid passage and seal it against leakage, as in Fig. 1(a). As additional pressure is applied, the O-ring deforms into a D shape, as in Fig. 1(b). If the clearance gap between the sealing surface and the groove corners is too large or if the pressure exceeds the deformation limits of the O-ring material (compound), the O-ring will extrude into the clearance gap, reducing the effective life of the seal. For very low-pressure static applications, the effectiveness of the seal can be improved by using a softer durometer compound or by increasing the initial squeeze on the ring, but at higher pressures, the additional squeeze may reduce the ring's dynamic sealing ability, increase friction, and shorten ring life.

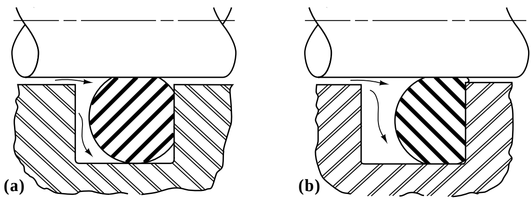


Fig. 1.

The initial diametral squeeze of the ring is very important in the success of an O-ring application. The squeeze is the difference between the ring width W and the gland depth F (Fig. 2) and has a great effect on the sealing ability and life of an O-ring application.

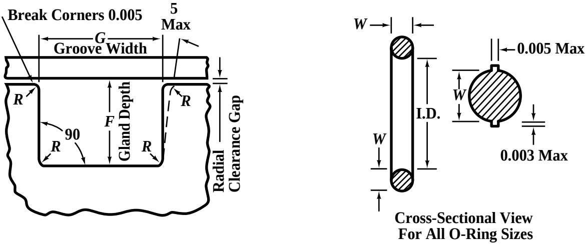


Fig. 2. Groove and Ring Details

The ideal squeeze varies according to the ring cross-section, with the average being about 20 per cent, i.e., the ring's cross-section *W* is about 20 per cent greater than the gland depth *F* (groove depth plus clearance gap). The groove width is normally about 1.5 times larger than the ring width *W*. When installed, an O-ring compresses slightly and distorts into the free space within the groove. Additional expansion or swelling may also occur due to contact of the ring with fluid or heat. The groove must be large enough to accommodate the maximum expansion of the ring or the ring may extrude into the clearance gap or rupture the assembly. In a dynamic application, the extruded ring material will quickly wear and fray, severely limiting seal life.

To prevent O-ring extrusion or to correct an O-ring application, reduce the clearance gap by modifying the dimensions of the system, reduce the system operating pressure, install antiextrusion backup rings in the groove with the O-ring, as in Fig. 3, or use a harder O-ring compound. A harder compound may result in higher friction and a greater tendency of the seal to leak at low pressures. Backup rings, frequently made of leather, Teflon, metal, phenolic, hard rubber, and other hard materials, prevent extrusion and nibbling where large clearance gaps and high pressure are necessary.

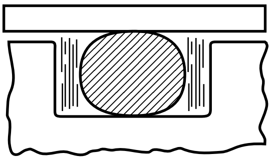


Fig. 3. Preferred Use of Backup Washers

The most effective and reliable sealing is generally provided by using the diametrical clearances given in manufacturers' literature. However, the information in Table 1 may be used to estimate the gland depth (groove depth plus radial clearance) required in O-ring applications. The radial clearance used (radial clearance equals one-half the diametral clearance) also depends on the system pressure, the ring compound and hardness, and specific details of the application.

Table 1. Gland Depth for O-Ring Applications

Standard O-Ring Cross-Sectional Diameter (in.)	Gland Depth (in.)	
	Reciprocating Seals	Static Seals
0.070	0.055 to 0.057	0.050 to 0.052
0.103	0.088 to 0.090	0.081 to 0.083
0.139	0.121 to 0.123	0.111 to 0.113
0.210	0.185 to 0.188	0.170 to 0.173
0.275	0.237 to 0.240	0.226 to 0.229

Source: Auburn Manufacturing Co. When possible, use manufacturer recommendations for clearance gaps and groove depth.

Fig. 4 indicates conditions where O-ring seals may be used, depending on the fluid pressure and the O-ring hardness. If the conditions of use fall to the right of the curve, extrusion of the O-ring into the surrounding clearance gap will occur, greatly reducing the life of the ring. If conditions fall to the left of the curve, no extrusion of the ring will occur, and the ring may be used under these conditions. For example, in an O-ring application with a 0.004-in. diametral clearance and 2500-psi pressure, extrusion will occur with a 70 durometer O-ring but not with an 80 durometer O-ring. As the graph indicates, high-pressure applications require lower clearances and harder O-rings for effective sealing.

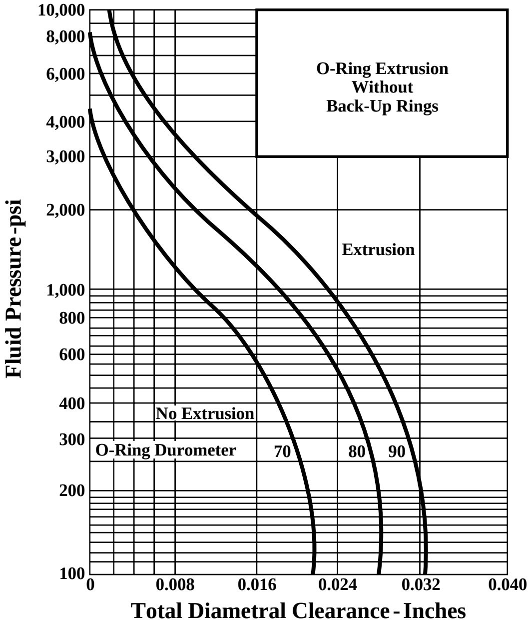


Fig. 4. Extrusion Potential of O-Rings as a Function of Hardness and Clearance

Recommended groove width, clearance dimensions, and bottom-of-groove radius for O-ring numbers up to 475 (25.940-in. ID by 0.275-in. width) can be found using Table 2 in conjunction with Fig. 5. In general, except for ring cross-sections smaller than $\frac{1}{16}$ in., the groove width is approximately 1.5*W*, where *W* is the ring cross-sectional diameter. Straight-sided grooves are best for preventing extrusion of the ring or nibbling; however, for low-pressure applications (less than 1500 psi) sloped sides with an angle up to 5° can be used to simplify machining of the groove. The groove surfaces should be free of burrs, nicks, or scratches. For static seals (i.e., no contact between the O-ring and any moving parts), the groove surfaces should have a maximum roughness of 32 to 63 μin. rms for liquid-sealing applications and 16 to 32 μin. rms for gaseous-sealing applications. In dynamic seals, relative motion exists between the O-ring and one or more parts and the maximum groove surface roughness should be 8 to 16 μin. rms for sliding contact applications (reciprocating seals, for example) and 16 to 32 μin. rms for rotary contact applications (rotating and oscillating seals).

In dynamic seal applications, the roughness of surfaces in contact with O-rings (bores, pistons, and shafts, for example) should be 8 to 16 μin. rms, without longitudinal or circumferential scratches. Surface finishes of less than 5 μin. rms are too smooth to give a good seal life because they wipe too clean, causing the ring to wear against the housing in the absence of a lubricating film. The best-quality surfaces are honed, burnished, or hard chromium plated. Soft and stringy metals such as aluminum, brass, bronze, Monel, or free machining stainless steel should not be used in contact with moving seals. In static applications, O-ring contacting surfaces should have a maximum surface roughness of 64 to 125 μin. rms.

Table 2. Diametral Clearance and Groove Sizes for O-Ring Applications

ANSI/SAE AS568 Number	Tolerances		Diametral Clearance, *D*		Groove Width, *G*			Bottom of Groove Radius, *R*
					Backup Rigs			
	A	*B*	Reciprocating & Static Seals	Rotary Seals	None	One	Two	
001	+0.001 −0.000	+0.000 −0.001	0.002 to 0.004	0.012 to 0.016	0.063			0.005 to 0.015
002					0.073			
003					0.083			
004 to 012					0.094	0.149	0.207	
013 to 050	+0.002 −0.000	+0.000 −0.002						
102 to 129			0.002 to 0.005		0.141	0.183	0.245	
130 to 178			0.002 to 0.006					
201 to 284				0.016 to 0.020	0.188	0.235	0.304	0.010 to 0.025
309 to 395			0.003 to 0.007		0.281	0.334	0.424	0.020 to 0.035
425 to 475	+0.003 −0.000	+0.000 −0.003	0.004 to 0.010		0.375	0.475	0.579	

Source: Auburn Manufacturing Co. All dimensions are in inches. Clearances listed are minimum and maximum values; standard groove widths may be reduced by about 10 per cent for use with ring compounds that free swell less than 15 per cent. Dimension *A* is the ID of any surface contacted by the outside circumference of the ring; *B* is the OD of any surface contacted by the inside circumference of the ring.

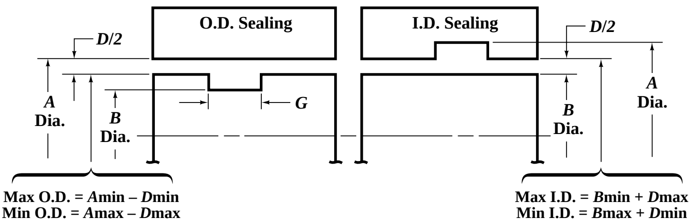


Fig. 5. Installation data for use with Table 2. Max and Min are maximum and minimum piston and bore diameters for O.D. and I.D., respectively.

The preferred bore materials are steel and cast iron, and pistons should be softer than the bore to avoid scratching them. The bore sections should be thick enough to resist expansion and contraction under pressure so that the radial clearance gap remains constant, reducing the chance of damage to the O-ring by extrusion and nibbling. Some compatibility problems may occur when O-rings are used with plastics parts because certain compounding ingredients may attack the plastics, causing crazing of the plastics surface.

O-rings are frequently used as driving belts in round bottom or V-grooves with light tension for low-power drive elements. Special compounds are available with high resistance to stress relaxation and fatigue for these applications. Best service is obtained in drive belt applications when the initial belt tension is between 80 and 200 psi and the initial installed stretch is between 8 and 25 per cent of the circumferential length. Most of the compounds used for drive belts operate best between 10 and 15 per cent stretch, although polyurethane has good service life when stretched as much as 20 to 25 per cent.

Table 3. Typical O-Ring Compounds

Nitrile	General-purpose compound for use with most petroleum oils, greases, gasoline, alcohols and glycols, LP gases, propane and butane fuels. Also for food service to resist vegetable and animal fats. Effective temperature range is about −40° to 250°F. Excellent compression set, tear and abrasion resistance, but poor resistance to ozone, sunlight and weather. Higher-temperature nitrile compounds with similar properties are also available.
Hydrogenated Nitrile	Similar to general-purpose nitrile compounds with improved high-temperature performance, resistance to aging, and petroleum product compatibility.
Polychloroprene (Neoprene)	General-purpose compound with low compression set and good resistance to elevated temperatures. Good resistance to sunlight, ozone, and weathering, and fair oil resistance. Frequently used for refrigerator gases such as Freon. Effective temperature range is about −40° to 250°F.
Ethylene Propylene	General-purpose compound with excellent resistance to polar fluids such as water, steam, ketones, and phosphate esters, and brake fluids, but not resistant to petroleum oils and solvents. Excellent resistance to ozone and flexing. Recommended for belt-drive applications. Continuous duty service in temperatures up to 250°F.
Silicon	Widest temperature range (−150° to 500°F) and best low-temperature flexibility of all elastomeric compounds. Not recommended for dynamic applications, due to low strength, or for use with most petroleum oils. Shrinkage characteristics similar to organic rubber, allowing existing molds to be used.
Polyurethane	Toughest of the elastomers used for O-rings, characterized by high tensile strength, excellent abrasion resistance, and tear strength. Compression set and heat resistance are inferior to nitrile. Suitable for hydraulic applications that anticipate abrasive contaminants and shock loads. Temperature service range of −65° to 212°F.
Fluorosilicone	Wide temperature range (−80° to 450°F) for continuous duty and excellent resistance to petroleum oils and fuels. Recommended for static applications only, due to limited strength and low abrasion resistance.
Polyacrylate	Heat resistance better than nitrile compounds, but inferior low temperature, compression set, and water resistance. Often used in power steering and transmission applications due to excellent resistance to oil, automatic transmission fluids, oxidation, and flex cracking. Temperature service range of −20° to 300°F.
Fluorocarbon (Viton)	General-purpose compound suitable for applications requiring resistance to aromatic or halogenated solvents or to high temperatures (−20° to 500°F with limited service to 600°F). Outstanding resistance to blended aromatic fuels, straight aromatics, and halogenated hydrocarbons and other petroleum products. Good resistance to strong acids (temperature range in acids (−20° to 250°F), but not effective for use with very hot water, steam, and brake fluids.

Ring Materials.—Thousands of O-ring compounds have been formulated for specific applications. Some of the most common types of compounds and their typical applications are given in Table 3. The Shore A durometer is the standard instrument used for measuring the hardness of elastomeric compounds. The softest O-rings are 50 and 60 Shore A and stretch more easily, exhibit lower breakout friction, seal better on rough surfaces, and need less clamping pressure than harder rings. For a given squeeze, the higher the durometer hardness of a ring, the greater the associated friction because a greater compressive force is exerted by hard rings than soft rings.

The most widely used rings are medium-hard O-rings with 70 Shore A hardness, which have the best wear resistance and frictional properties for running seals. Applications that involve oscillating or rotary motion frequently use 80 Shore A materials. Rings with a hardness above 85 Shore A often leak more because of less effective wiping action. These harder rings have a greater resistance to extrusion, but for small sizes may break easily during installation. O-ring hardness varies inversely with temperature, but when used for continuous service at high temperatures, the hardness may eventually increase after an initial softening of the compound.

O-ring compounds have thermal coefficients of expansion in the range of 7 to 20 times that of metal components, so shrinkage or expansion with temperature change can pose problems of leakage past the seal at low temperatures and excessive pressures at high temperatures when a ring is installed in a tight-fitting groove. Likewise, when an O-ring is immersed in a fluid, the compound usually absorbs some of the fluid and consequently increases in volume. Manufacturer's data give volumetric increase data for compounds completely immersed in various fluids. For confined rings (those with only a portion of the ring exposed to fluid), the size increase may be considerably lower than for rings completely immersed in fluid. Certain fluids can also cause ring shrinkage during “idle” periods, i.e., when the seal has a chance to dry out. If this shrinkage is more than 3 to 4 per cent, the seal may leak.

Excessive swelling due to fluid contact and high temperatures softens all compounds approximately 20 to 30 Shore A points from room temperature values and designs should anticipate the expected operating conditions. At low temperatures, swelling may be beneficial because fluid absorption may make the seal more flexible. However, the combination of low temperature and low pressure makes a seal particularly difficult to maintain. A soft compound should be used to provide a resilient seal at low temperatures. Below −65°F, only compounds formulated with silicone are useful; other compounds are simply too stiff, especially for use with air and other gases.

Compression set is another material property and a very important sealing factor. It is a measure of the shape memory of the material, that is, the ability to regain shape after being deformed. Compression set is a ratio, expressed as a percentage, of the unrecovered to original thickness of an O-ring compressed for a specified period of time between two heated plates and then released. O-rings with excessive compressive set will fail to maintain a good seal because, over time, the ring will be unable to exert the necessary compressive force (squeeze) on the enclosing walls. Swelling of the ring due to fluid contact tends to increase the squeeze and may partially compensate for the loss due to compression set. Generally, compression set varies by compound and ring cross-sectional diameter, and increases with the operating temperature.

