INTRODUCTION

The Dayton Die Cushions Machine Products Handbook is a newly organized handbook that will be devoted to Die Cushions, Press Accessories and other products manufactured by Dayton Die Cushions. Its purpose will be to provide needed information regarding the design, manufacture, sale, installation, and servicing of this equipment; as well as advice and suggestions that will assist in its selection and use.

The Handbook is organized on an "open" basis; meaning that there is no firm predetermination as to contents or format. Topics will be covered, generally, in the order of their relative importance, respective of the needs of handbook users; and materials will be issued as they are completed, without adherence to any particular schedule. The format used for any given topic will be selected in terms of the topic itself, with due consideration to the intended scope and purpose of its coverage, and its relation to the rest of the Handbook.

Overall, the Handbook will be divided into three primary "Parts", each of which will contain a different type of material. These Parts will be:

a. Part One - This Part will be devoted to general discussions of basic subjects, such as for instance the principles of Die Cushion Installation Design. These discussions will attempt to explain what our various products are, how they are made, and how they are used.

b. Part Two - Here we will concentrate on "how-to-do" information, such as "how to select Die Cushions", or "how to measure presses".

c. Part Three - This Part will be used for coverage of topics of a more specialized or more technical nature than would be consistent with the purposes of the first two Parts of the Handbook, and its content and format will be more closely related to engineering activities.

Each Part of the Handbook will be divided into chapters, or into chapter-like collections of bulletins and data sheets where chapter organization is not practical. These will be identified numerically within each part, and arranged in numerical order. Consecutive numbering of pages will be undertaken only within each individual chapter, with each having its own numbering sequence. This will require that all page identifications be given in three parts, with the first two indicating the Handbook Part and Chapter, respectively. Thus, in Part One, Chapter One, page 17 is identified as "page 1-1-17". This three-part numbering system will also be applied to sections and figures. Thus, in the same chapter, Section Two is identified as "Section 1-1-2", and Figure 6 as "Fig. 1-1-6".

In placing new material in the Handbook, the general rule will be to place new pages according to numerical order, treating each of the three parts of the page numbers individually. For example, the following are in proper numerical order:

1-1-2, 1-4-3, 1-8-1, 2-1-7, 2-3-4, 3-2-5, 3-2-16, 3-11-2, 4-1-3, 4-3-4, 4-6-1, etc.

We should point out that there will be no guarantee that chapters and pages will be issued consecutively, so that gaps will be normal, and should be expected. At any time that additional information on the numbering system is needed to assure correct arrangement of material, this will be supplied in the form of a supplement to this Introduction.

1-1-1: PRESSES AND PRESSWORK

In simple terms, a "press” is a mechanical device designed to apply large amounts of force to the cutting and shaping of metals and other materials. Sometimes this force is applied slowly, over a period of time; but more often it is applied very quickly, with the work being done in a fraction of a second. The principle of very rapid application of force, which is characteristic of the presses we will be concerned with in this handbook, dates back for many thousands of years, and a modern press in its bare essentials is merely an improvement and sophistication of the old blacksmith's hammer and anvil. In Fig. 1-1-1 we have indicated this relationship, with the sketch on the right illustrating the fundamental operation. Here one surface of a piece of metal is held against a stationary part of a press and a moving part of the press is driven against the other surface. This operation is basic to all presses, and the many different structures and devices that are built around it are for the purpose of using it to the best advantage in performing different types of work.

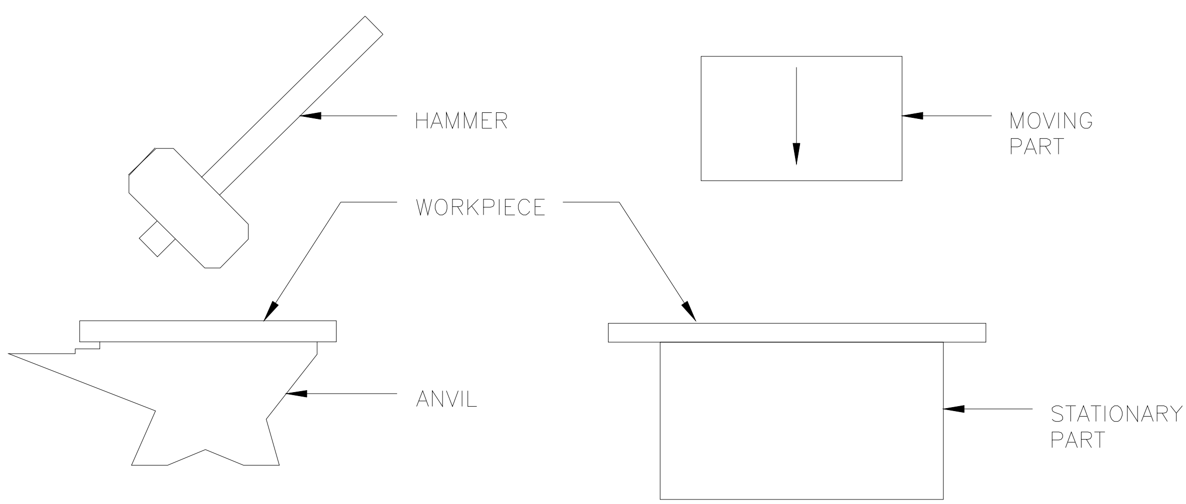


Fig. 1-1-1 Basic press operation

In Fig. 1-1-2 we have provided a rough sketch of a typical modern press in order to show how this principle is incorporated. The stationary part referred to above is known as the "bed" of the press. The actual working surface is provided by a thick steel plate known as a "bolster plate" fastened on top. The moving part referred to above is known as the "ram". This is driven downward by one of several types of mechanical systems using energy supplied by the motor and flywheel. The purpose of the "frame" is partly to support the upper parts of the press, and partly to hold it firmly together when the ram strikes the work piece. This permits the press to exert a squeezing force as well as a hitting force. The importance of this squeezing force varies with the application of the press, but it is present to some degree in all applications; it represents a situation not present in the use of a hammer and anvil, and it increases both the potential for useful work and the potential for mechanical problems.

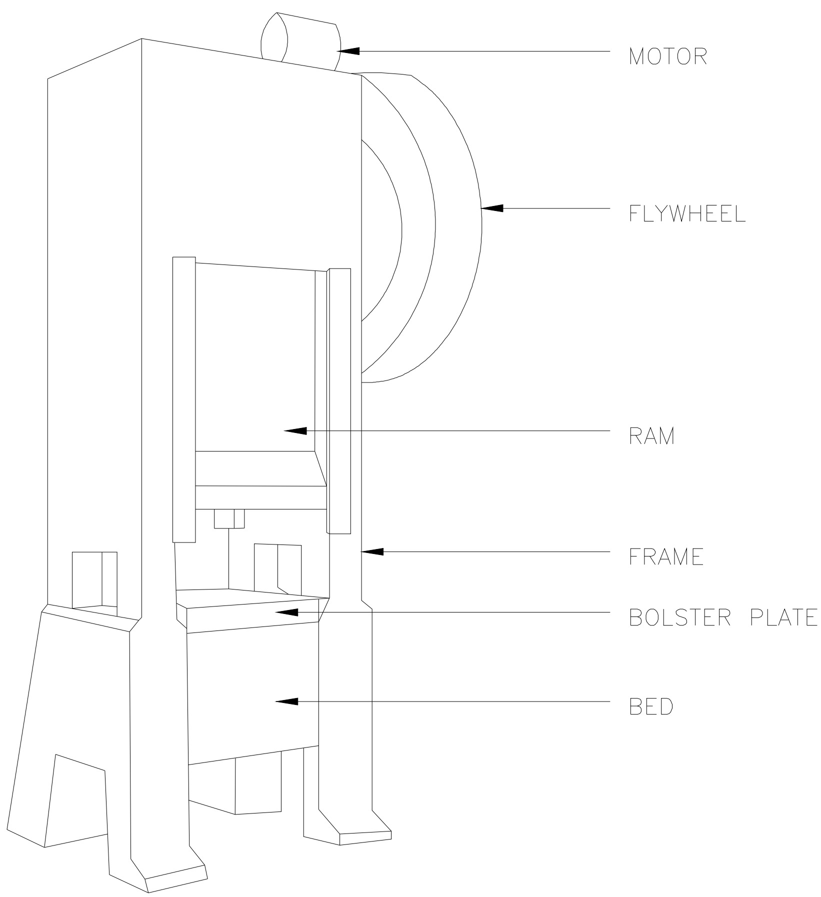


Fig. 1-1-2 Press parts relating to the basic press operation

The actual working of metal (the workpiece) is accomplished by specially designed parts known collectively as “tooling". Part of the tooling is fastened to the bolster plate, and part is fastened to the ram. These parts are so constructed that they will "mate" in such a way as to perform a particular operation on any piece of metal placed between them. If the tooling is so designed that in mating one part is moved inside of the other, this part is usually called a "punch", and the other part a "die". This is illustrated in Fig. 1-1-3a. However, these labels are not always easily applied, and it is often better to use the terms "upper tooling” and "lower tooling", as illustrated in Fig. 1-1-3b.