ROLLED STEEL SECTIONS, WIRE AND SHEET-METAL GAGES

Rolled Steel Sections

Lengths of Angles Bent to Circular Shape.—To calculate the length of an angle-iron used either inside or outside of a tank or smokestack, the following table of constants may be used: Assume, for example, that a stand-pipe, 20 feet inside diameter, is provided with a 3 by 3 by $\frac{3}{8}$ inch angle-iron on the inside at the top. The circumference of a circle 20 feet in diameter is 754 inches. From the table of constants, find the constant for a 3 by 3 by $\frac{3}{8}$ inch angle-iron, which is 4.319. The length of the angle then is $754 - 4.319 = 749.681$ inches. Should the angle be on the outside, add the constant instead of subtracting it; thus, $754 + 4.319 = 758.319$ inches.

Size of Angle	Const.	Size of Angle	Const.	Size of Angle	Const.
$\frac{1}{4} \times 2 \times 2$	2.879	$\frac{5}{16} \times 3 \times 3$	4.123	$\frac{1}{2} \times 5 \times 5$	6.804
$\frac{5}{16} \times 2 \times 2$	3.076	$\frac{3}{8} \times 3 \times 3$	4.319	$\frac{3}{8} \times 6 \times 6$	7.461
$\frac{3}{8} \times 2 \times 2$	3.272	$\frac{1}{2} \times 3 \times 3$	4.711	$\frac{1}{2} \times 6 \times 6$	7.854
$\frac{1}{4} \times 2\frac{1}{2} \times 2\frac{1}{2}$	3.403	$\frac{3}{8} \times 3\frac{1}{2} \times 3\frac{1}{2}$	4.843	$\frac{3}{4} \times 6 \times 6$	8.639
$\frac{5}{16} \times 2\frac{1}{2} \times 2\frac{1}{2}$	3.600	$\frac{1}{2} \times 3\frac{1}{2} \times 3\frac{1}{2}$	5.235	$\frac{1}{2} \times 8 \times 8$	9.949
$\frac{3}{8} \times 2\frac{1}{2} \times 2\frac{1}{2}$	3.796	$\frac{3}{8} \times 4 \times 4$	5.366	$\frac{3}{4} \times 8 \times 8$	10.734
$\frac{1}{2} \times 2\frac{1}{2} \times 2\frac{1}{2}$	4.188	$\frac{1}{2} \times 4 \times 4$	5.758	$1 \times 8 \times 8$	11.520
$\frac{1}{4} \times 3 \times 3$	3.926	$\frac{3}{8} \times 5 \times 5$	6.414	…	…

Standard Designations of Rolled Steel Shapes.—Through a joint effort, the American Iron and Steel Institute (AISI) and the American Institute of Steel Construction (AISC) have changed most of the designations for their hot-rolled structural steel shapes. The present designations, standard for steel producing and fabricating industries, should be used when designing, detailing, and ordering steel. The accompanying table compares the present designations with the previous descriptions.

Hot-Rolled Structural Steel Shape Designations (AISI and AISC)

Present Designation	Type of Shape	Previous Designation
W 24 × 76	W shape	24 WF 76
W 14 × 26	W shape	14 B 26
S 24 × 100	S shape	24 I 100
M 8 × 18.5	M shape	8 M 18.5
M 10 × 9	M shape	10 JR 9.0
M 8 × 34.3	M shape	8 × 8 M 34.3
C 12 × 20.7	American Standard Channel	12 [20.7
MC 12 × 45	Miscellaneous Channel	12 × 4 [45.0
MC 12 × 10.6	Miscellaneous Channel	12 JR [10.6
HP 14 × 73	HP shape	14 BP 73
L 6 × 6 × $\frac{3}{4}$	Equal Leg Angle	∠ 6 × 6 × $\frac{3}{4}$
L 6 × 4 × $\frac{5}{8}$	Unequal Leg Angle	∠ 6 × 4 × $\frac{5}{8}$
WT 12 × 38	Structural Tee cut from W shape	ST 12 WF 38
WT 7 × 13	Structural Tee cut from W shape	ST 7 B 13
St 12 × 50	Structural Tee cut from S shape	ST 12 I 50
MT 4 × 9.25	Structural Tee cut from M shape	ST 4 M 9.25
MT 5 × 4.5	Structural Tee cut from M shape	ST 5 JR 4.5
MT 4 × 17.15	Structural Tee cut from M shape	ST 4 M 17.15
PL $\frac{1}{2}$ × 18	Plate	PL 18 × $\frac{1}{2}$
Bar 1	Square Bar	Bar 1
Bar 1 $\frac{1}{4}$ ∅	Round Bar	Bar 1$\frac{1}{4}$∅
Bar 2$\frac{1}{2}$ × $\frac{1}{2}$	Flat Bar	Bar 2$\frac{1}{2}$ × $\frac{1}{2}$
Pipe 4 Std.	Pipe	Pipe 4 Std.
Pipe 4 X-Strong	Pipe	Pipe 4 X-Strong
Pipe 4 XX-Strong	Pipe	Pipe 4 XX-Strong
TS 4 × 4 × .375	Structural Tubing: Square	Tube 4 × 4 × .375
TS 5 × 3 × .375	Structural Tubing: Rectangular	Tube 5 × 3 × .375
TS 3 OD × .250	Structural Tubing: Circular	Tube 3 OD × .250

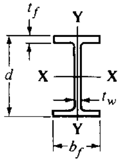
Data taken from the “Manual of Steel Construction,” 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Wide-Flange Sections—1

Wide-flange sections are designated, in order, by a section letter, nominal depth of the member in inches, and the nominal weight in pounds per foot; thus:

W 18 × 64

indicates a wide-flange section having a nominal depth of 18 inches, and a nominal weight per foot of 64 pounds. Actual geometry for each section can be obtained from the values below.



			Flange		Web	Axis X-X			Axis Y-Y		
	Area, A	Depth, d	Width, b_f	Thick-ness, t_f	Thick-ness, t_w	I	S	r	I	S	r
Designation	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
[a]W 27 × 178	52.3	27.81	14.085	1.190	0.725	6990	502	11.6	555	78.8	3.26
× 161	47.4	27.59	14.020	1.080	0.660	6280	455	11.5	497	70.9	3.24
× 146	42.9	27.38	13.965	0.975	0.605	5630	411	11.4	443	63.5	3.21
× 114	33.5	27.29	10.070	0.930	0.570	4090	299	11.0	159	31.5	2.18
× 102	30.0	27.09	10.015	0.830	0.515	3620	267	11.0	139	27.8	2.15
× 94	27.7	26.92	9.990	0.745	0.490	3270	243	10.9	124	24.8	2.12
× 84	24.8	26.71	9.960	0.640	0.460	2850	213	10.7	106	21.2	2.07
W 24 × 162	47.7	25.00	12.955	1.220	0.705	5170	414	10.4	443	68.4	3.05
× 146	43.0	24.74	12.900	1.090	0.650	4580	371	10.3	391	60.5	3.01
× 131	38.5	24.48	12.855	0.960	0.605	4020	329	10.2	340	53.0	2.97
× 117	34.4	24.26	12.800	0.850	0.550	3540	291	10.1	297	46.5	2.94
× 104	30.6	24.06	12.750	0.750	0.500	3100	258	10.1	259	40.7	2.91
× 94	27.7	24.31	9.065	0.875	0.515	2700	222	9.87	109	24.0	1.98
× 84	24.7	24.10	9.020	0.770	0.470	2370	196	9.79	94.4	20.9	1.95
× 76	22.4	23.92	8.990	0.680	0.440	2100	176	9.69	82.5	18.4	1.92
× 68	20.1	23.73	8.965	0.585	0.415	1830	154	9.55	70.4	15.7	1.87
× 62	18.2	23.74	7.040	0.590	0.430	1550	131	9.23	34.5	9.80	1.38
× 55	16.2	23.57	7.005	0.505	0.395	1350	114	9.11	29.1	8.30	1.34
W 21 × 147	43.2	22.06	12.510	1.150	0.720	3630	329	9.17	376	60.1	2.95
× 132	38.8	21.83	12.440	1.035	0.650	3220	295	9.12	333	53.5	2.93
× 122	35.9	21.68	12.390	0.960	0.600	2960	273	9.09	305	49.2	2.92
× 111	32.7	21.51	12.340	0.875	0.550	2670	249	9.05	274	44.5	2.90
× 101	29.8	21.36	12.290	0.800	0.500	2420	227	9.02	248	40.3	2.89
× 93	27.3	21.62	8.420	0.930	0.580	2070	192	8.70	92.9	22.1	1.84
× 83	24.3	21.43	8.355	0.835	0.515	1830	171	8.67	81.4	19.5	1.83
× 73	21.5	21.24	8.295	0.740	0.455	1600	151	8.64	70.6	17.0	1.81
× 68	20.0	21.13	8.270	0.685	0.430	1480	140	8.60	64.7	15.7	1.80
× 62	18.3	20.99	8.240	0.615	0.400	1330	127	8.54	57.5	13.9	1.77
× 57	16.7	21.06	6.555	0.650	0.405	1170	111	8.36	30.6	9.35	1.35
× 50	14.7	20.83	6.530	0.535	0.380	984	94.5	8.18	24.9	7.64	1.30
× 44	13.0	20.66	6.500	0.450	0.350	843	81.6	8.06	20.7	6.36	1.26
W 18 × 119	35.1	18.97	11.265	1.060	0.655	2190	231	7.90	253	44.9	2.69
× 106	31.1	18.73	11.200	0.940	0.590	1910	204	7.84	220	39.4	2.66
× 97	28.5	18.59	11.145	0.870	0.535	1750	188	7.82	201	36.1	2.65
× 86	25.3	18.39	11.090	0.770	0.480	1530	166	7.77	175	31.6	2.63
× 76	22.3	18.21	11.035	0.680	0.425	1330	146	7.73	152	27.6	2.61
× 71	20.8	18.47	7.635	0.810	0.495	1170	127	7.50	60.3	15.8	1.70
× 65	19.1	18.35	7.590	0.750	0.450	1070	117	7.49	54.8	14.4	1.69
× 60	17.6	18.24	7.555	0.695	0.415	984	108	7.47	50.1	13.3	1.69
× 55	16.2	18.11	7.530	0.630	0.390	890	98.3	7.41	44.9	11.9	1.67
× 50	14.7	17.99	7.495	0.570	0.355	800	88.9	7.38	40.1	10.7	1.65
× 46	13.5	18.06	6.060	0.605	0.360	712	78.8	7.25	22.5	7.43	1.29
× 40	11.8	17.90	6.015	0.525	0.315	612	68.4	7.21	19.1	6.35	1.27
× 35	10.3	17.70	6.000	0.425	0.300	510	57.6	7.04	15.3	5.12	1.22

[a] Consult the AISC Manual, noted above, for W steel shapes having nominal depths greater than 27 in.

Symbols: I = moment of inertia; S = section modulus; r = radius of gyration.

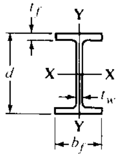
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Wide-Flange Sections–2

Wide-flange sections are designated, in order, by a section letter, nominal depth of the member in inches, and the nominal weight in pounds per foot; thus:

W 16 × 78

indicates a wide-flange section having a nominal depth of 16 inches, and a nominal weight per foot of 78 pounds. Actual geometry for each section can be obtained from the values below.



Designation	Area, A	Depth, d	Flange Width, b_f	Flange Thickness, t_f	Web Thickness, t_w	Axis X-X I	Axis X-X S	Axis X-X r	Axis Y-Y I	Axis Y-Y S	Axis Y-Y r
	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
W 16 × 100	29.4	16.97	10.425	0.985	0.585	1490	175	7.10	186	35.7	2.51
× 89	26.2	16.75	10.365	0.875	0.525	1300	155	7.05	163	31.4	2.49
× 77	22.6	16.52	10.295	0.760	0.455	1110	134	7.00	138	26.9	2.47
× 67	19.7	16.33	10.235	0.665	0.395	954	117	6.96	119	23.2	2.46
× 57	16.8	16.43	7.120	0.715	0.430	758	92.2	6.72	43.1	12.1	1.60
× 50	14.7	16.26	7.070	0.630	0.380	659	81.0	6.68	37.2	10.5	1.59
× 45	13.3	16.13	7.035	0.565	0.345	586	72.7	6.65	32.8	9.34	1.57
× 40	11.8	16.01	6.995	0.505	0.305	518	64.7	6.63	28.9	8.25	1.57
× 36	10.6	15.86	6.985	0.430	0.295	448	56.5	6.51	24.5	7.00	1.52
× 31	9.12	15.88	5.525	0.440	0.275	375	47.2	6.41	12.4	4.49	1.17
× 26	7.68	15.69	5.500	0.345	0.250	301	38.4	6.26	9.59	3.49	1.12
W 14 × 730	215.0	22.42	17.890	4.910	3.070	14300	1280	8.17	4720	527	4.69
× 665	196.0	21.64	17.650	4.520	2.830	12400	1150	7.98	4170	472	4.62
× 605	178.0	20.92	17.415	4.160	2.595	10800	1040	7.80	3680	423	4.55
× 550	162.0	20.24	17.200	3.820	2.380	9430	931	7.63	3250	378	4.49
× 500	147.0	19.60	17.010	3.500	2.190	8210	838	7.48	2880	339	4.43
× 455	134.0	19.02	16.835	3.210	2.015	7190	756	7.33	2560	304	4.38
× 426	125.0	18.67	16.695	3.035	1.875	6600	707	7.26	2360	283	4.34
× 398	117.0	18.29	16.590	2.845	1.770	6000	656	7.16	2170	262	4.31
× 370	109.0	17.92	16.475	2.660	1.655	5440	607	7.07	1990	241	4.27
× 342	101.0	17.54	16.360	2.470	1.540	4900	559	6.98	1810	221	4.24
× 311	91.4	17.12	16.230	2.260	1.410	4330	506	6.88	1610	199	4.20
× 283	83.3	16.74	16.110	2.070	1.290	3840	459	6.79	1440	179	4.17
× 257	75.6	16.38	15.995	1.890	1.175	3400	415	6.71	1290	161	4.13
× 233	68.5	16.04	15.890	1.720	1.070	3010	375	6.63	1150	145	4.10
× 211	62.0	15.72	15.800	1.560	0.980	2660	338	6.55	1030	130	4.07
× 193	56.8	15.48	15.710	1.440	0.890	2400	310	6.50	931	119	4.05
× 176	51.8	15.22	15.650	1.310	0.830	2140	281	6.43	838	107	4.02
× 159	46.7	14.98	15.565	1.190	0.745	1900	254	6.38	748	96.2	4.00
× 145	42.7	14.78	15.500	1.090	0.680	1710	232	6.33	677	87.3	3.98

Symbols: I = moment of inertia; S = section modulus; r = radius of gyration.

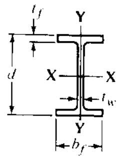
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Wide-Flange Sections—3

Wide-flange sections are designated, in order, by a section letter, nominal depth of the member in inches, and the nominal weight in pounds per foot; thus:

W 14 × 38

indicates a wide-flange section having a nominal depth of 14 inches, and a nominal weight per foot of 38 pounds. Actual geometry for each section can be obtained from the values below.