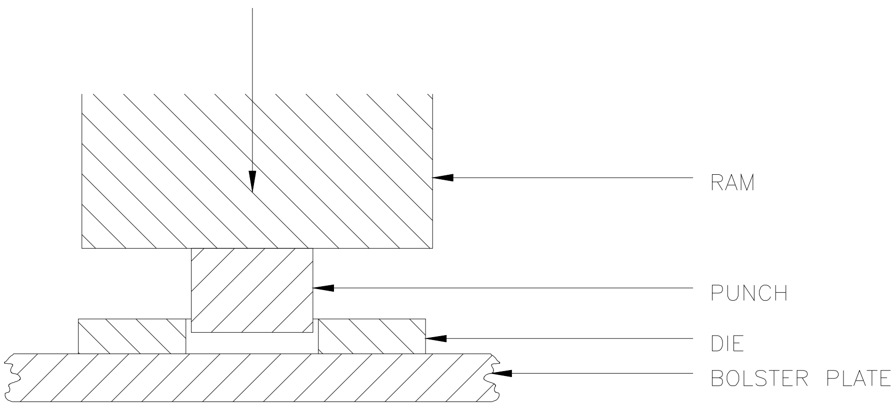


Fig. 1-1-3a Punch and die

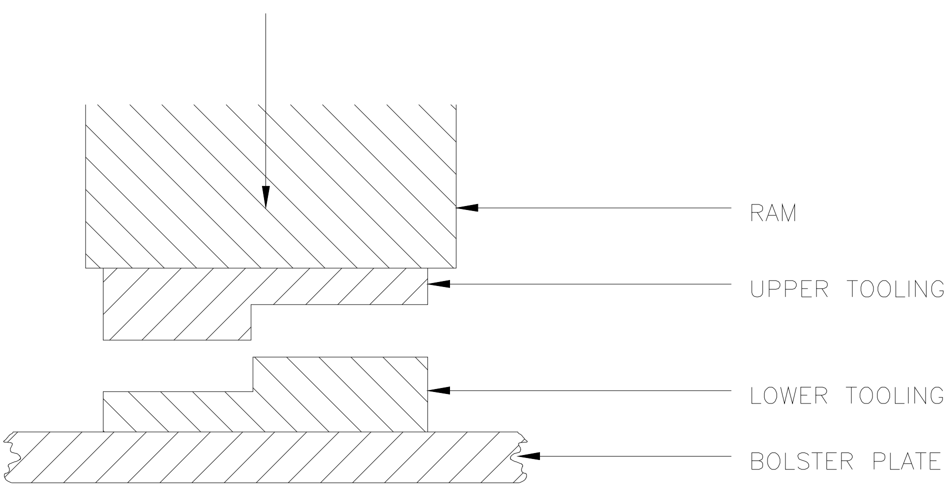


Fig. 1-1-3b Upper and lower tooling

DAYTON DIE CUSHIONS

"Cutting" in presswork is usually known as "shearing", and "shaping" is known as "forming" or "squeezing". Shearing operations are usually divided into three categories, "blanking", "piercing", and "cutting"; forming operations are less clearly distinguishable, but are usually divided into the categories of "bending". "Forming", and "drawing": and squeezing is usually divided into the categories of "forging", "coining", and "extruding". In order that the reader may better understand the nature of presswork we will discuss each of these briefly.

Shearing operations:

a. **Blanking** — This involves the cutting of the contour of a part from a plain flat piece of metal. An example is illustrated in Fig. 1-1-4a.

b. **Piercing** — This operation punches a hole in a flat piece of metal, often after the contour has been blanked by a previous operation. Blanking and piercing are frequently done simultaneously. Again, refer to Fig. 1-1-4a.

c. **Cutting** — Sometimes regarded as a simple variety of blanking — usually involves the cutting of a straight line, or a single irregular edge, such as night be done by a cut-off die.

Forming operations:

a. **Bending** — Often included under forming — as the name implies, a simple bending of a flat piece, usually after blanking and piercing have been performed. A bent piece is illustrated in Fig. 1-1-4b. A single bend may be known as a "vee bend" or "vee form", two bends fairly close together as a "u form", *and* a gradual bend with a large radius as a radius form. Other specific varieties are seldom given individual names.

b. **Forming** — Very often used to include bending as well — In stricter usage refers to operations in which bending is accomplished along curved lines, and around corners. Roughly, a more complicated variety of bending. See Fig. 1-1-4c.

c. **Drawing** — Here a cup-like part is formed from a flat piece by forcing metal to flow in three dimensions. The results are usually referred to as shells or drawn shells; an example is illustrated in Fig. 1-1-4d. Drawing is normally done in a series of steps, with different tooling for each, in which the height of the sides is progressively increased while the diameter of the shell is progressively decreased. Since drawing operations afford the most frequent application of die cushions, they constitute as a group the operations with which we are most concerned.

Squeezing operations:

a. **Forging** — This is the most general of the metal squeezing operations. The tooling constitutes a variety of mold; and metal, in some cases hot, and in some cases cold, is forced into it. A typical forged piece would be a large bolt, such as illustrated in Fig. 1-1-4e.

b. **Coining** — This is a simpler metal squeezing operation than forging. An example would be a coin such as is illustrated in Fig. 1-1-4f. In general, the category includes those operations in which an impression is forced into a flat piece of metal.

C. **Extruding** — Here metal is made to flow through a die so as to acquire a particular cross section; generally this is the most difficult of all press operations. Extruding and forging are sometimes combined in a single operation; hence it may occasionally be difficult to differentiate between them. Fig. 1-1-4g illustrates a simple extruded part.

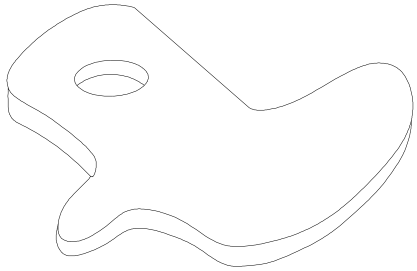


Fig. 1-1-4a Illustrations of presswork: blanked and pierced

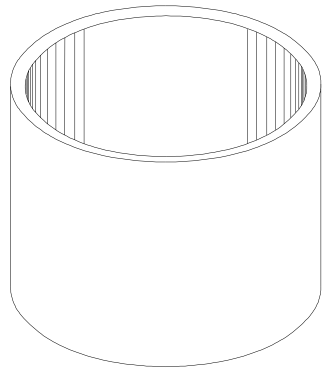


Fig. 1-1-4d Illustrations of presswork: drawn

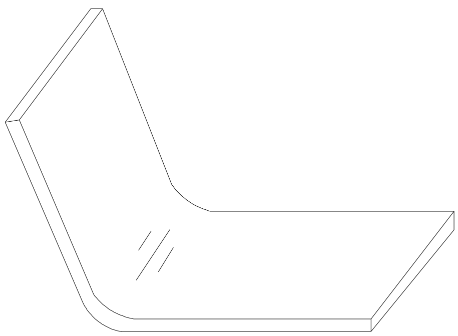


Fig. 1-1-4b Illustrations of presswork: bent

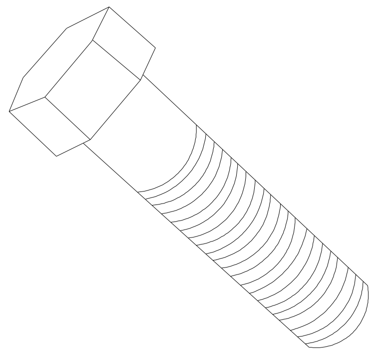


Fig. 1-1-4e Illustrations of presswork: forged

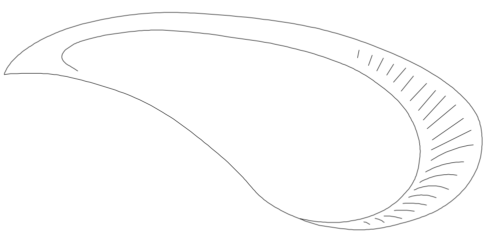


Fig. 1-1-4c Illustrations of presswork: formed

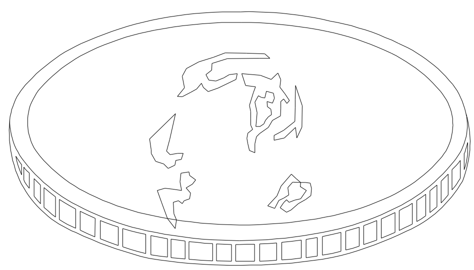


Fig. 1-1-4a Illustrations of presswork: coined

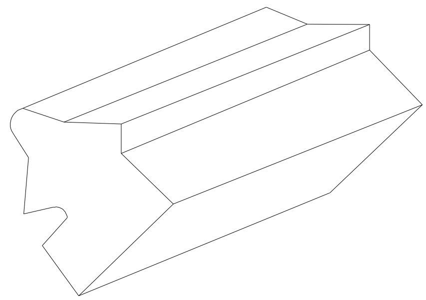


Fig. 1-1-4a Illustrations of presswork: extruded

It should be noted that the general terms, shearing, forming, and squeezing, are seldom used outside of theoretical discussions such as the present one. Normally the specific name of the operation as given above is used. This avoids confusion that might arise in conversation, since shearing is also a specific operation performed by a shearing machine, forming has a more specific meaning as a group of press operations between bending and drawing, and squeezing is too vague for shop use. However, the general terms should not be discounted entirely; they may be quite useful on occasion. Insofar as die cushions are concerned, four terms — **blanking, piercing, forming, and drawing** — are the only ones likely to be encountered with any frequency; and if the reader makes on effort to understand these he is not likely to have any trouble.

1-1-2: THE PARTS OF A PRESS

The purpose of this section is to provide the reader with a basic knowledge of the parts of a press and their relationships to each other. In a sense, most of this information could be regarded as of a "critical" nature, since it is likely that the reader will need to be familiar with it if he is to engage in profitable conversations with plant personnel pursuant to the selling and installing of die cushions. However, this is by no means to be a complete rundown on press construction and operation; we will cover only a few essential points, and the render should encounter no difficulty with any of them.

Of the many things that are to be noticed about a press, perhaps the first is its size. Presses range in size from small bench-mounted models to large "monster" varieties several stories high; and each has its own special role to play in the overall manufacturing picture. There is in this respect an obvious rough correlation between the overall size of a press and the size of the work it is able to turn out. However, for our purposes here, size involves more specific things than this; and it is these specific features with which we are most frequently concerned. In general these involve dimensions of press parts and openings, and such things as tonnages and operating rates; they may also involve ratios of the size of one part to another, etc.

The second thing that we should notice about a press is its most general construction features. The primary consideration in the present context involves the difference between the **straight side press** and the **open back inclinable** (OBI). Figure 1-1-5 illustrates the principal parts of a typical straight side press; and Fig 1-1-7 illustrates the principal parts of a typical inclinable press. We will discuss these two varieties of press in that order.

Straight side press

Referring to Fig 1-1-6, we should take note of the following parts and construction features of straight side presses:

a. **Bed** — As indicated in the previous section, this is the basic stationary part of the press. It provides support for the bolster plate, and provides an enclosure for special components, the most important variety of which at present are die cushions.

b. **Bolster plate** — This is a heavy steel plate that is fastened to the top of the bed, and provides the basic working surface. Bolster plates are removable, which facilitates modification, refinishing, or replacement. They normally contain a great many holes, tapped and untapped, which are used for fastening down tooling.

c. **Foot** — The purpose of the feet on a press is obviously to support it, but they also serve a secondary purpose of determining the height of the working surface. Thus on small presses the feet are usually proportionally quite long, whereas on large presses they are proportionally quite short, and the bed may extend a considerable distance below them, necessitating that the press be installed over a pit.

d. **Frame** — Essentially, the frame on a straight side press consists of three pieces, two uprights and one crown; variations are however quite possible. These three pieces and the press bed are normally fastened tightly together with tie rods, which are long steel rods, threaded at the ends, that pass through the crown, the uprights, and a portion of the bed; nuts are placed on both the upper and lower ends of these rods to hold the press together. In order to obtain maximum rigidity, tie rods are heated when installed, and the nuts turned on tightly while the rods are still hot. As the rods cool, they shrink, and tend to pull the press parts more firmly together. This operation is known as prestressing, and its purpose is to minimize distortion of the frame, and hence possible deflection of the tooling that may occur while the tooling is in contact with the workpiece during press operation.

e. **Ram or Slide** — These two terms may be used interchangeably. This is the basic moving part of the press; its lower surface has provisions for the mounting of tooling.

f. **Gibs or Ways** — These two terms may also be used interchangeably. These parts provide guidance for the ram as it moves up and down. Their purpose is to see that it moves along precisely the same vertical line on each and every stroke of the press.

g. **Motor** — This is the primary source of energy for the operation of the press; it is usually mounted on the top of the crown.

h. **Flywheel** — The flywheel is a large, heavy wheel connected through V-belts to the motor. Its fundamental purpose is to store energy for use in operating the press.

In order to better understand the relationships between the moving parts of a press, we should take a moment to consider the way in which rotary motion, produced by the motor and transferred to the flywheel, is converted to the linear or reciprocating motion of the ram. Referring to Fig. 1-1-6, we see that three fundamental parts are involved, a rotating part, a reciprocating part, and a connecting part. The rotating part in this figure is a crankshaft; this has an eccentric or crank that moves in a relatively large circle around the basic shaft. The crank is attached to the connecting part, which we call a pitman, by means of a journal-and-bearing arrangement; i.e., the crank rotates within a bearing in the pitman as it revolves around the basic shaft. The pitman, which by the motion of the crank is forced to move from left to right to left in the figure, is connected to the reciprocating part, which is the ram, by means of a flexible joint, usually involving a ball or hinge. The ram is then guided by the gibs into a straight reciprocating motion.

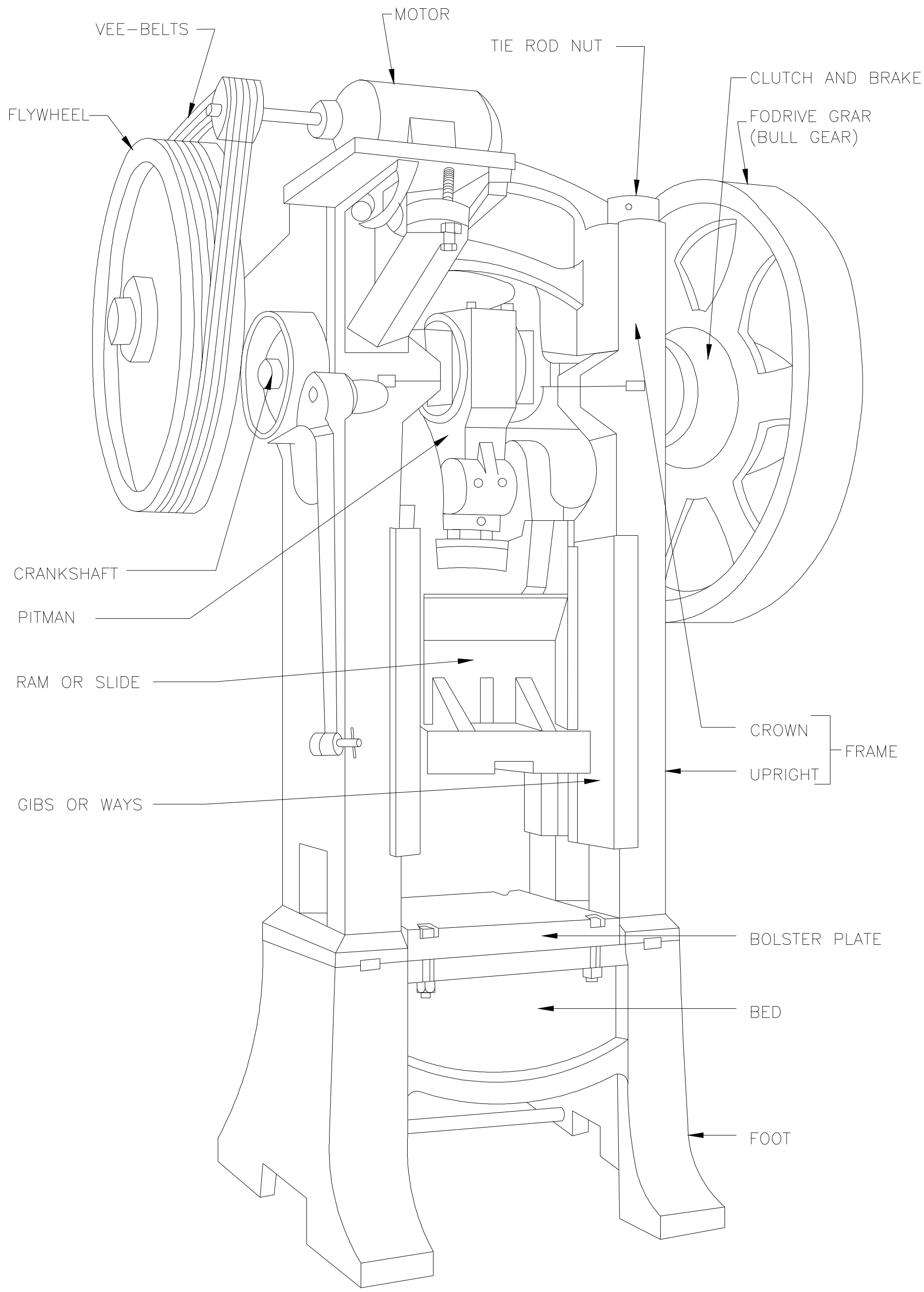


Fig. 1-1-5 Typical straight side press

Referring again to Fig. 1-1-5, we conclude our discussion of straight side presses with the following parts:

i. **Clutch and Brake** — The clutch forms a vital link between the flywheel, which must be in constant motion, and the crankshaft, which must revolve only when the press is actually doing work. When the clutch is engaged, the flywheel and crankshaft are connected, and the rotary motion of the flywheel is converted to the reciprocating motion of the ram; when the clutch is disengaged, the flywheel is able to continue its rapid rotation while the ram remains stationary. The brake stops the rotation of the crankshaft when the clutch is disengaged.

j. **Drive gear** — sometimes the clutch connects the flywheel directly to the crankshaft; at other times, however, this connection is indirect, with the crankshaft being connected to an intermediate rotating member known as the drive gear. The purpose of the drive gear is to convert the very rapid rotary motion of the flywheel to a slower rotary motion, more suitable for the activation of the crankshaft. One or more drive gears may be involved in this conversion, and a variety of mechanical arrangements have been devised to accommodate them. — Drive gears are often called Bull Gears.