Designation	Area, A	Depth, d	Flange Width, b_f	Flange Thickness, t_f	Web Thickness, t_w	Axis X-X I	Axis X-X S	Axis X-X r	Axis Y-Y I	Axis Y-Y S	Axis Y-Y r
	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
W 14 × 132	38.8	14.66	14.725	1.030	0.645	1530	209	6.28	548	74.5	3.76
× 120	35.3	14.48	14.670	0.940	0.590	1380	190	6.24	495	67.5	3.74
× 109	32.0	14.32	14.605	0.860	0.525	1240	173	6.22	447	61.2	3.73
× 99	29.1	14.16	14.565	0.780	0.485	1110	157	6.17	402	55.2	3.71
× 90	26.5	14.02	14.520	0.710	0.440	999	143	6.14	362	49.9	3.70
× 82	24.1	14.31	10.130	0.855	0.510	882	123	6.05	148	29.3	2.48
× 74	21.8	14.17	10.070	0.785	0.450	796	112	6.04	134	26.6	2.48
×68	20.0	14.04	10.035	0.720	0.415	723	103	6.01	121	24.2	2.46
× 61	17.9	13.89	9.995	0.645	0.375	640	92.2	5.98	107	21.5	2.45
× 53	15.6	13.92	8.060	0.660	0.370	541	77.8	5.89	57.7	14.3	1.92
× 48	14.1	13.79	8.030	0.595	0.340	485	70.3	5.85	51.4	12.8	1.91
× 43	12.6	13.66	7.995	0.530	0.305	428	62.7	5.82	45.2	11.3	1.89
× 38	11.2	14.10	6.770	0.515	0.310	385	54.6	5.87	26.7	7.88	1.55
× 34	10.0	13.98	6.745	0.455	0.285	340	48.6	5.83	23.3	6.91	1.53
× 30	8.85	13.84	6.730	0.385	0.270	291	42.0	5.73	19.6	5.82	1.49
× 26	7.69	13.91	5.025	0.420	0.255	245	35.3	5.65	8.91	3.54	1.08
× 22	6.49	13.74	5.000	0.335	0.230	199	29.0	5.54	7.00	2.80	1.04
W 12 × 336	98.8	16.82	13.385	2.955	1.775	4060	483	6.41	1190	177	3.47
× 305	89.6	16.32	13.235	2.705	1.625	3550	435	6.29	1050	159	3.42
× 279	81.9	15.85	13.140	2.470	1.530	3110	393	6.16	937	143	3.38
× 252	74.1	15.41	13.005	2.250	1.395	2720	353	6.06	828	127	3.34
× 230	67.7	15.05	12.895	2.070	1.285	2420	321	5.97	742	115	3.31
× 210	61.8	14.71	12.790	1.900	1.180	2140	292	5.89	664	104	3.28
× 190	55.8	14.38	12.670	1.735	1.060	1890	263	5.82	589	93.0	3.25
× 170	50.0	14.03	12.570	1.560	0.960	1650	235	5.74	517	82.3	3.22
× 152	44.7	13.71	12.480	1.400	0.870	1430	209	5.66	454	72.8	3.19
× 136	39.9	13.41	12.400	1.250	0.790	1240	186	5.58	398	64.2	3.16
× 120	35.3	13.12	12.320	1.105	0.710	1070	163	5.51	345	56.0	3.13
× 106	31.2	12.89	12.220	0.990	0.610	933	145	5.47	301	49.3	3.11
× 96	28.2	12.71	12.160	0.900	0.550	833	131	5.44	270	44.4	3.09
× 87	25.6	12.53	12.125	0.810	0.515	740	118	5.38	241	39.7	3.07
× 79	23.2	12.38	12.080	0.735	0.470	662	107	5.34	216	35.8	3.05
× 72	21.1	12.25	12.040	0.670	0.430	597	97.4	5.31	195	32.4	3.04
× 65	19.1	12.12	12.000	0.605	0.390	533	87.9	5.28	174	29.1	3.02
× 58	17.0	12.19	10.010	0.640	0.360	475	78.0	5.28	107	21.4	2.51
× 53	15.6	12.06	9.995	0.575	0.345	425	70.6	5.23	95.8	19.2	2.48
× 50	14.7	12.19	8.080	0.640	0.370	394	64.7	5.18	56.3	13.9	1.96
× 45	13.2	12.06	8.045	0.575	0.335	350	58.1	5.15	50.0	12.4	1.94
× 40	11.8	11.94	8.005	0.515	0.295	310	51.9	5.13	44.1	11.0	1.93
× 35	10.3	12.50	6.560	0.520	0.300	285	45.6	5.25	24.5	7.47	1.54
× 30	8.79	12.34	6.520	0.440	0.260	238	38.6	5.21	20.3	6.24	1.52
× 26	7.65	12.22	6.490	0.380	0.230	204	33.4	5.17	17.3	5.34	1.51
× 22	6.48	12.31	4.030	0.425	0.260	156	25.4	4.91	4.66	2.31	0.847
× 19	5.57	12.16	4.005	0.350	0.235	130	21.3	4.82	3.76	1.88	0.822
× 16	4.71	11.99	3.990	0.265	0.220	103	17.1	4.67	2.82	1.41	0.773
× 14	4.16	11.91	3.970	0.225	0.200	88.6	14.9	4.62	2.36	1.19	0.753

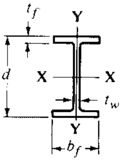
Data taken from the “Manual of Steel Construction,” 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Wide-Flange Sections—4

Wide-flange sections are designated, in order, by a section letter, nominal depth of the member in inches, and the nominal weight in pounds per foot; thus:

W 8 × 67

indicates a wide-flange section having a nominal depth of 8 inches, and a nominal weight per foot of 67 pounds. Actual geometry for each section can be obtained from the values below.



			Flange		Web	Axis X-X			Axis Y-Y		
	Area, A	Depth, d	Width, b_f	Thickness, t_f	Thickness, t_w	I	S	r	I	S	r
Designation	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
W 10 × 112	32.9	11.36	10.415	1.250	0.755	716	126	4.66	236	45.3	2.68
× 100	29.4	11.10	10.340	1.120	0.680	623	112	4.60	207	40.0	2.65
× 88	25.9	10.84	10.265	0.990	0.605	534	98.5	4.54	179	34.8	2.63
× 77	22.6	10.60	10.190	0.870	0.530	455	85.9	4.49	154	30.1	2.60
× 68	20.0	10.40	10.130	0.770	0.470	394	75.7	4.44	134	26.4	2.59
× 60	17.6	10.22	10.080	0.680	0.420	341	66.7	4.39	116	23.0	2.57
× 54	15.8	10.09	10.030	0.615	0.370	303	60.0	4.37	103	20.6	2.56
× 49	14.4	9.98	10.000	0.560	0.340	272	54.6	4.35	93.4	18.7	2.54
× 45	13.3	10.10	8.020	0.620	0.350	248	49.1	4.32	53.4	13.3	2.01
× 39	11.5	9.92	7.985	0.530	0.315	209	42.1	4.27	45.0	11.3	1.98
× 33	9.71	9.73	7.960	0.435	0.290	170	35.0	4.19	36.6	9.20	1.94
× 30	8.84	10.47	5.810	0.510	0.300	170	32.4	4.38	16.7	5.75	1.37
× 26	7.61	10.33	5.770	0.440	0.260	144	27.9	4.35	14.1	4.89	1.36
× 22	6.49	10.17	5.750	0.360	0.240	118	23.2	4.27	11.4	3.97	1.33
× 19	5.62	10.24	4.020	0.395	0.250	96.3	18.8	4.14	4.29	2.14	0.874
× 17	4.99	10.11	4.010	0.330	0.240	81.9	16.2	4.05	3.56	1.78	0.844
× 15	4.41	9.99	4.000	0.270	0.230	68.9	13.8	3.95	2.89	1.45	0.810
× 12	3.54	9.87	3.960	0.210	0.190	53.8	10.9	3.90	2.18	1.10	0.785
W 8 × 67	19.7	9.00	8.280	0.935	0.570	272	60.4	3.72	88.6	21.4	2.12
× 58	17.1	8.75	8.220	0.810	0.510	228	52.0	3.65	75.1	18.3	2.10
× 48	14.1	8.50	8.110	0.685	0.400	184	43.3	3.61	60.9	15.0	2.08
× 40	11.7	8.25	8.070	0.560	0.360	146	35.5	3.53	49.1	12.2	2.04
× 35	10.3	8.12	8.020	0.495	0.310	127	31.2	3.51	42.6	10.6	2.03
× 31	9.13	8.00	7.995	0.435	0.285	110	27.5	3.47	37.1	9.27	2.02
× 28	8.25	8.06	6.535	0.465	0.285	98.0	24.3	3.45	21.7	6.63	1.62
× 24	7.08	7.93	6.495	0.400	0.245	82.8	20.9	3.42	18.3	5.63	1.61
× 21	6.16	8.28	5.270	0.400	0.250	75.3	18.2	3.49	9.77	3.71	1.26
× 18	5.26	8.14	5.250	0.330	0.230	61.9	15.2	3.43	7.97	3.04	1.23
× 15	4.44	8.11	4.015	0.315	0.245	48.0	11.8	3.29	3.41	1.70	0.876
× 13	3.84	7.99	4.000	0.255	0.230	39.6	9.91	3.21	2.73	1.37	0.843
× 10	2.96	7.89	3.940	0.205	0.170	30.8	7.81	3.22	2.09	1.06	0.841
W 6 × 25	7.34	6.38	6.080	0.455	0.320	53.4	16.7	2.70	17.1	5.61	1.52
× 20	5.87	6.20	6.020	0.365	0.260	41.4	13.4	2.66	13.3	4.41	1.50
× 16	4.74	6.28	4.030	0.405	0.260	32.1	10.2	2.60	4.43	2.20	0.966
× 15	4.43	5.99	5.990	0.260	0.230	29.1	9.72	2.56	9.32	3.11	1.46
× 12	3.55	6.03	4.000	0.280	0.230	22.1	7.31	2.49	2.99	1.50	0.918
× 9	2.68	5.90	3.940	0.215	0.170	16.4	5.56	2.47	2.19	1.11	0.905
W 5 × 19	5.54	5.15	5.030	0.430	0.270	26.2	10.2	2.17	9.13	3.63	1.28
× 16	4.68	5.01	5.000	0.360	0.240	21.3	8.51	2.13	7.51	3.00	1.27
W 4 × 13	3.83	4.16	4.060	0.345	0.280	11.3	5.46	1.72	3.86	1.90	1.00

Symbols: I = moment of inertia; S = section modulus; r = radius of gyration.

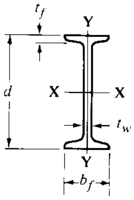
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel S Sections

"S" is the section symbol for "I" Beams. S shapes are designated, in order, by their section letter, actual depth in inches, and nominal weight in pounds per foot. Thus:

S 5 × 14.75

indicates an S shape (or I beam) having a depth of 5 inches and a nominal weight of 14.75 pounds per foot.



			Flange		Web	Axis-X-X			Axis Y-Y		
	Area A	Depth, d	Width, b_f	Thick-ness, t_f	Thick-ness, t_w	I	S	r	I	S	r
Designation	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
S 24 × 121	35.6	24.50	8.050	1.090	0.800	3160	258	9.43	83.3	20.7	1.53
× 106	31.2	24.50	7.870	1.090	0.620	2940	240	9.71	77.1	19.6	1.57
×100	29.3	24.00	7.245	0.870	0.745	2390	199	9.02	47.7	13.2	1.27
× 90	26.5	24.00	7.125	0.870	0.625	2250	187	9.21	44.9	12.6	1.30
× 80	23.5	24.00	7.000	0.870	0.500	2100	175	9.47	42.2	12.1	1.34
S 20 × 96	28.2	20.30	7.200	0.920	0.800	1670	165	7.71	50.2	13.9	1.33
× 86	25.3	20.30	7.060	0.920	0.660	1580	155	7.89	46.8	13.3	1.36
× 75	22.0	20.00	6.385	0.795	0.635	1280	128	7.62	29.8	9.32	1.16
× 66	19.4	20.00	6.255	0.795	0.505	1190	119	7.83	27.7	8.85	1.19
S 18 × 70	20.6	18.00	6.251	0.691	0.711	926	103	6.71	24.1	7.72	1.08
× 54.7	16.1	18.00	6.001	0.691	0.461	804	89.4	7.07	20.8	6.94	1.14
S 15 × 50	14.7	15.00	5.640	0.622	0.550	486	64.8	5.75	15.7	5.57	1.03
× 42.9	12.6	15.00	5.501	0.622	0.411	447	59.6	5.95	14.4	5.23	1.07
S 12 × 50	14.7	12.00	5.477	0.659	0.687	305	50.8	4.55	15.7	5.74	1.03
× 40.8	12.0	12.00	5.252	0.659	0.462	272	45.4	4.77	13.6	5.16	1.06
× 35	10.3	12.00	5.078	0.544	0.428	229	38.2	4.72	9.87	3.89	0.980
× 31.8	9.35	12.00	5.000	0.544	0.350	218	36.4	4.83	9.36	3.74	1.00
S 10 × 35	10.3	10.00	4.944	0.491	0.594	147	29.4	3.78	8.36	3.38	0.901
× 25.4	7.46	10.00	4.661	0.491	0.311	124	24.7	4.07	6.79	2.91	0.954
S 8 × 23	6.77	8.00	4.171	0.426	0.441	64.9	16.2	3.10	4.31	2.07	0.798
× 18.4	5.41	8.00	4.001	0.426	0.271	57.6	14.4	3.26	3.73	1.86	0.831
S 7 × 20	5.88	7.00	3.860	0.392	0.450	42.4	12.1	2.69	3.17	1.64	0.734
× 15.3	4.50	7.00	3.662	0.392	0.252	36.7	10.5	2.86	2.64	1.44	0.766
S 6 × 17.25	5.07	6.00	3.565	0.359	0.465	26.3	8.77	2.28	2.31	1.30	0.675
× 12.5	3.67	6.00	3.332	0.359	0.232	22.1	7.37	2.45	1.82	1.09	0.705
S 5 × 14.75	4.34	5.00	3.284	0.326	0.494	15.2	6.09	1.87	1.67	1.01	0.620
× 10	2.94	5.00	3.004	0.326	0.214	12.3	4.92	2.05	1.22	0.809	0.643
S 4 × 9.5	2.79	4.00	2.796	0.293	0.326	6.79	3.39	1.56	0.903	0.646	0.569
× 7.7	2.26	4.00	2.663	0.293	0.193	6.08	3.04	1.64	0.764	0.574	0.581
S 3 × 7.5	2.21	3.00	2.509	0.260	0.349	2.93	1.95	1.15	0.586	0.468	0.516
× 5.7	1.67	3.00	2.330	0.260	0.170	2.52	1.68	1.23	0.455	0.390	0.522

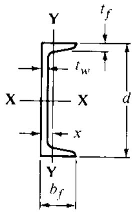
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

American Standard Steel Channels

American Standard Channels are designated, in order, by a section letter, actual depth in inches, and nominal weight per foot in pounds. Thus:

C 7 × 14.75

indicates an American Standard Channel with a depth of 7 inches and a nominal weight of 14.75 pounds per foot.