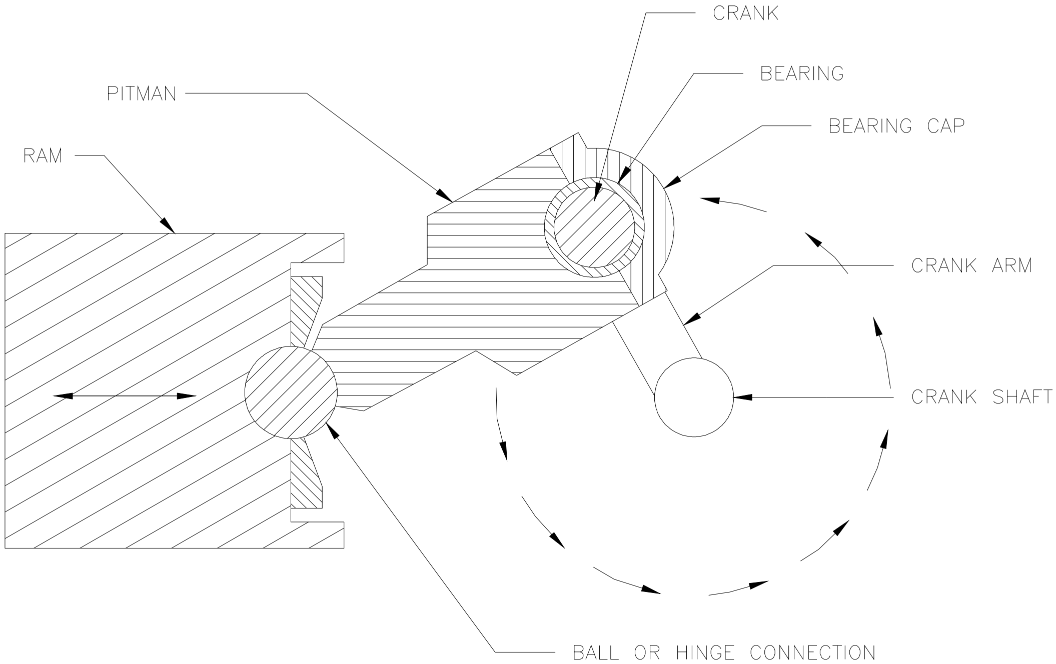


Fig. 1-1-6 Converting rotary motion to reciprocating motion

Open back inclinable press

The open back inclinable (OBI) press differs from the straight side press in a number of ways, the most important of which is the provision made for inclining the working area. Our discussion of OBI presses will not be as complete as that of straight side presses, but will cover only those points of significant difference. Referring to Fig 1-1-7, we may note the following construction features:

a. **Frame** — Essentially the frame is a one-piece combination of the crown, uprights, and bed, although it may be divided into right- and left-hand parts for some designs. Viewed from the side it will be seen to be roughly semi-circular, with the arc broken around the working surface. This gap in the arc of the frame gives rise to the term gap frame; gap frames are characteristic of OBI presses, as well as a number of varieties of non-inclinable presses. Note the great depth from front to back of the sides of the frame; this is intended to minimize deflection of the ram during the working portion of the press cycle.

b. **Bed** — As mentioned above, this is essentially part of the frame; however, it is clearly distinguishable, serving much the same purpose as on straight side presses; i.e., it supports the bolster plate, and provides a cavity for die cushions and other devices.

c **Base** — This may be of one or more pieces. Note that it must be able to provide ample support for the press when it is in an inclined position.

d. **Inclining mechanism** — This consists of an inclining screw, which may be located at the back of the press, or on one side; and bolt-and-slot combinations on the rear of the sides of the base. The inclining screw provides for adjustment of the angle of inclination, and the bolt-and-slot combinations provide a means of locking the frame in the selected position.

The advantages of the OBI design are two-fold: first — there is the greater accessibility of the working area, characteristic of the gap frame construction; and second — there is the ability to make use of gravity in removing parts from the work area through the use of the inclining feature. The weakness of this type of press is in its relative instability and tendency to excessive deflection when compared with the straight side design.

In addition to the construction features involved in the above designs, there are a number of less common arrangements with which the reader is likely to have occasional contact. We may list these as follows:

a. **Double- and triple-acting presses** — The terms single-acting, double-acting and triple-acting refer basically to the number of slides in a press. The presses discussed above have only one slide and hence are single-acting. Double-acting presses have two slides, and triple-acting presses have three slides; obviously the mechanical arrangements for these presses are much more complicated, even though the fundamental principle of press operation remains the same.

b. **Horn presses** — some presses are designed so as to permit an adjustment of the position of the bed; the bed in this case being a movable part that may be locked in a number of different positions. On such presses it is usually possible to remove the standard bed and replace it with a horn-shaped bed that will accommodate curved and cylindrical workplaces; hence the name of the press.

c. **Under drive -presses** — on presses of this type, motors, flywheels, and drive gears are all located in or beneath the press bed, and. the ram is activated by means of shafts and levers. In some designs the positions of the ram and bed are reversed, with the stationary part being above, and the movement of the ram in doing work being upward.

d. **High-production presses** — Essentially these are presses designed for extremely high operating speeds. However, other features such as automatic control, increased tool life, and reduced maintenance costs are also bought, the primary objective being the achievement of a low piece-price. Such presses are usually of a much heavier construction than conventional presses of comparable tonnage ratings, and are equipped with a greater number of accessories, such as automatic feeding devices and lubricating systems.

e. **Transfer presses** — Fundamentally; a transfer press is a miniature production line. It contains a number of different stations, which are tooling combinations designed to perform different operations on the same workpiece in a progressive sequence. The workpiece is transferred from station to station within the press by automatic feeding equipment, and ejected after the last operation has been performed. Such presses are quite expensive, and their use is normally justified only by a requirement for a very large output, often well in excess of one million pieces.

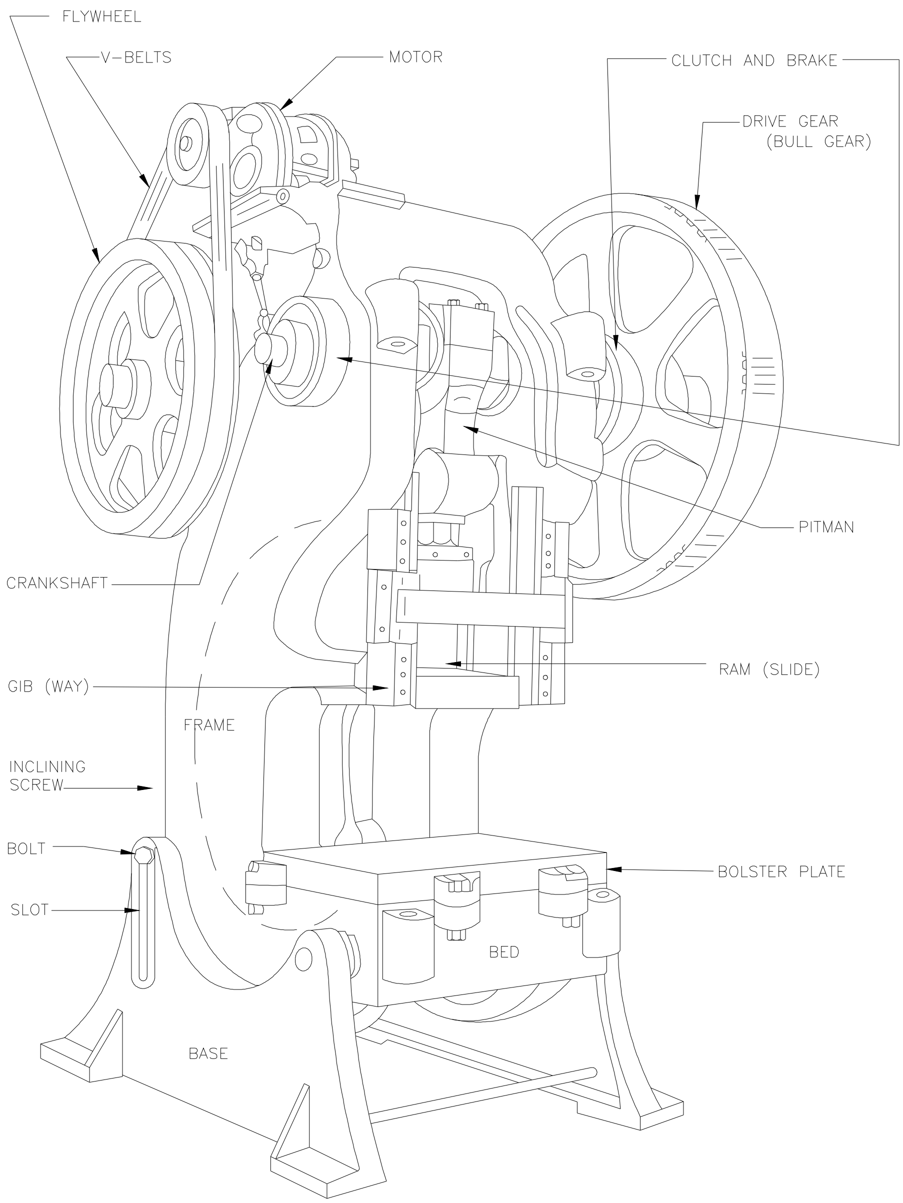


Fig. 1-1-7 Typical inclinable press

As a final note on press construction features we should include a few words concerning tonnage ratings. The tonnage of a press operation is the maximum force developed in the course of performing work on the workpiece. This may be adjusted upward or downward by adjusting the length of the pitman or of some other part of the ram or connecting pieces so as to vary the position of the tooling as the ram reaches its lowest point in the press cycle. The tonnage rating of a press itself is a somewhat theoretical figure representing the maximum tonnage for which such adjustments should be made for safe and smooth operation. It does not represent the maximum force that the press is capable of developing.