Designation	Area, A	Depth, d	Flange Width, b_f	Flange Thick-ness, t_f	Web Thick-ness, t_w	Axis X-X I	Axis X-X S	Axis X-X r	Axis Y-Y I	Axis Y-Y S	Axis Y-Y r	x
	in.^{2}	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.	in.
C 15 × 50	14.7	15.00	3.716	0.650	0.716	404	53.8	5.24	11.0	3.78	0.867	0.798
× 40	11.8	15.00	3.520	0.650	0.520	349	46.5	5.44	9.23	3.37	0.886	0.777
× 33.9	9.96	15.00	3.400	0.650	0.400	315	42.0	5.62	8.13	3.11	0904	0.787
C 12 × 30	8.82	12.00	3.170	0.501	0.510	162	27.0	4.29	5.14	2.06	0.763	0.674
× 25	7.35	12.00	3.047	0.501	0.387	144	24.1	4.43	4.47	1.88	0.780	0.674
× 20.7	6.09	12.00	2.942	0.501	0.282	129	21.5	4.61	3.88	1.73	0.799	0.698
C 10 × 30	8.82	10.00	3.033	0.436	0.673	103	20.7	3.42	3.94	1.65	0.669	0.649
× 25	7.35	10.00	2.886	0.436	0.526	91.2	18.2	3.52	3.36	1.48	0.676	0.617
× 20	5.88	10.00	2.739	0.436	0.379	78.9	15.8	3.66	2.81	1.32	0.692	0.606
× 15.3	4.49	10.00	2.600	0.436	0.240	67.4	13.5	3.87	2.28	1.16	0.713	0.634
C 9 × 20	5.88	9.00	2.648	0.413	0.448	60.9	13.5	3.22	2.42	1.17	0.642	0.583
× 15	4.41	9.00	2.485	0.413	0.285	51.0	11.3	3.40	1.93	1.01	0.661	0.586
× 13.4	3.94	9.00	2.433	0.413	0.233	47.9	10.6	3.48	1.76	0.962	0.669	0.601
C 8 × 18.75	5.51	8.00	2.527	0.390	0.487	44.0	11.0	2.82	1.98	1.01	0.599	0.565
× 13.75	4.04	8.00	2.343	0.390	0.303	36.1	9.03	2.99	1.53	0.854	0.615	0.553
× 11.5	3.38	8.00	2.260	0.390	0.220	32.6	8.14	3.11	1.32	0.781	0.625	0.571
C 7 × 14.75	4.33	7.00	2.299	0.366	0.419	27.2	7.78	2.51	1.38	0.779	0.564	0.532
× 12.25	3.60	7.00	2.194	0.366	0.314	24.2	6.93	2.60	1.17	0.703	0.571	0.525
× 9.8	2.87	7.00	2.090	0.366	0.210	21.3	6.08	2.72	0.968	0.625	0.581	0.540
C 6 × 13	3.83	6.00	2.157	0.343	0.437	17.4	5.80	2.13	1.05	0.642	0.525	0.514
× 10.5	3.09	6.00	2.034	0.343	0.314	15.2	5.06	2.22	0.866	0.564	0.529	0.499
× 8.2	2.40	6.00	1.920	0.343	0.200	13.1	4.38	2.34	0.693	0.492	0.537	0.511
C 5 × 9	2.64	5.00	1.885	0.320	0.325	8.90	3.56	1.83	0.632	0.450	0.489	0.478
× 6.7	1.97	5.00	1.750	0.320	0.190	7.49	3.00	1.95	0.479	0.378	0.493	0.484
C 4 × 7.25	2.13	4.00	1.721	0.296	0.321	4.59	2.29	1.47	0.433	0.343	0.450	0.459
× 5.4	1.59	4.00	1.584	0.296	0.184	3.85	1.93	1.56	0.319	0.283	0.449	0.457
C 3 × 6	1.76	3.00	1.596	0.273	0.356	2.07	1.38	1.08	0.305	0.268	0.416	0.455
× 5	1.47	3.00	1.498	0.273	0.258	1.85	1.24	1.12	0.247	0.233	0.410	0.438
× 4.1	1.21	3.00	1.410	0.273	0.170	1.66	1.10	1.17	0.197	0.202	0.404	0.436

Symbols: I = moment of inertia; S = section modulus; r = radius of gyration; x = distance from center of gravity of section to outer face of structural shape.

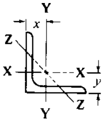
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Angles with Equal Legs

These angles are commonly designated by section symbol, width of each leg, and thickness, thus:

L 3 × 3 × 1/4

indicates a 3 × 3-inch angle of 1/4-inch thickness.



Size	Thickness	Weight per Foot	Area	Axis X-X & Y-Y			Z-Z
				I	*r*	*x* or *y*	r
in.	in.	lb.	$in.^2$	$in.^4$	in.	in.	in.
8 × 8	1 1/8	56.9	16.7	98.0	2.42	2.41	1.56
	1	51.0	15.0	89.0	2.44	2.37	1.56
	7/8	45.0	13.2	79.6	2.45	2.32	1.57
	3/4	38.9	11.4	69.7	2.47	2.28	1.58
	5/8	32.7	9.61	59.4	2.49	2.23	1.58
	9/16	29.6	8.68	54.1	2.50	2.21	1.59
	1/2	26.4	7.75	48.6	2.50	2.19	1.59
6 × 6	1	37.4	11.00	35.5	1.80	1.86	1.17
	7/8	33.1	9.73	31.9	1.81	1.82	1.17
	3/4	28.7	8.44	28.2	1.83	1.78	1.17
	5/8	24.2	7.11	24.2	1.84	1.73	1.18
	9/16	21.9	6.43	22.1	1.85	1.71	1.18
	1/2	19.6	5.75	19.9	1.86	1.68	1.18
	7/16	17.2	5.06	17.7	1.87	1.66	1.19
	3/8	14.9	4.36	15.4	1.88	1.64	1.19
	5/16	12.4	3.65	13.0	1.89	1.62	1.20
5 × 5	7/8	27.2	7.98	17.8	1.49	1.57	.973
	3/4	23.6	6.94	15.7	1.51	1.52	.975
	5/8	20.0	5.86	13.6	1.52	1.48	.978
	1/2	16.2	4.75	11.3	1.54	1.43	.983
	7/16	14.3	4.18	10.0	1.55	1.41	.986
	3/8	12.3	3.61	8.74	1.56	1.39	.990
	5/16	10.3	3.03	7.42	1.57	1.37	.994
4 × 4	3/4	18.5	5.44	7.67	1.19	1.27	.778
	5/8	15.7	4.61	6.66	1.20	1.23	.779
	1/2	12.8	3.75	5.56	1.22	1.18	.782
	7/16	11.3	3.31	4.97	1.23	1.16	.785
	3/8	9.8	2.86	4.36	1.23	1.14	.788
	5/16	8.2	2.40	3.71	1.24	1.12	.791
	1/4	6.6	1.94	3.04	1.25	1.09	.795
3 1/2 × 3 1/2	1/2	11.1	3.25	3.64	1.06	1.06	.683
	7/16	9.8	2.87	3.26	1.07	1.04	.684
	3/8	8.5	2.48	2.87	1.07	1.01	.687
	5/16	7.2	2.09	2.45	1.08	.990	.690
	1/4	5.8	1.69	2.01	1.09	.968	.694
3 × 3	1/2	9.4	2.75	2.22	.898	.932	.584
	7/16	8.3	2.43	1.99	.905	.910	.585
	3/8	7.2	2.11	1.76	.913	.888	.587
	5/16	6.1	1.78	1.51	.922	.865	.589
	1/4	4.9	1.44	1.24	.930	.842	.592
	3/16	3.71	1.09	.962	.939	.820	.596
2 1/2 × 2 1/2	1/2	7.7	2.25	1.23	.739	.806	.487
	3/8	5.9	1.73	.984	.753	.762	.487
	5/16	5.0	1.46	.849	.761	.740	.489
	1/4	4.1	1.19	.703	.769	.717	.491
	3/16	3.07	.902	.547	.778	.694	.495
2 × 2	3/8	4.7	1.36	.479	.594	.636	.389
	5/16	3.92	1.15	.416	.601	.614	.390
	1/4	3.19	.938	.348	.609	.592	.391
	3/16	2.44	.715	.272	.617	.569	.394
	1/8	1.65	.484	.190	.626	.546	.398

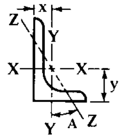
Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Steel Angles with Unequal Legs

These angles are commonly designated by section symbol, width of each leg, and thickness, thus:

L 7 × 4 × ½

indicates a 7 × 4-inch angle of ½-inch thickness.



Size	Thick-ness	Weight per Ft.	Area	Axis-X-X				Axis Y-Y			Axis Z-Z		
				I	S	r	y	I	S	r	x	r	Tan A
in.	in.	lb.	in.^{2}	in.^{4}	in.^{3}	in.	in.	in.^{4}	in.^{3}	in.	in.	in.	
9 × 4	⅝	26.3	7.73	64.9	11.5	2.90	3.36	8.32	2.65	1.04	.858	.847	.216
	9⁄16	23.8	7.00	59.1	10.4	2.91	3.33	7.63	2.41	1.04	.834	.850	.218
	½	21.3	6.25	53.2	9.34	2.92	3.31	6.92	2.17	1.05	.810	.854	.220
8 × 6	1	44.2	13.0	80.8	15.1	2.49	2.65	38.8	8.92	1.73	1.65	1.28	.543
	⅞	39.1	11.5	72.3	13.4	2.51	2.61	34.9	7.94	1.74	1.61	1.28	.547
	¾	33.8	9.94	63.4	11.7	2.53	2.56	30.7	6.92	1.76	1.56	1.29	.551
	⅝	28.5	8.36	54.1	9.87	2.54	2.52	26.3	5.88	1.77	1.52	1.29	.554
	9⁄16	7	7.56	49.3	8.95	2.55	2.50	24.0	5.34	1.78	1.50	1.30	.556
	½	23.0	6.75	44.3	8.02	2.56	2.47	21.7	4.79	1.79	1.47	1.30	.558
	7⁄16	20.2	5.93	39.2	7.07	2.57	2.45	19.3	4.23	1.80	1.45	1.31	.560
8 × 4	1	37.4	11.0	69.6	14.1	2.52	3.05	11.6	3.94	1.03	1.05	.846	.247
	¾	28.7	8.44	54.9	10.9	2.55	2.95	9.36	3.07	1.05	.953	.852	.258
	9⁄16	21.9	6.43	42.8	8.35	2.58	2.88	7.43	2.38	1.07	.882	.861	.265
	½	19.6	5.75	38.5	7.49	2.59	2.86	6.74	2.15	1.08	.859	.865	.267
7 × 4	¾	26.2	7.69	37.8	8.42	2.22	2.51	9.05	3.03	1.09	1.01	.860	.324
	⅝	22.1	6.48	32.4	7.14	2.24	2.46	7.84	2.58	1.10	.963	.865	.329
	½	17.9	5.25	26.7	5.81	2.25	2.42	6.53	2.12	1.11	.917	.872	.335
	⅜	13.6	3.98	20.6	4.44	2.27	2.37	5.10	1.63	1.13	.870	.880	.340
6 × 4	⅞	27.2	7.98	27.7	7.15	1.86	2.12	9.75	3.39	1.11	1.12	.857	.421
	¾	23.6	6.94	24.5	6.25	1.88	2.08	8.68	2.97	1.12	1.08	.860	.428
	⅝	20.0	5.86	21.1	5.31	1.90	2.03	7.52	2.54	1.13	1.03	.864	.435
	9⁄16	18.1	5.31	19.3	4.83	1.90	2.01	6.91	2.31	1.14	1.01	.866	.438
	½	16.2	4.75	17.4	4.33	1.91	1.99	6.27	2.08	1.15	.987	.870	.440
	7⁄16	14.3	4.18	15.5	3.83	1.92	1.96	5.60	1.85	1.16	.964	.873	.443
	⅜	12.3	3.61	13.5	3.32	1.93	1.94	4.90	1.60	1.17	.941	.877	.446
	5⁄16	10.3	3.03	11.4	2.79	1.94	1.92	4.18	1.35	1.17	.918	.882	.448
6 × 3½	½	15.3	4.50	16.6	4.24	1.92	2.08	4.25	1.59	.972	.833	.759	.344
	⅜	11.7	3.42	12.9	3.24	1.94	2.04	3.34	1.23	.988	.787	.676	.350
	5⁄16	9.8	2.87	10.9	2.73	1.95	2.01	2.85	1.04	.996	.763	.772	.352
5 × 3½	¾	19.8	5.81	13.9	4.28	1.55	1.75	5.55	2.22	.977	.996	.748	.464
	⅝	16.8	4.92	12.0	3.65	1.56	1.70	4.83	1.90	.991	.951	.751	.472
	½	13.6	4.00	9.99	2.99	1.58	1.66	4.05	1.56	1.01	.906	.755	.479
	7⁄16	12.0	3.53	8.90	2.64	1.59	1.63	3.63	1.39	1.01	.883	.758	.482
	⅜	10.4	3.05	7.78	2.29	1.60	1.61	3.18	1.21	1.02	.861	.762	.486
	5⁄16	8.7	2.56	6.60	1.94	1.61	1.59	2.72	1.02	1.03	.838	.766	.489
	¼	7.0	2.06	5.39	1.57	1.62	1.56	2.23	.830	1.04	.814	.770	.492
5 × 3	⅝	15.7	4.61	11.4	3.55	1.57	1.80	3.06	1.39	.815	.796	.644	.349

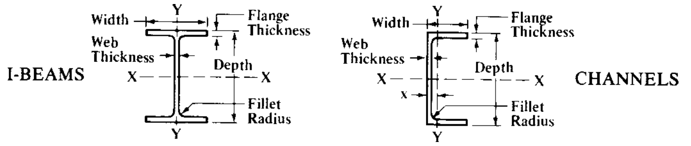
Steel Angles with Unequal Legs

5 × 3	1/2	12.8	3.75	9.45	2.91	1.59	1.75	2.58	1.15	.829	.750	.648	.357
	7/16	11.3	3.31	8.43	2.58	1.60	1.73	2.32	1.02	.837	.727	.651	.361
	3/8	9.8	2.86	7.37	2.24	1.61	1.70	2.04	.888	.845	.704	.654	.364
	5/16	8.2	2.40	6.26	1.89	1.61	1.68	1.75	.753	.853	.681	.658	.368
	1/4	6.6	1.94	5.11	1.53	1.62	1.66	1.44	.614	.861	.657	.663	.371
4 × 3½	5/8	14.7	4.30	6.37	2.35	1.22	1.29	4.52	1.84	1.03	1.04	.719	.745
	1/2	11.9	3.50	5.32	1.94	1.23	1.25	3.79	1.52	1.04	1.00	.722	.750
	7/16	10.6	3.09	4.76	1.72	1.24	1.23	3.40	1.35	1.05	.978	.724	.753
	3/8	9.1	2.67	4.18	1.49	1.25	1.21	2.95	1.17	1.06	.955	.727	.755
	5/16	7.7	2.25	3.56	1.26	1.26	1.18	2.55	.994	1.07	.932	.730	.757
	1/4	6.2	1.81	2.91	1.03	1.27	1.16	2.09	.808	1.07	.909	.734	.759
4 × 3	5/8	13.6	3.98	6.03	2.30	1.23	1.37	2.87	1.35	.849	.871	.637	.534
	1/2	11.1	3.25	5.05	1.89	1.25	1.33	2.42	1.12	.864	.827	.639	.543
	7/16	9.8	2.87	4.52	1.68	1.25	1.30	2.18	.992	.871	.804	.641	.547
	3/8	8.5	2.48	3.96	1.46	1.26	1.28	1.92	.866	.879	.782	.644	.551
	5/16	7.2	2.09	3.38	1.23	1.27	1.26	1.65	.734	.887	.759	.647	.554
	1/4	5.8	1.69	2.77	1.00	1.28	1.24	1.36	.599	.896	.736	.651	.558
3½ × 3	1/2	10.2	3.00	3.45	1.45	1.07	1.13	2.33	1.10	.881	.875	.621	.714
	7/16	9.1	2.65	3.10	1.29	1.08	1.10	2.09	.975	.889	.853	.622	.718
	3/8	7.9	2.30	2.72	1.13	1.09	1.08	1.85	.851	.897	.830	.625	.721
	5/16	6.6	1.93	2.33	.954	1.10	1.06	1.58	.722	.905	.808	.627	.724
	1/4	5.4	1.56	1.91	.776	1.11	1.04	1.30	.589	.914	.785	.631	.727
3½ × 2½	1/2	9.4	2.75	3.24	1.41	1.09	1.20	1.36	.760	.704	.705	.534	.486
	7/16	8.3	2.43	2.91	1.26	1.09	1.18	1.23	.677	.711	.682	.535	.491
	3/8	7.2	2.11	2.56	1.09	1.10	1.16	1.09	.592	.719	.660	.537	.496
	5/16	6.1	1.78	2.19	.927	1.11	1.14	.939	.504	.727	.637	.540	.501
	1/4	4.9	1.44	1.80	.755	1.12	1.11	.777	.412	.735	.614	.544	.506
3 × 2½	1/2	8.5	2.50	2.08	1.04	.913	1.00	1.30	.744	.722	.750	.520	.667
	7/16	7.6	2.21	1.88	.928	.920	.978	1.18	.664	.729	.728	.521	.672
	3/8	6.6	1.92	1.66	.810	.928	.956	1.04	.581	.736	.706	.522	.676
	5/16	5.6	1.62	1.42	.688	.937	.933	.898	.494	.744	.683	.525	.680
	1/4	4.5	1.31	1.17	.561	.945	.911	.743	.404	.753	.661	.528	.684
	3/16	3.39	.996	.907	.430	.954	.888	.577	.310	.761	.638	.533	.688
3 × 2	1/2	7.7	2.25	1.92	1.00	.924	1.08	.672	.474	.546	.583	.428	.414
	7/16	6.8	2.00	1.73	.894	.932	1.06	.609	.424	.553	.561	.429	.421
	3/8	5.9	1.73	1.53	.781	.940	1.04	.543	.371	.559	.539	.430	.428
	5/16	5.0	1.46	1.32	.664	.948	1.02	.740	.317	.567	.516	.432	.435
	1/4	4.1	1.19	1.09	.542	.957	.993	.392	.260	.574	.493	.435	.440
	3/16	3.07	.902	.842	.415	.966	.970	.307	.200	.583	.470	.439	.446
2½ × 2	3/8	5.3	1.55	.912	.547	.768	.831	.514	.363	.577	.581	.420	.614
	5/16	4.5	1.31	.788	.466	.776	.809	.446	.310	.584	.559	.422	.620
	1/4	3.62	1.06	.654	.381	.784	.787	.372	.254	.592	.537	.424	.626
	3/16	2.75	.809	.509	.293	.793	.764	.291	.196	.600	.514	.427	.631

Symbols: I = moment of inertia; S = section modulus; r = radius of gyration; x = distance from center of gravity of section to outer face of structural shape.

Data taken from the "Manual of Steel Construction," 8th Edition, 1980, with permission of the American Institute of Steel Construction.

Aluminum Association Standard Structural Shapes



Depth	Width	Weight per Foot	Area	Flange Thickness	Web Thickness	Fillet Radius	Axis X-X I	Axis X-X S	Axis X-X r	Axis Y-Y I	Axis Y-Y S	Axis Y-Y r	Axis Y-Y x
in.	in.	lb.	in.^{2}	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.	in.
I-BEAMS													
3.00	2.50	1.637	1.392	0.20	0.13	0.25	2.24	1.49	1.27	0.52	0.42	0.61	…
3.00	2.50	2.030	1.726	0.26	0.15	0.25	2.71	1.81	1.25	0.68	0.54	0.63	…
4.00	3.00	2.311	1.965	0.23	0.15	0.25	5.62	2.81	1.69	1.04	0.69	0.73	…
4.00	3.00	2.793	2.375	0.29	0.17	0.25	6.71	3.36	1.68	1.31	0.87	0.74	…
5.00	3.50	3.700	3.146	0.32	0.19	0.30	13.94	5.58	2.11	2.29	1.31	0.85	…
6.00	4.00	4.030	3.427	0.29	0.19	0.30	21.99	7.33	2.53	3.10	1.55	0.95	…
6.00	4.00	4.692	3.990	0.35	0.21	0.30	25.50	8.50	2.53	3.74	1.87	0.97	…
7.00	4.50	5.800	4.932	0.38	0.23	0.30	42.89	12.25	2.95	5.78	2.57	1.08	…
8.00	5.00	6.181	5.256	0.35	0.23	0.30	59.69	14.92	3.37	7.30	2.92	1.18	…
8.00	5.00	7.023	5.972	0.41	0.25	0.30	67.78	16.94	3.37	8.55	3.42	1.20	…
9.00	5.50	8.361	7.110	0.44	0.27	0.30	102.02	22.67	3.79	12.22	4.44	1.31	…
10.00	6.00	8.646	7.352	0.41	0.25	0.40	132.09	26.42	4.24	14.78	4.93	1.42	…
10.00	6.00	10.286	8.747	0.50	0.29	0.40	155.79	31.16	4.22	18.03	6.01	1.44	…
12.00	7.00	11.672	9.925	0.47	0.29	0.40	255.57	42.60	5.07	26.90	7.69	1.65	…
12.00	7.00	14.292	12.153	0.62	0.31	0.40	317.33	52.89	5.11	35.48	10.14	1.71	…
CHANNELS													
2.00	1.00	0.577	0.491	0.13	0.13	0.10	0.288	0.288	0.766	0.045	0.064	0.303	0.298
2.00	1.25	1.071	0.911	0.26	0.17	0.15	0.546	0.546	0.774	0.139	0.178	0.391	0.471
3.00	1.50	1.135	0.965	0.20	0.13	0.25	1.41	0.94	1.21	0.22	0.22	0.47	0.49
3.00	1.75	1.597	1.358	0.26	0.17	0.25	1.97	1.31	1.20	0.42	0.37	0.55	0.62
4.00	2.00	1.738	1.478	0.23	0.15	0.25	3.91	1.95	1.63	0.60	0.45	0.64	0.65
4.00	2.25	2.331	1.982	0.29	0.19	0.25	5.21	2.60	1.62	1.02	0.69	0.72	0.78
5.00	2.25	2.212	1.881	0.26	0.15	0.30	7.88	3.15	2.05	0.98	0.64	0.72	0.73
5.00	2.75	3.089	2.627	0.32	0.19	0.30	11.14	4.45	2.06	2.05	1.14	0.88	0.95
6.00	2.50	2.834	2.410	0.29	0.17	0.30	14.35	4.78	2.44	1.53	0.90	0.80	0.79
6.00	3.25	4.030	3.427	0.35	0.21	0.30	21.04	7.01	2.48	3.76	1.76	1.05	1.12
7.00	2.75	3.205	2.725	0.29	0.17	0.30	22.09	6.31	2.85	2.10	1.10	0.88	0.84
7.00	3.50	4.715	4.009	0.38	0.21	0.30	33.79	9.65	2.90	5.13	2.23	1.13	1.20
8.00	3.00	4.147	3.526	0.35	0.19	0.30	37.40	9.35	3.26	3.25	1.57	0.96	0.93
8.00	3.75	5.789	4.923	0.471	0.25	0.35	52.69	13.17	3.27	7.13	2.82	1.20	1.22
9.00	3.25	4.983	4.237	0.35	0.23	0.35	54.41	12.09	3.58	4.40	1.89	1.02	0.93
9.00	4.00	6.970	5.927	0.44	0.29	0.35	78.31	17.40	3.63	9.61	3.49	1.27	1.25
10.00	3.50	6.136	5.218	0.41	0.25	0.35	83.22	16.64	3.99	6.33	2.56	1.10	1.02
10.00	4.25	8.360	7.109	0.50	0.31	0.40	116.15	23.23	4.04	13.02	4.47	1.35	1.34
12.00	4.00	8.274	7.036	0.47	0.29	0.40	159.76	26.63	4.77	11.03	3.86	1.25	1.14
12.00	5.00	11.822	10.053	0.62	0.35	0.45	239.69	39.95	4.88	25.74	7.60	1.60	1.61

Structural sections are available in 6061-T6 aluminum alloy. Data supplied by The Aluminum Association.

Wire and Sheet-metal Gages

The thicknesses of sheet metals and the diameters of wires conform to various gaging systems. These gage sizes are indicated by numbers, and the following tables give the decimal equivalents of the different gage numbers. Much confusion has resulted from the use of gage numbers, and in ordering materials it is preferable to give the exact dimensions in decimal fractions of an inch. While the dimensions thus specified should conform to the gage ordinarily used for a given class of material, any error in the specification due, for example, to the use of a table having "rounded off" or approximate equivalents, will be apparent to the manufacturer at the time the order is placed. Furthermore, the decimal method of indicating wire diameters and sheet metal thicknesses has the advantage of being self-explanatory, whereas arbitrary gage numbers are not. The decimal system of indicating gage sizes is now being used quite generally, and gage numbers are gradually being discarded. Unfortunately, there is considerable variation in the use of different gages. For example, a gage ordinarily used for copper, brass and other non-ferrous materials, may at times be used for steel, and vice versa. The gages specified in the following are the ones ordinarily employed for the materials mentioned, but there are some minor exceptions and variations in the different industries.

Wire Gages.—The wire gage system used by practically all of the steel producers in the United States is known by the name Steel Wire Gage or to distinguish it from the Standard Wire Gage (S.W.G.) used in Great Britain it is called the United States Steel Wire Gage. It is the same as the Washburn and Moen, American Steel and Wire Company, and Roebling Wire Gages. The name has the official sanction of the Bureau of Standards at Washington but is not legally effective. The only wire gage which has been recognized in Acts of Congress is the Birmingham Gage (also known as Stub's Iron Wire). The Birmingham Gage is, however, nearly obsolete in both the United States and Great Britain, where it originated. Copper and aluminum wires are specified in decimal fractions. They were formerly universally specified in the United States by the American or Brown & Sharpe Wire Gage. Music spring steel wire, one of the highest quality wires of several types used for mechanical springs, is specified by the piano or music wire gage.

In Great Britain one wire gage has been legalized. This is called the Standard Wire Gage (S.W.G.), formerly called Imperial Wire Gage.

Gages for Rods.—Steel wire rod sizes are designated by fractional or decimal parts of an inch and by the gage numbers of the United States Steel Wire Gage. Copper and aluminum rods are specified by decimal fractions and fractions. Drill rod may be specified in decimal fractions but in the carbon and alloy tool steel grades may also be specified in the Stub's Steel Wire Gage and in the high-speed steel drill rod grade may be specified by the Morse Twist Drill Gage (Manufacturers' Standard Gage for Twist Drills). For gage numbers with corresponding decimal equivalents see the tables of American Standard Straight Shank Twist Drills.

Gages for Wall Thicknesses of Tubing.—At one time the Birmingham or Stub's Iron Wire Gage was used to specify the wall thickness of the following classes of tubing: seamless brass, seamless copper, seamless steel, and aluminum. The Brown & Sharpe Wire Gage was used for brazed brass and brazed copper tubing. Wall thicknesses are now specified by decimal parts of an inch but the wall thickness of steel pressure tubes and steel mechanical tubing may be specified by the Birmingham or Stub's Iron Wire Gage. In Great Britain the Standard Wire Gage (S.W.G.) is used to specify the wall thickness of some kinds of steel tubes.

Table 1. Wire Gages in Approximate Decimals of an Inch

No. of Wire Gage	American Wire or Brown & Sharpe Gage	Steel Wire Gage (U.S.)[a]	British Standard Wire Gage (Imperial Wire Gage)	Music or Piano Wire Gage	Birmingham or Stub's Iron Wire Gage	Stub's Steel Wire Gage	No. of Wire Gage	Stub's Steel Wire Gage
7/0	…	0.4900	0.5000	…	…	…	51	0.066
6/0	0.5800	0.4615	0.4640	0.004	…	…	52	0.063
5/0	0.5165	0.4305	0.4320	0.005	0.5000	…	53	0.058
4/0	0.4600	0.3938	0.4000	0.006	0.4540	…	54	0.055
3/0	0.4096	0.3625	0.3720	0.007	0.4250	…	55	0.050
2/0	0.3648	0.3310	0.3480	0.008	0.3800	…	56	0.045
1/0	0.3249	0.3065	0.3240	0.009	0.3400	…	57	0.042
1	0.2893	0.2830	0.3000	0.010	0.3000	0.227	58	0.041
2	0.2576	0.2625	0.2760	0.011	0.2840	0.219	59	0.040
3	0.2294	0.2437	0.2520	0.012	0.2590	0.212	60	0.039
4	0.2043	0.2253	0.2320	0.013	0.2380	0.207	61	0.038
5	0.1819	0.2070	0.2120	0.014	0.2200	0.204	62	0.037
6	0.1620	0.1920	0.1920	0.016	0.2030	0.201	63	0.036
7	0.1443	0.1770	0.1760	0.018	0.1800	0.199	64	0.035
8	0.1285	0.1620	0.1600	0.020	0.1650	0.197	65	0.033
9	0.1144	0.1483	0.1440	0.022	0.1480	0.194	66	0.032
10	0.1019	0.1350	0.1280	0.024	0.1340	0.191	67	0.031
11	0.0907	0.1205	0.1160	0.026	0.1200	0.188	68	0.030
12	0.0808	0.1055	0.1040	0.029	0.1090	0.185	69	0.029
13	0.0720	0.0915	0.0920	0.031	0.0950	0.182	70	0.027
14	0.0641	0.0800	0.0800	0.033	0.0830	0.180	71	0.026
15	0.0571	0.0720	0.0720	0.035	0.0720	0.178	72	0.024
16	0.0508	0.0625	0.0640	0.037	0.0650	0.175	73	0.023
17	0.0453	0.0540	0.0560	0.039	0.0580	0.172	74	0.022
18	0.0403	0.0475	0.0480	0.041	0.0490	0.168	75	0.020
19	0.0359	0.0410	0.0400	0.043	0.0420	0.164	76	0.018
20	0.0320	0.0348	0.0360	0.045	0.0350	0.161	77	0.016
21	0.0285	0.0318	0.0320	0.047	0.0320	0.157	78	0.015
22	0.0253	0.0286	0.0280	0.049	0.0280	0.155	79	0.014
23	0.0226	0.0258	0.0240	0.051	0.0250	0.153	80	0.013
24	0.0201	0.0230	0.0220	0.055	0.0220	0.151	…	…
25	0.0179	0.0204	0.0200	0.059	0.0200	0.148	…	…
26	0.0159	0.0181	0.0180	0.063	0.0180	0.146	…	…
27	0.0142	0.0173	0.0164	0.067	0.0160	0.143	…	…
28	0.0126	0.0162	0.0149	0.071	0.0140	0.139	…	…
29	0.0113	0.0150	0.0136	0.075	0.0130	0.134	…	…
30	0.0100	0.0140	0.0124	0.080	0.0120	0.127	…	…
31	0.00893	0.0132	0.0116	0.085	0.0100	0.120	…	…
32	0.00795	0.0128	0.0108	0.090	0.0090	0.115	…	…
33	0.00708	0.0118	0.0100	0.095	0.0080	0.112	…	…
34	0.00630	0.0104	0.0092	0.100	0.0070	0.110	…	…
35	0.00561	0.0095	0.0084	0.106	0.0050	0.108	…	…
36	0.00500	0.0090	0.0076	0.112	0.0040	0.106	…	…
37	0.00445	0.0085	0.0068	0.118	…	0.103	…	…
38	0.00396	0.0080	0.0060	0.124	…	0.101	…	…
39	0.00353	0.0075	0.0052	0.130	…	0.099	…	…
40	0.00314	0.0070	0.0048	0.138	…	0.097	…	…
41	0.00280	0.0066	0.0044	0.146	…	0.095	…	…
42	0.00249	0.0062	0.0040	0.154	…	0.092	…	…
43	0.00222	0.0060	0.0036	0.162	…	0.088	…	…
44	0.00198	0.0058	0.0032	0.170	…	0.085	…	…
45	0.00176	0.0055	0.0028	0.180	…	0.081	…	…
46	0.00157	0.0052	0.0024	…	…	0.079	…	…
47	0.00140	0.0050	0.0020	…	…	0.077	…	…
48	0.00124	0.0048	0.0016	…	…	0.075	…	…
49	0.00111	0.0046	0.0012	…	…	0.072	…	…
50	0.00099	0.0044	0.0010	…	…	0.069	…	…

[a] Also known as Washburn and Moen, American Steel and Wire Co. and Roebling Wire Gages. A greater selection of sizes is available and is specified by what are known as split gage numbers. They can be recognized by ½ fractions which follow the gage number; i.e., 4½. The decimal equivalents of split gage numbers are in the Steel Products Manual entitled: *Wire and Rods, Carbon Steel* published by the American Iron and Steel Institute, Washington, DC.