1-1-3: THE FUNCTION OF DIE CUSHIONS

The term **die cushion** is somewhat of a misnomer, but persists through wide spread use and understanding. The reader will do well to regard this only as a name, and not as a description, From an operational standpoint, a die cushion is a press accessory, and its purpose in a very rough sense is to provide an additional source of force for use in the press operation (i.e., in addition to that provided by the movement of the ram). The vast majority of die cushions are specially designed pneumatic cylinders; however hydraulic cylinders, and hydro-pneumatic combinations are nevertheless quite numerous. In Fig. 1-1-8 we have illustrated a typical pneumatic die cushion; its principal features are as follows:

a. **Simple construction** — Two major parts, a cylinder and a piston, constitute the primary working parts. Note that the combination may be collapsed by moving the cylinder down over the piston

b. **Compact construction** — The entire combination takes up no- more space than is necessary to accommodate the pressurized air it contains; and to provide for mounting flanges, and sufficiently thick walls and plates to withstand the forces involved in its operation.

c. **Spring-like operation** — The pressurized air, held inside by the overlap of the walls of the cylinder and the piston, and sealed in by the packing, tends to expand the combination; thereby resisting any force acting to collapse it, and causing a "rebound" when the collapsing force is removed,

d. **Constant force** — The magnitude of the force with which the cushion resists collapsing depends only upon its cross-sectional area and the pressure of the air confined inside. If the volume of this air is sufficiently large in comparison with the amount displaced as the cushion collapses, its pressure will change only slightly, and the force exerted by the cushion will remain fairly constant.

e. **Adjustability** — Since the force of the cushion depends on the pressure of the air inside, this force may be easily adjusted upward or downward by adjusting the pressure of the air with a pneumatic pressure regulator.

In rough terms, the operation of a die cushion consists of exerting an upward force while being collapsed downward. A secondary operation is that of exerting an upward force while rebounding. A more precise explanation of the basic die cushion operation may be developed with reference to Fig. 1-1-9. Here we see the workpiece held between the upper tooling, which is attached to the ram, and the lower tooling, which is supported by the die cushion, via a steel "cushion pressure pin" that passes through a hole in the bolster plate. The die cushion is mounted in the bed of the press by means of a mounting plate and suitable fastening devices that hold the cushion piston rigidly in place. As the ram of the press moves downward, the force it exerts on the upper tooling is transferred through the workpiece, the lower tooling, and the cushion pin to the cushion cylinder, moving it downward and causing the cushion to collapse. However, as the cushion collapses it exerts an upward force that holds the workpiece firmly between the upper and lower tooling. Note that the workpiece moves downward during this operation. It is this holding of the workpiece while it is moving downward that constitutes the most important feature of the operation. We would note also that since the force exerted by the cushion is adjustable, it is possible to vary the force with which the workpiece is held until it is exactly right for the particular operation being performed on it.