Sheet-Metal Gages.—Thicknesses of steel sheets are based upon a weight of 41.82 pounds per square foot per inch of thickness, which is known as the Manufacturers' Standard Gage for Sheet Steel. This gage differs from the older United States Standard Gage for iron and steel sheets and plates, established by Congress in 1893, based upon a weight of 40 pounds per square foot per inch of thickness which is the weight of wrought-iron plate.

Thicknesses of aluminum, copper, and copper-base alloys were formerly designated by the American or Brown & Sharpe Wire Gage but now are specified in decimals or fractions of an inch. American National Standard B32.1-1952 (R1988) entitled Preferred Thicknesses for Uncoated Thin Flat Metals (see accompanying Table 2) gives thicknesses that are based on the 20- and 40-series of preferred numbers in American National Standard Preferred Numbers — ANSI Z17.1 (see Handbook page 19) and are applicable to uncoated, thin, flat metals and alloys. Each number of the 20-series is approximately 12 percent greater than the next smaller one and each number of the 40-series is approximately 6 percent greater than the next smaller one.

Table 2. Preferred Thicknesses for Uncoated Metals and Alloys—Under 0.250 Inch in Thickness *ANSI B32.1-1952 (R1994)*

Preferred Thickness, Inches							
Based on 20-Series	Based on 40-Series	Based on 20-Series	Based on 40-Series	Based on 20-Series	Based on 40-Series	Based on 20-Series	Based on 40-Series
…	0.236	0.100	0.100	…	0.042	0.018	0.018
0.224	0.224	…	0.095	0.040	0.040	…	0.017
…	0.212	0.090	0.090	…	0.038	0.016	0.016
0.200	0.200	…	0.085	0.036	0.036	…	0.015
…	0.190	0.080	0.080	…	0.034	0.014	0.014
0.180	0.180	…	0.075	0.032	0.032	…	0.013
…	0.170	0.071	0.071	…	0.030	0.012	0.012
0.160	0.160	…	0.067	0.028	0.028	0.011	0.011
…	0.150	0.063	0.063	…	0.026	0.010	0.010
0.140	0.140	…	0.060	0.025	0.025	0.009	0.009
…	0.132	0.056	0.056	…	0.024	0.008	0.008
0.125	0.125	…	0.053	0.022	0.022	0.007	0.007
…	0.118	0.050	0.050	…	0.021	0.006	0.006
0.112	0.112	…	0.048	0.020	0.020	0.005	0.005
…	0.106	0.045	0.045	…	0.019	0.004	0.004

The American National Standard ANSI B32.1-1952 (R1994) lists preferred thicknesses that are based on the 20- and 40-series of preferred numbers and states that those based on the 40-series should provide adequate coverage. However, where intermediate thicknesses are required, the Standard recommends that thicknesses be based on the 80-series of preferred numbers (see Handbook page 19).

Thicknesses for copper and copper-base alloy flat products below ¼ inch thick are specified by the 20-series of American National Standard Preferred Numbers given in ANSI B32.1. Although the table in ANSI B32.1 gives only the 20- and 40-series of numbers, it states that when intermediate thicknesses are required they should be selected from thicknesses based on the 80-series of numbers (see Handbook page 19).

Zinc sheets are usually ordered by specifying decimal thickness although a zinc gage exists and is shown in Table 3.

Table 3. Sheet-Metal Gages in Approximate Decimals of an Inch

Gage No.	Steel Gage	B.G.[a]	Galvanized Sheet	Zinc Gage	Gage No.	Steel Gage	B.G.[a]	Galvanized Sheet	Zinc Gage
15⁄0	…	1.000	…	…	20	0.0359	0.0392	0.0396	0.070
14⁄0	…	0.9583	…	…	21	0.0329	0.0349	0.0366	0.080
13⁄0	…	0.9167	…	…	22	0.0299	0.03125	0.0336	0.090
12⁄0	…	0.8750	…	…	23	0.0269	0.02782	0.0306	0.100
11⁄0	…	0.8333	…	…	24	0.0239	0.02476	0.0276	0.125
10⁄0	…	0.7917	…	…	25	0.0209	0.02204	0.0247	…
9⁄0	…	0.7500	…	…	26	0.0179	0.01961	0.0217	…
8⁄0	…	0.7083	…	…	27	0.0164	0.01745	0.0202	…
7⁄0	…	0.6666	…	…	28	0.0149	0.01562	0.0187	…
6⁄0	…	0.6250	…	…	29	0.0135	0.01390	0.0172	…
5⁄0	…	0.5883	…	…	30	0.0120	0.01230	0.0157	…
4⁄0	…	0.5416	…	…	31	0.0105	0.01100	0.0142	…
3⁄0	…	0.5000	…	…	32	0.0097	0.00980	0.0134	…
2⁄0	…	0.4452	…	…	33	0.0090	0.00870	…	…
1⁄0	…	0.3964	…	…	34	0.0082	0.00770	…	…
1	…	0.3532	…	…	35	0.0075	0.00690	…	…
2	…	0.3147	…	…	36	0.0067	0.00610	…	…
3	0.2391	0.2804	…	0.006	37	0.0064	0.00540	…	…
4	0.2242	0.2500	…	0.008	38	0.0060	0.00480	…	…
5	0.2092	0.2225	…	0.010	39	…	0.00430	…	…
6	0.1943	0.1981	…	0.012	40	…	0.00386	…	…
7	0.1793	0.1764	…	0.014	41	…	0.00343	…	…
8	0.1644	0.1570	0.1681	0.016	42	…	0.00306	…	…
9	0.1495	0.1398	0.1532	0.018	43	…	0.00272	…	…
10	0.1345	0.1250	0.1382	0.020	44	…	0.00242	…	…
11	0.1196	0.1113	0.1233	0.024	45	…	0.00215	…	…
12	0.1046	0.0991	0.1084	0.028	46	…	0.00192	…	…
13	.0897	0.0882	0.0934	0.032	47	…	0.00170	…	…
14	0.0747	0.0785	0.0785	0.036	48	…	0.00152	…	…
15	0.0673	0.0699	0.0710	0.040	49	…	0.00135	…	…
16	0.0598	0.0625	0.0635	0.045	50	…	0.00120	…	…
17	0.0538	0.0556	0.0575	0.050	51	…	0.00107	…	…
18	.0478	0.0495	0.0516	0.055	52	…	0.00095	…	…
19	0.0418	0.0440	0.0456	0.060	…	…	…	…	…

[a] B.G. is the Birmingham Gage for sheets and hoops.

The United States Standard Gage (not shown above) for iron and steel sheets and plates was established by Congress in 1893 and was primarily a *weight* gage rather than a thickness gage. The equivalent thicknesses were derived from the weight of wrought iron. The weight per cubic foot was taken at 480 pounds, thus making the weight of a plate 12 inches square and 1 inch thick, 40 pounds. In converting weight to equivalent thickness, gage tables formerly published contained thicknesses equivalent to the basic weights just mentioned. For example, a No. 3 U.S. gage represents a wrought-iron plate having a weight of 10 pounds per square foot; hence, if the weight per square foot per inch thick is 40 pounds, the plate thickness for a No. 3 gage = 10 ÷ 40 = 0.25 inch, which was the original thickness equivalent for this gage number. Because this and the other thickness equivalents were derived from the weight of wrought iron, they are not correct for steel.

Most sheet-metal products in Great Britain are specified by the British Standard Wire Gage (Imperial Wire Gage). Black iron and steel sheet and hooping, and galvanized flat and corrugated steel sheet, however, are specified by the Birmingham Gage (B.G.), which was legalized in 1914. This Birmingham Gage should not be confused with the Birmingham or Stub's Iron Wire Gage mentioned previously.

Metric Sizes for Flat Metal Products.—American National Standard B32.3 M-1984 establishes a preferred series of metric thicknesses, widths, and lengths for flat metal products of rectangular cross section; the thickness and width values are also applicable to base metals that may be coated in later operations. Table 4 lists the preferred thicknesses. Whenever possible, the Preferred Thickness values should be used, with the Second or Third Preference chosen only if no suitable Preferred size is available. Since not all metals and grades are produced in each of the sizes given in Table 4, producers or distributors should be consulted to determine a particular product and size combination's availability.

Table 4. Preferred Metric Thicknesses for All Flat Metal Products
ANSI/ASME B32.3M-1984

Preferred Thickness	Second Preference	Third Preference	Preferred Thickness	Second Preference	Third Preference
0.050	…	…	…	…	3.6
0.060	…	…	…	3.8	…
0.080	…	…	4.0	…	…
0.10	…	…	…	4.2	…
0.12	…	…	…	4.5	…
…	0.14	…	…	4.8	…
0.16	…	…	5.0	…	…
…	0.18	…	…	5.5	…
0.20	…	…	6.0	…	…
…	0.22	…	…	…	6.5
0.25	…	…	…	7.0	…
…	0.28	…	…	…	7.5
0.30	…	…	8.0	…	…
…	0.35	…	…	9.0	…
0.40	…	…	10	…	…
…	0.45	…	…	11	…
0.50	…	…	12	…	…
…	0.55	…	…	14	…
0.60	…	…	16	…	…
…	0.65	…	…	18	…
…	0.70	…	20	…	…
…	…	0.75	…	22	…
0.80	…	…	25	…	…
…	…	0.85	…	28	…
…	0.90	…	30	…	…
…	…	0.95	…	32	…
1.0	…	…	35	…	…
…	…	1.05	…	38	…
…	1.1	…	40	…	…
1.2	…	…	…	45	…
…	…	1.3	50	…	…
…	1.4	…	…	55	…
…	…	1.5	60	…	…
1.6	…	…	…	70	…
…	…	1.7	80	…	…
…	1.8	…	…	90	…
…	…	1.9	100	…	…
2.0	…	…	…	110	…
…	…	2.1	120	…	…
…	2.2	…	…	130	…
…	…	2.4	140	…	…
2.5	…	…	…	150	…
…	…	2.6	160	…	…
…	2.8	…	180	…	…
3.0	…	…	200	…	…
…	3.2	…	250	…	…
…	…	3.4	300	…	…
3.5	…	…			

All dimensions are in millimeters.

PIPE AND PIPE FITTINGS

Wrought Steel Pipe.—ANSI/ASME B36.10M-1995 covers dimensions of welded and seamless wrought steel pipe, for high or low temperatures or pressures.

The word *pipe* as distinguished from *tube* is used to apply to tubular products of dimensions commonly used for pipelines and piping systems. Pipe dimensions of sizes 12 inches and smaller have outside diameters numerically larger than the corresponding nominal sizes whereas outside diameters of tubes are identical to nominal sizes.

Size: The size of all pipe is identified by the nominal pipe size. The manufacture of pipe in the nominal sizes of 1⁄8 inch to 12 inches, inclusive, is based on a standardized outside diameter (OD). This OD was originally selected so that pipe with a standard OD and having a wall thickness which was typical of the period would have an inside diameter (ID) approximately equal to the nominal size. Although there is now no such relation between the existing standard thicknesses, ODs and nominal sizes, these nominal sizes and standard ODs continue in use as "standard."

The manufacture of pipe in nominal sizes of 14-inch OD and larger proceeds on the basis of an OD corresponding to the nominal size.

Weight: The nominal weights of steel pipe are calculated values and are tabulated in Table 1. They are based on the following formula:

$$W_{pe} = 10.68(D-t)t$$

where W_{pe} = nominal plain end weight to the nearest 0.01 lb/ft.
D = outside diameter to the nearest 0.001 inch
t = specified wall thickness rounded to the nearest 0.001 inch

Wall thickness: The nominal wall thicknesses are given in Table 1 which also indicates the wall thicknesses in API Standard 5L.

The wall thickness designations "Standard," "Extra-Strong," and "Double Extra-Strong" have been commercially used designations for many years. The Schedule Numbers were subsequently added as a convenient designation for use in ordering pipe. "Standard" and Schedule 40 are identical for nominal pipe sizes up to 10 inches, inclusive. All larger sizes of "Standard" have 3⁄8-inch wall thickness. "Extra-Strong" and Schedule 80 are identical for nominal pipe sizes up to 8 inch, inclusive. All larger sizes of "Extra-Strong" have 1⁄2-inch-wall thickness.

Wall Thickness Selection: When the selection of wall thickness depends primarily on capacity to resist internal pressure under given conditions, the designer shall compute the exact value of wall thickness suitable for conditions for which the pipe is required as prescribed in the "ASME Boiler and Pressure Vessel Code," "ANSI B31 Code for Pressure Piping," or other similar codes, whichever governs the construction. A thickness can then be selected from Table 1 to suit the value computed to fulfill the conditions for which the pipe is desired.

Metric Weights and Mass: Standard SI metric dimensions in millimeters for outside diameters and wall thicknesses may be found by multiplying the inch dimensions by 25.4. Outside diameters converted from those shown in Table 1 should be rounded to the nearest 0.1 mm and wall thicknesses to the nearest 0.01 mm.

The following formula may be used to calculate the SI metric plain end mass in kg/m using the converted metric diameters and thicknesses:

$$W_{pe} = 0.02466(D-t)t$$

where W_{pe} = nominal plain end mass rounded to the nearest 0.01 kg/m.
D = outside diameter to the nearest 0.1 mm for sizes shown in Table 1.
t = specified wall thickness rounded to the nearest 0.01 mm.