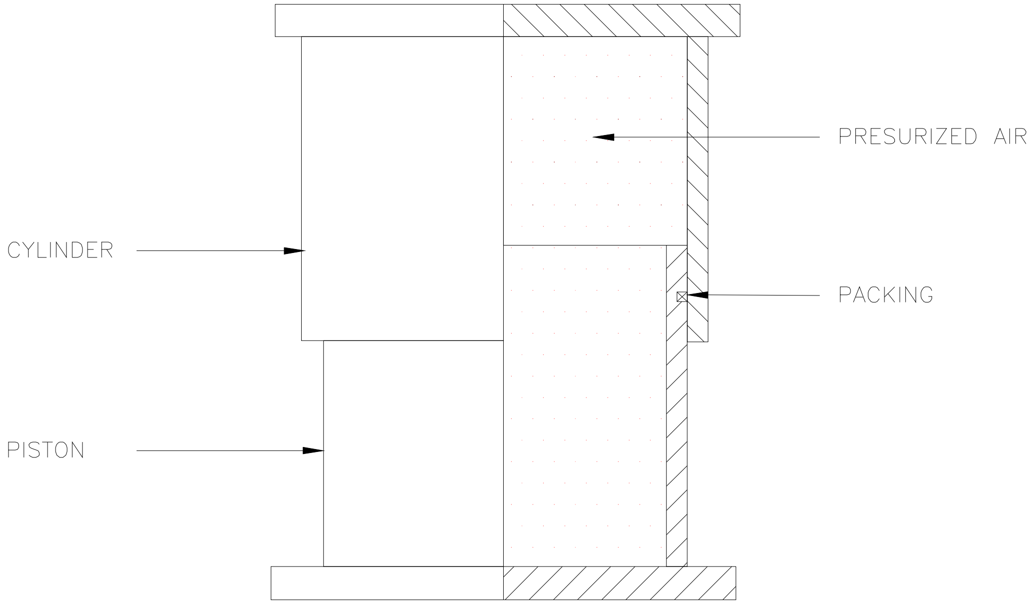


Fig. 1-1-8 Typical pneumatic die cushion

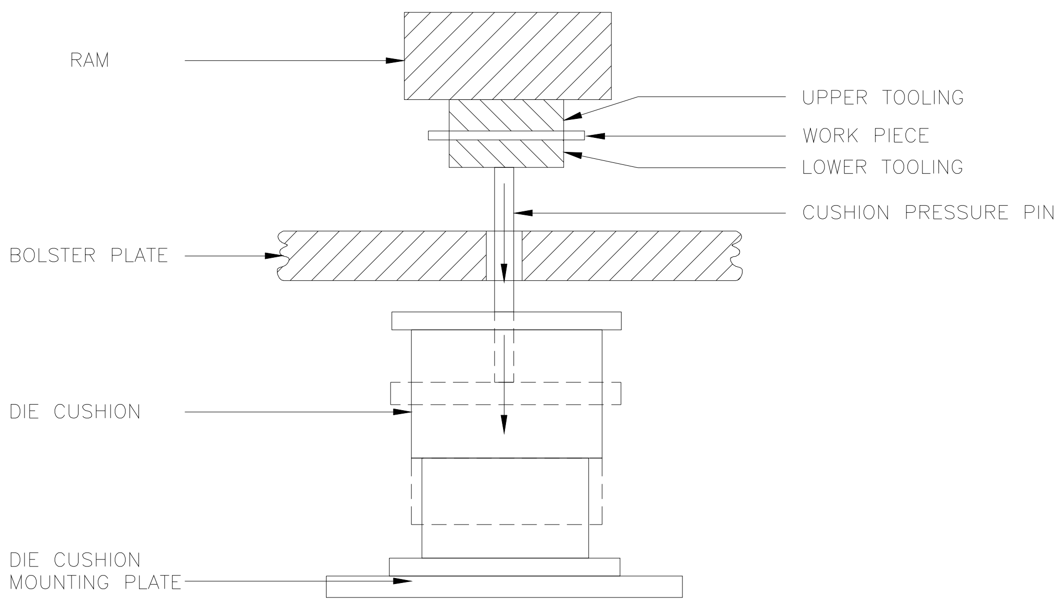


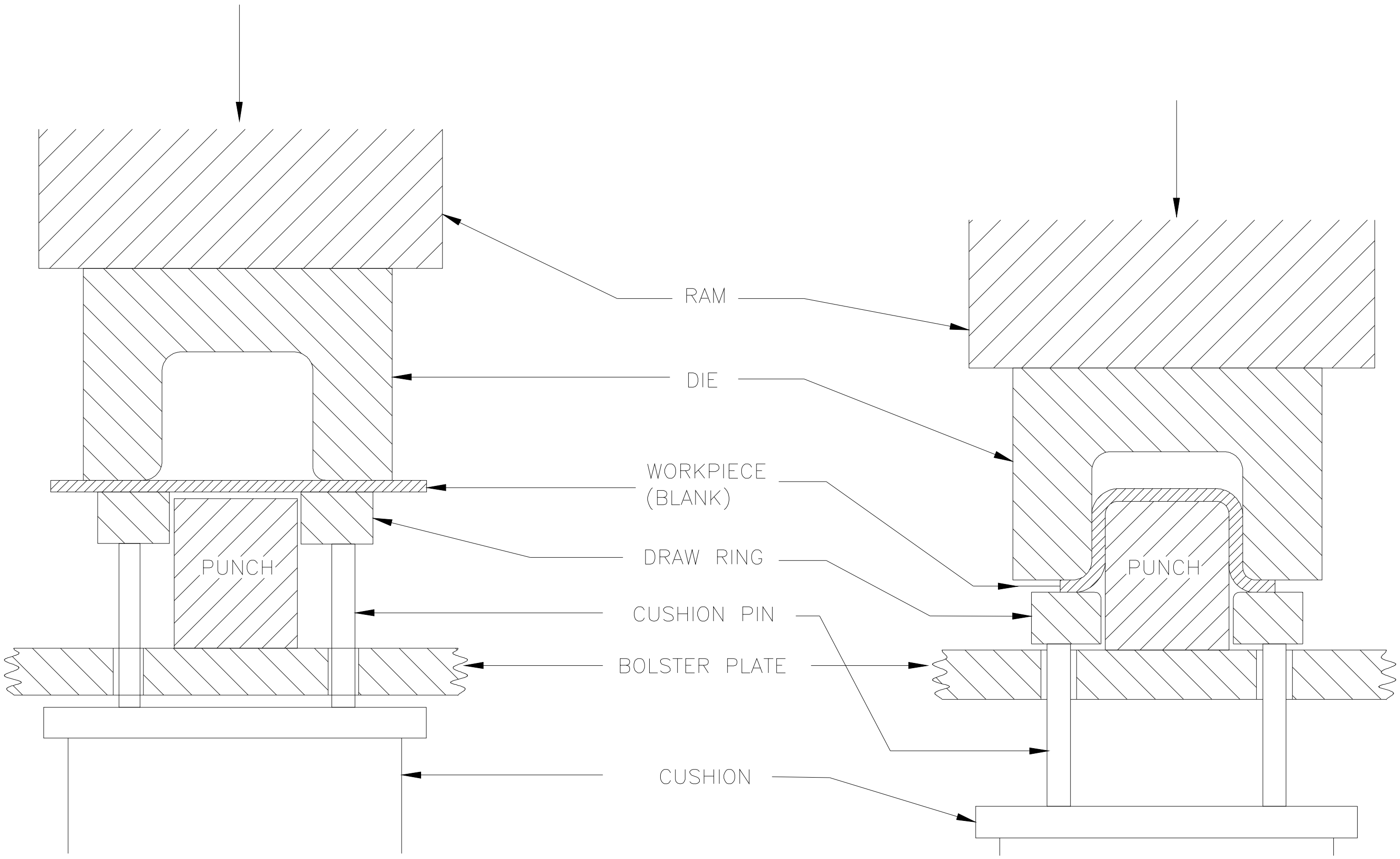
Fig. 1-1-9 Basic die cushion operation

In order to show how this basic cushion operation is employed in actual manufacturing processes we have illustrated in Figs 1-1-10 and 1-1-11 two of the most important die cushion applications. Before discussing these, however, it would be well to point out that these illustrations do not represent complete tooling arrangements. Since our purpose here is only to show the way in which die cushions participate in the press operations, and not to show the functioning of press tooling, many tooling parts and operations have been omitted from the drawings and from the discussion below.

Figure 1-1-10 illustrates an arrangement for producing drawn shells. This type of operation represents the most important die cushion application. In the illustration in Fig. 1-1-10a we see that the upper tooling consists of a die with a cylindrical cavity; and the lower tooling consists of a cylindrical punch, fastened to the bolster plate, and a draw ring supported by cushion pins. As the drawing operation begins, the workpiece, in this case called a blank, is held between the die and the draw ring, and is brought down over the punch. In Fig 1-1-10b we see where the downward movement of the die has forced the blank to deform and flow around the punch, producing a cup-like part. The cushion force for this operation has been adjusted so that the blank is held between the die and draw ring loosely enough that it can slide inward toward the punch and be deformed into the shell, but firmly enough that in so sliding it will not wrinkle or buckle. The adjustment is obviously an important one, and for many operations must be accomplished within close limits; it is also desirable that the holding force exerted by the draw ring be fairly constant throughout. The fact that a die cushion can easily provide for both of these requirements is the primary reason for its great value as a press accessory.

In Fig. 1-1-11 we have illustrated a somewhat simpler application, characteristic of forming operations; in this particular case the operation is simple bending, but the principle can be used for more complex situations. In Fig. 1-1-11a the workpiece (again called a blank) is held between the upper tooling, in this case a punch, and a movable part of the lower tooling, which is supported by a cushion pin. As the blank is brought down over the stationary part of the lower tooling it is forced to bend upward and wrap around the punch, as illustrated in Fig. 1-1-11b. We note that since in this operation little or no sliding of the blank is intended, the force exerted by the die cushion must be sufficient to prevent this.

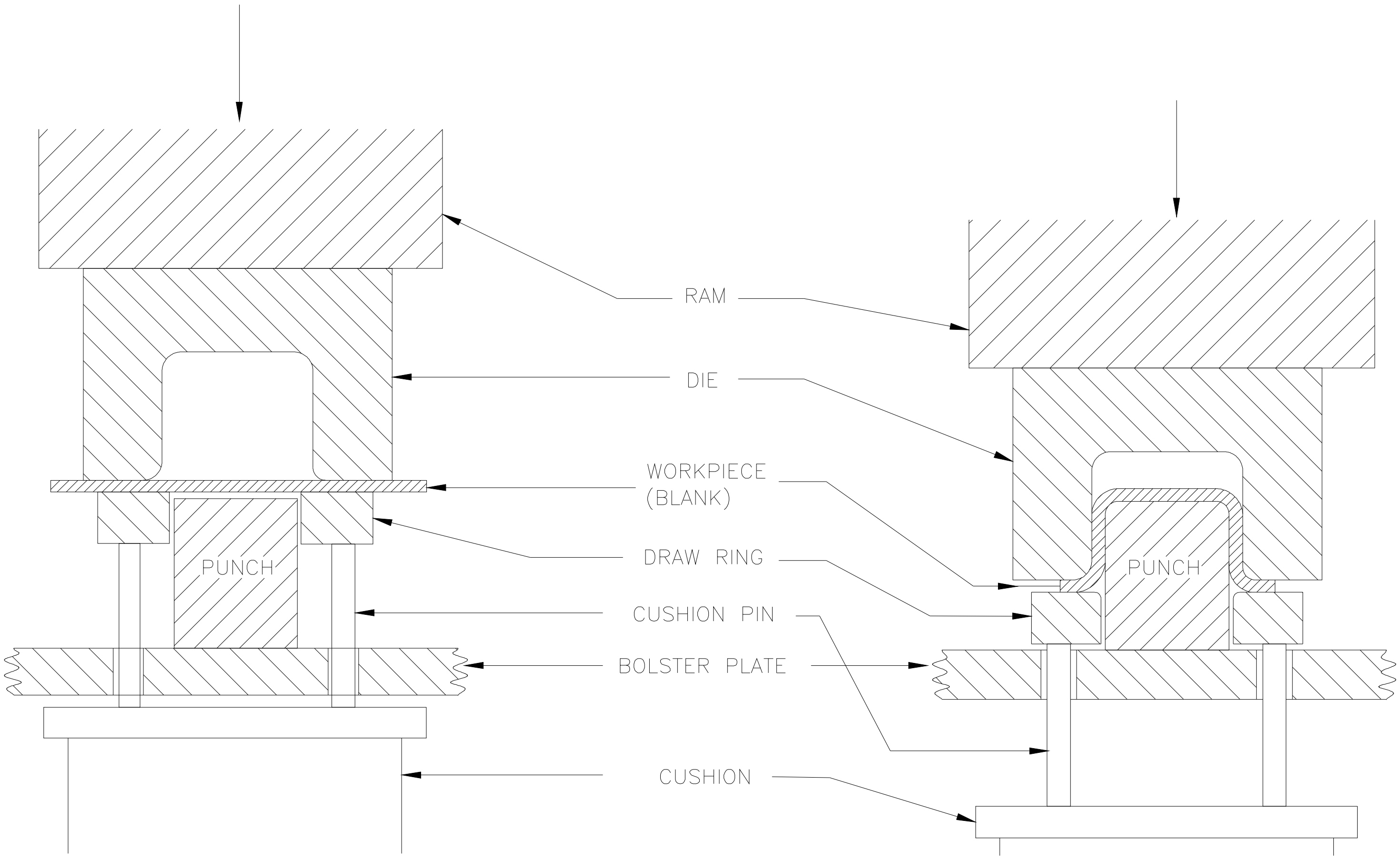
Figure 1-1-12 illustrates a somewhat different type of die cushion application. This is a special arrangement used for many short-run blanking operations, and is called cushion blanking. Here the punch is attached to the bolster plate, and the die is held above the punch by the cushion pins. Sufficient clearance is provided between the punch and die in this position for the workpiece, in this case a piece of stock material, which is slipped, is from the side. An additional piece (not shown in the illustration), called a nest, is fastened to the die from beneath, and serves to guide and support the stock. As the ram strikes the die from above, it forces the die down over the punch and blanks a piece from the stock; at the same time it forces the cushion pins down and collapses the cushion. This operation is shown in Figs. 1-1-12a and 1-1-12b. When the ram moves upward after the blanking operation, the cushion forces the cushion pins upward, lifting the die, the nest, and the stock above the punch. Suitable devices are provided to eject the blanked piece prior to the next cycle of the press. The primary advantage of the cushion in this operation is that it may be adjusted so as to provide only the minimum force required for proper functioning of the tooling. Springs and rubber, which could serve the same general purpose, tend to develop excessive forces as they are compressed, and will greatly decrease the amount of force available for the actual blanking.