Table 1. American National Standard Weights and Dimensions of Welded and Seamless Wrought Steel Pipe *ANSI/ASME B36.10M-1995*

Nom. Size and (O.D.), in.	Wall Thick., in.	Plain End Wgt., lb/ft	Identification		
			Sch. No.	Other	
⅛ (0.405)	0.057	0.21	30	…	…
	0.068	0.24	40	5L	STD
	0.095	0.31	80	5L	XS
¼ (0.540)	0.073	0.36	30	…	…
	0.088	0.42	40	5L	STD
	0.119	0.54	80	5L	XS
⅜ (0.675)	0.073	0.47	30	…	…
	0.091	0.57	40	5L	STD
	0.126	0.74	80	5L	XS
½ (0.840)	0.095	0.76	30	…	…
	0.109	0.85	40	5L	STD
	0.147	1.09	80	5L	XS
	0.188	1.31	160	…	…
	0.294	1.71	…	5L	XXS
¾ (1.050)	0.095	0.97	30	…	…
	0.113	1.13	40	5L	STD
	0.154	1.47	80	5L	XS
	0.219	1.94	160	…	…
	0.308	2.44	…	5L	XXS
1 (1.315)	0.114	1.46	30	…	…
	0.133	1.68	40	5L	STD
	0.179	2.17	80	5L	XS
	0.250	2.84	160	…	…
	0.358	3.66	…	5L	XXS
1¼ (1.660)	0.117	1.93	30	…	…
	0.140	2.27	40	5L	STD
	0.191	3.00	80	5L	XS
	0.250	3.76	160	…	…
	0.382	5.21	…	5L	XXS
1½ (1.900)	0.125	2.37	30	…	…
	0.145	2.72	40	5L	STD
	0.200	3.63	80	5L	XS
	0.281	4.86	160	…	…
	0.400	6.41	…	5L	XXS
2 (2.375)	0.083	2.03	…	5L	…
	0.109	2.64	…	5L	…
	0.125	3.00	…	5L	…
	0.141	3.36	…	5L	…
	0.154	3.65	40	5L	STD
	0.172	4.05	…	5L	…
	0.188	4.39	…	5L	…
	0.218	5.02	80	5L	XS
	0.250	5.67	…	5L	…
	0.281	6.28	…	…	…
	0.344	7.46	160	…	…
	0.436	9.03	…	…	XXS
2½ (2.875)	0.083	2.47	…	5L	…
	0.109	3.22	…	5L	…
	0.125	3.67	…	5L	…
	0.141	4.12	…	5L	…
	0.156	4.53	…	5L	…
	0.172	4.97	…	5L	…
	0.188	5.40	…	5L	…
	0.203	5.79	40	5L	STD
	0.216	6.13	…	5L	…
	0.250	7.01	…	5L	…
	0.276	7.66	80	5L	XS
	0.375	10.01	160	…	…
	0.552	13.69	…	5L	XXS
3 (3.500)	0.083	3.03	…	5L	…
	0.109	3.95	…	5L	…
	0.125	4.51	…	5L	…
3 (3.500)	0.141	5.06	…	5L	…
	0.156	5.57	…	5L	…
	0.172	6.11	…	5L	…
	0.188	6.65	…	5L	…
	0.216	7.58	40	5L	STD
	0.250	8.68	…	5L	…
	0.281	9.66	…	5L	…
	0.300	10.25	80	5L	XS
	0.438	14.32	160	…	…
	0.600	18.58	…	5L	XXS
3½ (4.000)	0.083	3.47	…	5L	…
	0.109	4.53	…	5L	…
	0.125	5.17	…	5L	…
	0.141	5.81	…	5L	…
	0.156	6.40	…	5L	…
	0.172	7.03	…	5L	…
	0.188	7.65	…	5L	…
	0.226	9.11	40	5L	STD
	0.250	10.01	…	5L	…
	0.281	11.16	…	5L	…
	0.318	12.50	80	5L	XS
4 (4.500)	0.083	3.92	…	5L	…
	0.109	5.11	…	5L	…
	0.125	5.84	…	5L	…
	0.141	6.56	…	5L	…
	0.156	7.24	…	5L	…
	0.172	7.95	…	5L	…
	0.188	8.66	…	5L	…
	0.203	9.32	…	5L	…
	0.219	10.01	…	5L	…
	0.237	10.79	40	5L	STD
	0.250	11.35	…	5L	…
	0.281	12.66	…	5L	…
	0.312	13.96	…	5L	…
	0.337	14.98	80	5L	XS
	0.438	19.00	120	5L	…
	0.531	22.51	160	5L	…
	0.674	27.54	…	5L	XXS
5 (5.563)	0.083	4.86	…	5L	…
	0.125	7.26	…	5L	…
	0.156	9.01	…	5L	…
	0.188	10.79	…	5L	…
	0.219	12.50	…	5L	…
	0.258	14.62	40	5L	STD
	0.281	15.85	…	5L	…
	0.312	17.50	…	5L	…
	0.344	19.17	…	5L	…
	0.375	20.78	80	5L	XS
	0.500	27.04	120	5L	…
	0.625	32.96	160	5L	…
	0.750	38.55	…	5L	XXS
6 (6.625)	0.083	5.80	…	5L	…
	0.109	7.59	…	5L	…
	0.125	8.68	…	5L	…
	0.141	9.76	…	5L	…
	0.156	10.78	…	5L	…
	0.172	11.85	…	5L	…

Table 1. *(Continued)* **American National Standard Weights and Dimensions of Welded and Seamless Wrought Steel Pipe** *ANSI/ASME B36.10M-1995*

Nom. Size and (O.D.), in.	Wall Thick., in.	Plain End Wgt., lb/ft	Identification		
			Sch. No.	Other	
6 (6.625)	0.188	12.92	…	5L	…
	0.203	13.92	…	5L	…
	0.219	14.98	…	5L	…
	0.250	17.02	…	5L	…
	0.280	18.97	40	5L	STD
	0.312	21.04	…	5L	…
	0.344	23.08	…	5L	…
	0.375	25.03	…	5L	…
	0.432	28.57	80	5L	XS
	0.500	32.71	…	5L	…
	0.562	36.39	120	5L	…
	0.625	40.05	…	5L	…
	0.719	45.35	160	5L	…
	0.750	47.06	…	5L	…
	0.864	53.16	…	5L	XXS
	0.875	53.73	…	5L	…
8 (8.625)	0.125	11.35	…	5L	…
	0.156	14.11	…	5L	…
	0.188	16.94	…	5L	…
	0.203	18.26	…	5L	…
	0.219	19.66	…	5L	…
	0.250	22.36	20	5L	…
	0.277	24.70	30	5L	…
	0.312	27.70	…	5L	…
	0.322	28.55	40	5L	STD
	0.344	30.42	…	5L	…
	0.375	33.04	…	5L	…
	0.406	35.64	60	…	…
	0.438	38.30	…	5L	…
	0.500	43.39	80	5L	XS
	0.562	48.40	…	5L	…
	0.594	50.95	100	…	…
	0.625	53.40	…	5L	…
	0.719	60.71	120	5L	…
	0.750	63.08	…	5L	…
	0.812	67.76	140	5L	…
	0.875	72.42	…	5L	XXS
	0.906	74.69	160	…	…
	1.000	81.44	…	5L	…
10 (10.750)	0.156	17.65	…	5L	…
	0.188	21.21	…	5L	…
	0.203	22.87	…	5L	…
	0.219	24.63	…	5L	…
	0.250	28.04	20	5L	…
	0.279	31.20	…	5L	…
	0.307	34.24	30	5L	…
	0.344	38.23	…	5L	…
	0.365	40.48	40	5L	STD
	0.438	48.24	…	5L	…
	0.500	54.74	60	5L	XS
	0.562	61.15	…	5L	…
	0.594	64.43	80	…	…
	0.625	67.58	…	5L	…
	0.719	77.03	100	5L	…
	0.812	86.18	…	5L	…
	0.844	89.29	120	…	…
	0.875	92.28	…	5L	…
	0.938	98.30	…	5L	…
	1.000	104.13	140	5L	XXS
10 (10.750)	1.125	115.64	160	…	…
	1.250	126.83	…	5L	…
12 (12.750)	0.172	23.11	…	5L	…
	0.188	25.22	…	5L	…
	0.203	27.20	…	…	…
	0.219	29.31	…	5L	…
	0.250	33.38	20	5L	…
	0.281	37.42	…	5L	…
	0.312	41.45	…	5L	…
	0.330	43.77	30	5L	…
	0.344	45.58	…	5L	…
	0.375	49.56	…	5L	STD
	0.406	53.52	40	5L	…
	0.438	57.59	…	5L	…
	0.500	65.42	…	5L	XS
	0.562	73.15	60	5L	…
	0.625	80.93	…	5L	…
	0.688	88.63	80	5L	…
	0.750	96.12	…	5L	…
	0.812	103.53	…	5L	…
	0.844	107.32	100	…	…
	0.875	110.97	…	5L	…
	0.938	118.33	…	5L	…
	1.000	125.49	120	5L	XXS
	1.062	132.57	…	5L	…
	1.125	139.67	140	5L	…
	1.250	153.53	…	5L	…
	1.312	160.27	160	5L	…
14 (14.000)	0.188	27.73	…	5L	…
	0.203	29.91	…	5L	…
	0.210	30.93	…	5L	…
	0.219	32.23	…	5L	…
	0.250	36.71	10	5L	…
	0.281	41.17	…	5L	…
	0.312	45.61	20	5L	…
	0.344	50.17	…	5L	…
	0.375	54.57	30	5L	STD
	0.406	58.94	…	5L	…
	0.438	63.44	40	5L	…
	0.469	67.78	…	5L	…
	0.500	72.09	…	5L	XS
	0.562	80.66	…	5L	…
	0.594	85.05	60	…	…
	0.625	89.28	…	5L	…
	0.688	97.81	…	5L	…
	0.750	106.13	80	5L	…
	0.812	114.37	…	5L	…
	0.875	122.65	…	5L	…
	0.938	130.85	100	5L	…
	1.000	138.84	…	5L	…
	1.062	146.74	…	5L	…
	1.094	150.79	120	…	…
	1.125	154.69	…	5L	…
	1.250	170.21	140	5L	…
	1.406	189.11	160	…	…
	2.000	256.32	…	…	…
	2.125	269.50	…	…	…
	2.200	277.25	…	…	…
	2.500	307.05	…	…	…

Table 2. Properties of American National Standard Schedule 40 Welded and Seamless Wrought Steel Pipe

Diameter, Inches			Wall Thickness, Inches	Cross-Sectional Area of Metal	Weight per Foot, Pounds		Capacity per Foot of Length		Length of Pipe in Feet to Contain		Properties of Sections		
Nominal	Actual Inside	Actual Outside			Of Pipe	Of Water in Pipe	In Cubic Inches	In Gallons	One Cubic Foot	One Gallon	Moment of Inertia	Radius of Gyration	Section Modulus
⅛	0.269	0.405	0.068	0.072	0.24	0.025	0.682	0.003	2532.	338.7	0.00106	0.122	0.00525
¼	0.364	0.540	0.088	0.125	0.42	0.045	1.249	0.005	1384.	185.0	0.00331	0.163	0.01227
⅜	0.493	0.675	0.091	0.167	0.57	0.083	2.291	0.010	754.4	100.8	0.00729	0.209	0.02160
½	0.622	0.840	0.109	0.250	0.85	0.132	3.646	0.016	473.9	63.35	0.01709	0.261	0.4070
¾	0.824	1.050	0.113	0.333	1.13	0.231	6.399	0.028	270.0	36.10	0.03704	0.334	0.07055
1	1.049	1.315	0.133	0.494	1.68	0.374	10.37	0.045	166.6	22.27	0.08734	0.421	0.1328
1¼	1.380	1.660	0.140	0.669	2.27	0.648	17.95	0.078	96.28	12.87	0.1947	0.539	0.2346
1½	1.610	1.900	0.145	0.799	2.72	0.882	24.43	0.106	70.73	9.456	0.3099	0.623	0.3262
2	2.067	2.375	0.154	1.075	3.65	1.454	40.27	0.174	42.91	5.737	0.6658	0.787	0.5607
2½	2.469	2.875	0.203	1.704	5.79	2.074	57.45	0.249	30.08	4.021	1.530	0.947	1.064
3	3.068	3.500	0.216	2.228	7.58	3.202	88.71	0.384	19.48	2.604	3.017	1.163	1.724
3½	3.548	4.000	0.226	2.680	9.11	4.283	118.6	0.514	14.56	1.947	4.788	1.337	2.394
4	4.026	4.500	0.237	3.174	10.79	5.515	152.8	0.661	11.31	1.512	7.233	1.510	3.215
5	5.047	5.563	0.258	4.300	14.62	8.666	240.1	1.04	7.198	0.9622	15.16	1.878	5.451
6	6.065	6.625	0.280	5.581	18.97	12.52	346.7	1.50	4.984	0.6663	28.14	2.245	8.496
8	7.981	8.625	0.322	8.399	28.55	21.67	600.3	2.60	2.878	0.3848	72.49	2.938	16.81
10	10.020	10.750	0.365	11.91	40.48	34.16	946.3	4.10	1.826	0.2441	160.7	3.674	29.91
12	11.938	12.750	0.406	15.74	53.52	48.49	1343.	5.81	1.286	0.1720	300.2	4.364	47.09
16	15.000	16.000	0.500	24.35	82.77	76.55	2121.	9.18	0.8149	0.1089	732.0	5.484	91.50
18	16.876	18.000	0.562	30.79	104.7	96.90	2684.	11.62	0.6438	0.0861	1172.	6.168	130.2
20	18.812	20.000	0.594	36.21	123.1	120.4	3335.	14.44	0.5181	0.0693	1706.	6.864	170.6
24	22.624	24.000	0.688	50.39	171.3	174.1	4824.	20.88	0.3582	0.0479	3426.	8.246	285.5
32	30.624	32.000	0.688	67.68	230.1	319.1	8839.	38.26	0.1955	0.0261	8299.	11.07	518.7

Note: Torsional section modulus equals twice section modulus.

Table 3. Properties of American National Standard Schedule 80 Welded and Seamless Wrought Steel Pipe

Diameter, Inches			Wall Thickness, Inches	Cross-Sectional Area of Metal	Weight per Foot, Pounds		Capacity per Foot of Length		Length of Pipe in Feet to Contain		Properties of Sections		
Nominal	Actual Inside	Actual Outside			Of Pipe	Of Water in Pipe	In Cubic Inches	In Gallons	One Cubic Foot	One Gallon	Moment of Inertia	Radius of Gyration	Section Modulus
⅛	0.215	0.405	0.095	0.093	0.315	0.016	0.436	0.0019	3966.	530.2	0.00122	0.115	0.00600
¼	0.302	0.540	0.119	0.157	0.537	0.031	0.860	0.0037	2010.	268.7	0.00377	0.155	0.01395
⅜	0.423	0.675	0.126	0.217	0.739	0.061	1.686	0.0073	1025.	137.0	0.00862	0.199	0.02554
½	0.546	0.840	0.147	0.320	1.088	0.101	2.810	0.0122	615.0	82.22	0.02008	0.250	0.04780
¾	0.742	1.050	0.154	0.433	1.474	0.187	5.189	0.0225	333.0	44.52	0.04479	0.321	0.08531
1	0.957	1.315	0.179	0.639	2.172	0.312	8.632	0.0374	200.2	26.76	0.1056	0.407	0.1606
1¼	1.278	1.660	0.191	0.881	2.997	0.556	15.39	0.0667	112.3	15.01	0.2418	0.524	0.2913
1½	1.500	1.900	0.200	1.068	3.631	0.766	21.21	0.0918	81.49	10.89	0.3912	0.605	0.4118
2	1.939	2.375	0.218	1.477	5.022	1.279	35.43	0.1534	48.77	6.519	0.8680	0.766	0.7309
2½	2.323	2.875	0.276	2.254	7.661	1.836	50.86	0.2202	33.98	4.542	1.924	0.924	1.339
3	2.900	3.500	0.300	3.016	10.25	2.861	79.26	0.3431	21.80	2.914	3.895	1.136	2.225
3½	3.364	4.000	0.318	3.678	12.50	3.850	106.7	0.4617	16.20	2.166	6.280	1.307	3.140
4	3.826	4.500	0.337	4.407	14.98	4.980	138.0	0.5972	12.53	1.674	9.611	1.477	4.272
5	4.813	5.563	0.375	6.112	20.78	7.882	218.3	0.9451	7.915	1.058	20.67	1.839	7.432
6	5.761	6.625	0.432	8.405	28.57	11.29	312.8	1.354	5.524	0.738	40.49	2.195	12.22
8	7.625	8.625	0.500	12.76	43.39	19.78	548.0	2.372	3.153	0.422	105.7	2.878	24.52
10	9.562	10.750	0.594	18.95	64.42	31.11	861.7	3.730	2.005	0.268	245.2	3.597	45.62
12	11.374	12.750	0.688	26.07	88.63	44.02	1219.	5.278	1.417	0.189	475.7	4.271	74.62
14	12.500	14.000	0.750	31.22	106.1	53.16	1473.	6.375	1.173	0.157	687.4	4.692	98.19
16	14.312	16.000	0.844	40.19	136.6	69.69	1931.	8.357	0.895	0.120	1158.	5.366	144.7
18	16.124	18.000	0.938	50.28	170.9	88.46	2450.	10.61	0.705	0.094	1835.	6.041	203.9
20	17.938	20.000	1.031	61.44	208.9	109.5	3033.	13.13	0.570	0.076	2772.	6.716	277.2
22	19.750	22.000	1.125	73.78	250.8	132.7	3676.	15.91	0.470	0.063	4031.	7.391	366.4

Note: Torsional section modulus equals twice section modulus.

Volume of Flow at 1 Foot Per-Minute Velocity in Pipe and Tube

Nominal Dia., Inches	Schedule 40 Pipe			Schedule 80 Pipe			Type K Copper Tube			Type L Copper Tube		
	Cu. Ft. per Minute	Gallons per Minute	Pounds 60 F Water per Min.	Cu. Ft. per Minute	Gallons per Minute	Pounds 60 F Water per Min.	Cu. Ft. per Minute	Gallons per Minute	Pounds 60 F Water per Min.	Cu. Ft. per Minute	Gallons per Minute	Pounds 60 F Water per Min.
1/8	0.0004	0.003	0.025	0.0003	0.002	0.016	0.0002	0.0014	0.012	0.0002	0.002	0.014
1/4	0.0007	0.005	0.044	0.0005	0.004	0.031	0.0005	0.0039	0.033	0.0005	0.004	0.034
3/8	0.0013	0.010	0.081	0.0010	0.007	0.061	0.0009	0.0066	0.055	0.0010	0.008	0.063
1/2	0.0021	0.016	0.132	0.0016	0.012	0.102	0.0015	0.0113	0.094	0.0016	0.012	0.101
3/4	0.0037	0.028	0.232	0.0030	0.025	0.213	0.0030	0.0267	0.189	0.0034	0.025	0.210
1	0.0062	0.046	0.387	0.0050	0.037	0.312	0.0054	0.0404	0.338	0.0057	0.043	0.358
1 1/4	0.0104	0.078	0.649	0.0088	0.067	0.555	0.0085	0.0632	0.53	0.0087	0.065	0.545
1 1/2	0.0141	0.106	0.882	0.0123	0.092	0.765	0.0196	0.1465	1.22	0.0124	0.093	0.770
2	0.0233	0.174	1.454	0.0206	0.154	1.280	0.0209	0.1565	1.31	0.0215	0.161	1.34
2 1/2	0.0332	0.248	2.073	0.0294	0.220	1.830	0.0323	0.2418	2.02	0.0331	0.248	2.07
3	0.0514	0.383	3.201	0.0460	0.344	2.870	0.0461	0.3446	2.88	0.0473	0.354	2.96
3 1/2	0.0682	0.513	4.287	0.0617	0.458	3.720	0.0625	0.4675	3.91	0.0640	0.479	4.00
4	0.0884	0.660	5.516	0.0800	0.597	4.970	0.0811	0.6068	5.07	0.0841	0.622	5.20
5	0.1390	1.040	8.674	0.1260	0.947	7.940	0.1259	0.9415	7.87	0.1296	0.969	8.10
6	0.2010	1.500	12.52	0.1820	1.355	11.300	0.1797	1.3440	11.2	0.1862	1.393	11.6
8	0.3480	2.600	21.68	0.3180	2.380	19.800	0.3135	2.3446	19.6	0.3253	2.434	20.3
10	0.5476	4.10	34.18	0.5560	4.165	31.130	0.4867	3.4405	30.4	0.5050	3.777	21.6
12	0.7773	5.81	48.52	0.7060	5.280	44.040	0.6978	5.2194	43.6	0.7291	5.454	45.6
14	0.9396	7.03	58.65	0.8520	6.380	53.180	—	—	—	—	—	—
16	1.227	9.18	76.60	1.1170	8.360	69.730	—	—	—	—	—	—
18	1.553	11.62	96.95	1.4180	10.610	88.500	—	—	—	—	—	—
20	1.931	14.44	120.5	1.7550	13.130	109.510	—	—	—	—	—	—

To obtain volume of flow at any other velocity, multiply values in table by velocity in feet per minute.

Plastics Pipe.— Shortly after World War II, plastics pipe became an acceptable substitute, under certain service conditions, for other piping materials. Now, however, plastics pipe is specified on the basis of its own special capabilities and limitations. The largest volume of application has been for water piping systems.

Besides being light in weight, plastics pipe performs well in resisting deterioration from corrosive or caustic fluids. Even if the fluid borne is harmless, the chemical resistance of plastics pipe offers protection against a harmful exterior environment, such as when buried in a corrosive soil.

Generally, plastics pipe is limited by its temperature and pressure capacities. The higher the operating pressure of the pipe system, the less will be its temperature capability. The reverse is true, also. Since it is formed from organic resins, plastics pipe will burn. For various piping compositions, ignition temperatures vary from 700° to 800°F (370° to 430°C).

The following are accepted methods for joining plastics pipe:

Solvent Welding is usually accomplished by brushing a solvent cement on the end of the length of pipe and into the socket end of a fitting or the flange of the next pipe section. A chemical weld then joins and seals the pipe after connection.

Threading is a procedure not recommended for thin-walled plastics pipe or for specific grades of plastics. During connection of thicker-walled pipe, strap wrenches are used to avoid damaging and weakening the plastics.

Heat Fusion involves the use of heated air and plastics filler rods to weld plastics pipe assemblies. A properly welded joint can have a tensile strength equal to 90 percent that of the pipe material.

Elastomeric Sealing is used with bell-end piping. It is a recommended procedure for large diameter piping and for underground installations. The joints are set quickly and have good pressure capabilities.

Table 1. Dimensions and Weights of Thermoplastics Pipe

Nominal Pipe Size		Outside Diameter		Schedule 40				Schedule 80			
				Nom. Wall Thickness		Nominal Weight		Nom. Wall Thickness		Nominal Weight	
in.	cm	in.	cm	in.	cm	lb/100′	kg/m	in.	cm	lb/100′	kg/m
1/8	0.3	0.405	1.03	0.072	0.18	3.27	0.05	0.101	0.256	4.18	0.06
1/4	0.6	0.540	1.37	0.093	0.24	5.66	0.08	0.126	0.320	7.10	0.11
3/8	1.0	0.675	1.71	0.096	0.24	7.57	0.11	0.134	0.340	9.87	0.15
1/2	1.3	0.840	2.13	0.116	0.295	11.4	0.17	0.156	0.396	14.5	0.22
3/4	2.0	1.050	2.67	0.120	0.305	15.2	0.23	0.163	0.414	19.7	0.29
1	2.5	1.315	3.34	0.141	0.358	22.5	0.33	0.190	0.483	29.1	0.43
1 1/4	3.2	1.660	4.22	0.148	0.376	30.5	0.45	0.202	0.513	40.1	0.60
1 1/2	3.8	1.900	4.83	0.154	0.391	36.6	0.54	0.212	0.538	48.7	0.72
2	5.1	2.375	6.03	0.163	0.414	49.1	0.73	0.231	0.587	67.4	1.00
2 1/2	6.4	2.875	7.30	0.215	0.546	77.9	1.16	0.293	0.744	103	1.5
3	7.6	3.500	8.89	0.229	0.582	102	1.5	0.318	0.808	138	2.1
3 1/2	8.9	4.000	10.16	0.240	0.610	123	1.8	0.337	0.856	168	2.5
4	10.2	4.500	11.43	0.251	0.638	145	2.2	0.357	0.907	201	3.0
5	12.7	5.563	14.13	0.273	0.693	197	2.9	0.398	1.011	280	4.2
6	15.2	6.625	16.83	0.297	0.754	256	3.8	0.458	1.163	385	5.7
8	20.3	8.625	21.91	0.341	0.866	385	5.7	0.530	1.346	584	8.7
10	25.4	10.75	27.31	0.387	0.983	546	8.1	0.629	1.598	867	12.9
12	30.5	12.75	32.39	0.430	1.09	722	10.7	0.728	1.849	1192	17.7

The nominal weights of plastics pipe given in this table are based on an empirically chosen material density of 1.00 g/cm^3. The nominal unit weight for a specific plastics pipe formulation can be obtained by multiplying the weight values from the table by the density in g/cm^3 or by the specific gravity of the particular platics composition.

The following are ranges of density factors for various plastics pipe materials: PE, 0.93 to 0.96; PVC, 1.35 to 1.40; CPVC, 1.55; ABS, 1.04 to 1.08; SR, 1.05; PB, 0.91 to 0.92; and PP, 0.91. For meanings of abbreviations see Table 2.

Information supplied by the Plastics Pipe Institute.

Insert Fitting is particularly useful for PE and PB pipe. For joining pipe sections, insert fittings are pushed into the pipe and secured by stainless steel clamps.

Transition Fitting involves specially designed connectors to join plastic pipe with other materials, such as cast iron, steel, copper, clay, and concrete.

Plastic pipe can be specified by means of Schedules 40, 80, and 120, which conform dimensionally to metal pipe, or through a Standard Dimension Ratio (SDR). The SDR is a rounded value obtained by dividing the average outside diameter of the pipe by the wall thickness. Within an individual SDR series of pipe, pressure ratings are uniform, regardless of pipe diameter.

Table 1 provides the weights and dimensions for Schedule 40 and 80 thermoplastic pipe, Table 2 gives properties of plastics pipe, Table 3 gives maximum non-shock operating pressures for several varieties of Schedule 40 and 80 plastics pipe at 73°F, and Table 4 gives correction factors to pressure ratings for elevated temperatures.

Table 2. General Properties and Uses of Plastic Pipe

Plastic Pipe Material	Properties	Common Uses	Operating Temperature[a]		Joining Methods
			With Pressure	Without Pressure	
ABS (Acrylonitrilebutadiene styrene)	Rigid; excellent impact strength at low temperatures; maintains rigidity at higher temperatures.	Water, Drain, Waste, Vent, Sewage.	100°F (38°C)	180°F (82°C)	Solvent cement, Threading, Transition fitting.
PE (Polyethylene)	Flexible; excellent impact strength; good performance at low temperatures.	Water, Gas, Chemical, Irrigation.	100°F (38°C)	180°F (82°C)	Heat fusion, Insert and Transition fitting.
PVC (Polyvinylchloride)	Rigid; fire self-extinguishing; high impact and tensile strength.	Water, Gas, Sewage, Industrial process, Irrigation.	100°F (38°C)	180°F (82°C)	Solvent cement, Elastomeric seal, Mechanical coupling, Transition fitting.
CPVC (Chlorinated polyvinyl chloride)	Rigid; fire self-extinguishing; high impact and tensile strength.	Hot and cold water, Chemical.	180°F (82°C) at 100 psig (690kPa) for SDR-11		Solvent cement, Threading, Mechanical coupling, Transition fitting.
PB (Polybutylene)	Flexible; good performance at elevated temperatures.	Water, Gas, Irrigation.	180°F (82°C)	200°F (93°C)	Insert fitting, Heat fusion, Transition fitting.
PP (Polypropylene)	Rigid; very light; high chemical resistance, particularly to sulfur-bearing compounds.	Chemical waste and processing.	100°F (38°C)	180°F (82°C)	Mechanical coupling, Heat fusion, Threading.
SR (Styrene rubber plastic)	Rigid; moderate chemical resistance; fair impact strength.	Drainage, Septic fields.	150°F (66°C)	…	Solvent cement, Transition fitting, Elastomeric seal.

[a] The operating temperatures shows are general guide points. For specific operating temperature and pressure data for various grades of the types of plastic pipe given, please consult the pipe manufacturer or the Plastics Pipe Institute.

From information supplied by the Plastics Pipe Institute.

Table 3. Maximum Nonshock Operating Pressure (psi) for Thermoplastic Piping at 73°F

Nominal Pipe Size (inch)	Schedule 40 PVC & CPVC (Socket End)	Schedule 40 ABS	Schedule 80 PVC & CPVC Socket End	PVC & CPVC Threaded End	Polypropylene Thermo-seal Joint	Polypropylene Threaded End[a]	PVDF Thermoseal Joint	PVDF Threaded End	ABS
½	600	476	850	420	410	20	580	290	678
¾	480	385	690	340	330	20	470	230	550
1	450	360	630	320	310	20	430	210	504
1¼	370	294	520	260	260	20	…	…	416
1½	330	264	470	240	230	20	326	160	376
2	280	222	400	200	200	…	270	140	323
2½	300	243	420	210	…	20	…	…	340
3	260	211	370	190	160	20	250	NR	297
4	220	177	320	160	140	NR	220	NR	259
6	180	141	280	NR	…	…	190	NR	222
8	160	…	250[b]	NR	…	…	…	…	…
10	140	…	230	NR	…	…	…	…	…
12	130	…	230	NR	…	…	…	…	…

[a] Recommended for intermittent drainage pressure not exceeding 20 psi.
[b] 8-inch CPVC Tee, 90° Ell, and 45° Ell are rated at half the pressure shown.

ABS pressures refer to unthreaded pipe only.

For service at higher temperature, multiply the pressure obtained from this table by the correction factor from Table 2.

NR is not recommended.

Table 4. Temperature-Correction Factors for Thermoplastic Piping Operating Pressures

Operating Temperature, °F	Pipe Material PVC	CPVC	Polypropylene	PVDF
70	1	1	1	1
80	0.90	0.96	0.97	0.95
90	0.75	0.92	0.91	0.87
100	0.62	0.85	0.85	0.80
110	0.50	0.77	0.80	0.75
115	0.45	0.74	0.77	0.71
120	0.40	0.70	0.75	0.68
125	0.35	0.66	0.71	0.66
130	0.30	0.62	0.68	0.62
140	0.22	0.55	0.65	0.58
150	NR	0.47	0.57	0.52
160	NR	0.40	0.50	0.49
170	NR	0.32	0.26	0.45
180	NR	0.25	[a]	0.42
200	NR	0.18	NR	0.36
210	NR	0.15	NR	0.33
240	NR	NR	NR	0.25
280	NR	NR	NR	0.18

[a] Recommended for intermittent drainage pressure not exceeding 20 psi.

NR = not recommended.

For more detailed information concerning the properties of a particular plastic pipe formulation, consult the pipe manufacturer or The Plastics Pipe Institute, 355 Lexington Ave., New York, N.Y. 10017.