(a) Starting to draw

(b) Near completion of draw

Fig. 1-1-10 Drawing

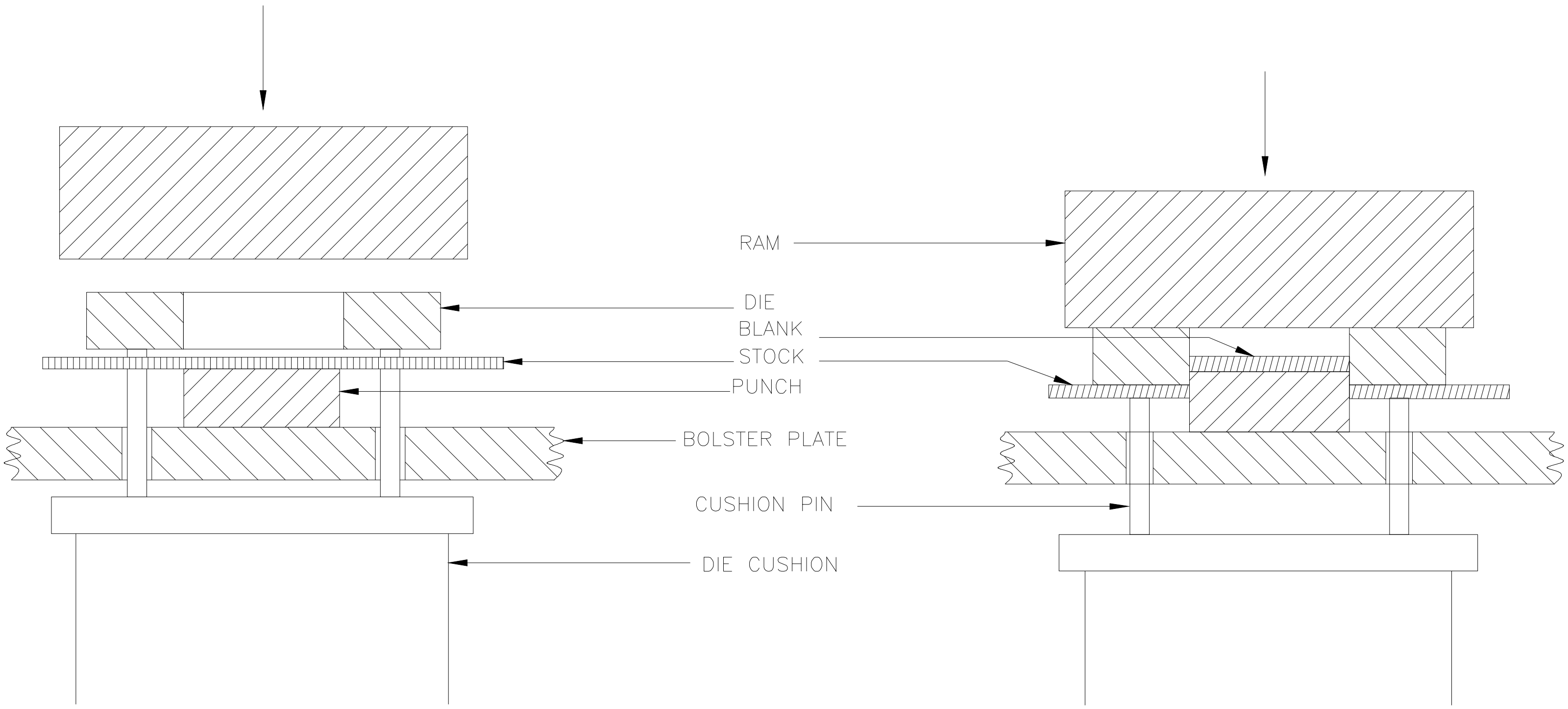


(a) Starting to form

(b) Near completion of form

Fig. 1-1-11 Forming

DAYTON DIE CUSHIONS



(a) Blanking started

(b) Blanking completed

Fig. 1-1-11 Blanking

In Fig. 1-1-10b we note that after the drawing operation is completed, the drawn shell is held inside the work area by the punch, which extends up inside it. Before the shell can be removed from the press it is necessary that it be pushed off the punch- This is accomplished by the draw ring, which is pushed upward by the cushion pins when the ram and die retreat. This removal operation is known as stripping; and because it is accomplished by the upward movement of the cushion, the retain stroke of the cushion is often called the stripping stroke; this is contrasted with the down stroke, which is often called the drawing stroke. In Fig. 1-1-13 we have illustrated the basic stripping action. An examination of Fig. 1-1-11 will indicate that stripping will also take place in forming operations, though it may not be as necessary as in drawing. In some cushion applications, the cushion is employed solely for this stripping action, with the pressure pins performing no useful work on the drawing stroke, but serving to remove the workpiece from the tooling on the upstroke as the first step in the ejection process.

While we have covered the most important applications of die cushions in the above discussion, it is well to observe that the availability of a die cushion on a press, and hence the availability of both the drawing and stripping actions, makes possible a number of simplifications and improvements in many manufacturing processes. Thus, while we stress the use of die cushions in drawing and forming, these by no means represent the limits of die cushion application, and a cushion need not always remain inoperative when a press is being used for some other type of work.

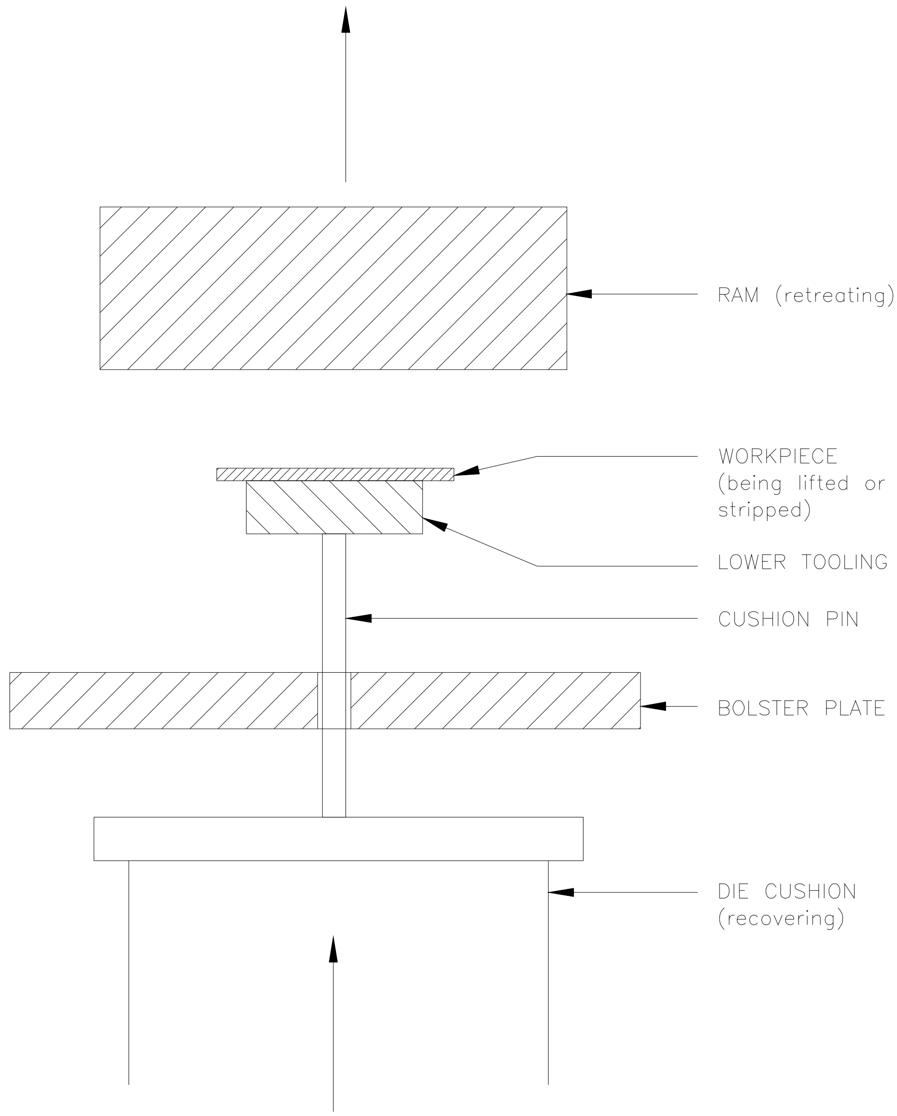


Fig. 1-1-13 Basic stripping action

1-1-4: PUTTING PRESSES AND DIE CUSHIONS TOGETHER

In the discussion of die cushions in the previous section the cushion was depicted as mounted in the bed of the press. While other arrangements, such as ram mounting, are possible, mounting die cushions in the cavity of the press bed is by far the most common. In installing a die cushion in the bed of a press there are three basic problems that must be solved; these we will treat under the headings of positioning, aligning, and supporting.

a. **Positioning** — Roughly, there are two things to consider here; first - there must be a space for the cushion in the bed cavity; and second - there ' must be a way of getting the cushion into that space; both of these are important, and each should be given careful attention. With regard to the location of the cushion inside the bed cavity, we should note that it would be advisable to mount the cushion so that its center is directly below the center of the slide. Otherwise an undesirable condition known as **off-center loading** is likely to occur, which can sometimes lead to a malfunction of the cushion and possible damage to cushion parts. Thus the cushion should be installed roughly in the center of the bed.

b. **Aligning** — In order for the cushion to function properly, the force acting to collapse it should be directed along its centerline; which means that it should act precisely vertically at the center of the top plate of the cylinder. Thus the cushion must be carefully aligned right-to-left and front-to-back with the line of movement of the ram.

C. **Supporting** — supporting the cushion properly means holding it rigidly both in correct position and in correct alignment. Thus it must be firmly attached to some portion of the press. For the cushion discussed in the previous section this requires a firm attachment of the cushion piston either to the press bed or to the bolster plate; the bolster plate being of course itself firmly attached to the bed. These two alternatives are illustrated in Figs. 1-1-14 and 1-1-15, and discussed below.

Referring to Fig. 1-1-14, we should note the following features of a typical press bed mounting arrangements:

1. The attachment of the cushion to the press bed is indirect: the cushion is actually attached to a mounting plate. The mounting plate provides a strong, rigid platform, and greatly simplifies the positioning and aligning of the cushion.

2. The mounting plate is attached to the bed of the press by heavy suspension rods. These are either tapped into the strongest areas of the lower bed, as in the illustration, or are extended through holes in special mounting lugs or ledges and fastened with nuts.

3. A special steel plate, called a pin pad, is fastened to the top of the cushion cylinder. This plate is intended as protection for the cylinder top plate against the wear caused by the cushion pins, and is often hardened, or topped by hardened wear plates or wear strips. It is usually cut to match the opening in the top of the bed.

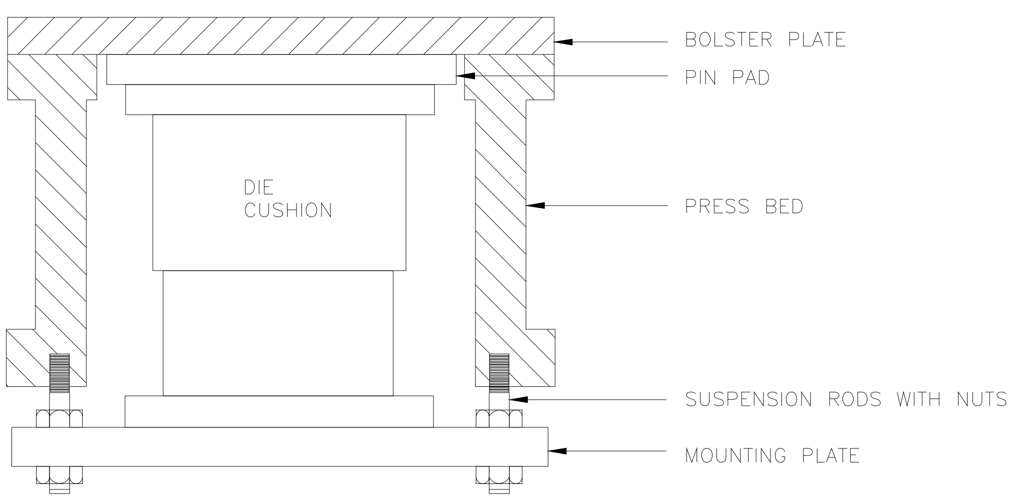


Fig. 1-1-14 Typical press bed mounting

Referring to Fig. 1-1-15, we should note the following features of a typical bolster plate mounting arrangements

a. The attachment of the cushion to the bolster plate is direct, being accomplished by means of special bolster mounting rods that extend through the flanges in the plates of the cushion piston and cylinder.

b. The positions of the mounting rods are fairly well determined by the need to center the cushion; however, most bolster plates are thick enough and strong enough that this presents no problems concerning the strength of the installation, or unsolvable problems concerning the arrangement of tooling.

c. A pin pad is provided as in the press bed mounting. Also, a reinforcement plate beneath the cushion piston is sometimes provided on larger cushions to minimize stresses in the piston flanges.

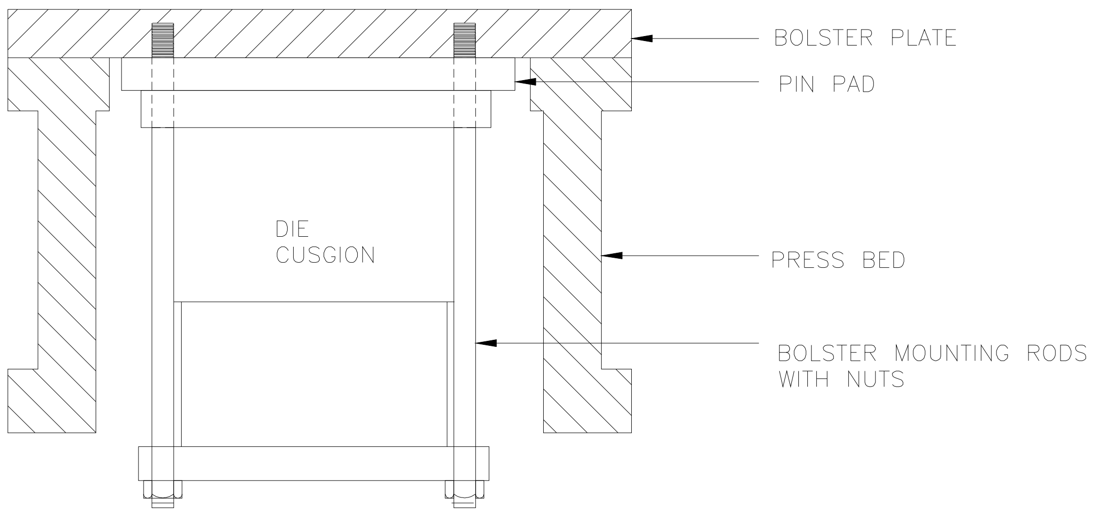


Fig. 1-1-15 Typical bolster plate mounting

In addition to the features discussed above, there is one more point that we should make. On a press bed mounted installation it is possible to remove the bolster plate without disturbing the die cushion; and vice versa; whereas on a bolster plate mounted installation this is not possible. While this situation is of only minor interest to the reader at present, it can have an important bearing on the operation of the press. Thus, we regard the press bed mounting as the more desirable of the two, and try to avoid bolster plate mountings where possible. Of course the fact that bolster plate mountings are usually less expensive than press bed mountings will have some significant bearing on choice, but it should not be allowed to carry too much weight; the price advantage can vanish the first time it becomes necessary to rework the bolster plate.

Besides the three major problems involved in the positioning, aligning, and supporting of a die cushion, there are two others of relatively minor proportions that must also be solved. These involve the surge and lubrication systems. As noted previously, in order for the force exerted by the cushion to be maintained fairly constant throughout the drawing stroke, the ratio of the total volume of pressurized air inside the cushion to the volume of air displaced by the downward movement of the cylinder must be quite large. When a long drawing stroke is desired, this necessitates a cushion of excessively large size. In order to hold the cushion size within acceptable limits, the internal air system is divided into two parts. This is accomplished by placing a large pressure tank, called a surge tank, outside the press and connecting it to the cushion with pipe. Thus as the cushion is collapsed during the drawing stroke, some of the air displaced by the cushion cylinder will flow out of the cushion, through the pipe, and into the surge tank. The location of surge piping can sometimes raise problems that will seriously affect the arrangement of the installation; hence some thought and attention must be given to it when determining the manner and method of mounting the cushion.

As to the matter of lubrication, all standard Dayton die cushions require regular greasing. Grease fittings are provided on the cushion piston, and should be serviced at regular intervals, preferably every eight hours of cushion operation. However, when a cushion is installed in the bed of a press, these grease fittings may not be easily accessible. If there is a suitable opening in the bed of the press, we usually attempt to orient the cushion so that the grease fittings may be serviced through this opening. Otherwise it will be necessary to run grease tubing from the cushion piston to a special header block mounted on the outside of the bed or on the frame of the press. The locating of such tubing can pose problems similar to those related to surge piping, and again it is necessary to give attention to it when determining the arrangement of the installation

While the simple arrangements illustrated in Figs. 1-1-14 and 1-1-15 will provide for a surprisingly large percentage of installations, problems do of course arise. The greatest number of these problems involves some sort of mismatch between the outside dimensions of the die cushion and the inside dimensions of the press bed cavity. If possible, we try in such cases to use modifications of the basic press bed and bolster plate mountings, arrived at by the addition of special structures. In order that the reader may have some rough idea as to the sort of structures that may be incorporated into an installation, we have illustrated two of these in Fig. 1-1-16. In this illustration we have replaced the mounting plate typical of a standard press bed mounting with a mounting structure fabricated by welding together two I-beams, cross pieces, a segment of steel tubing, and a flat plate. Its purpose is to support the cushion higher in the bed cavity than would be possible with a mounting plate alone; such a structure could also be used to provide support strength impractical for a simple plate. On the top of the cushion we have replaced the pin pad with a raised pin pad arrangement, fabricated by welding together two plates and a segment of steel tubing. This permits positioning the pin pad's upper surface at bolster plate level; which is not possible with a simple pin pad, since the top plate of the cushion cylinder is too large to fit up into the bed openings.

As a final comment on the arrangement of a die cushion installation we should point out to the reader that some provision must be made for maintenance and repair. It may perhaps be necessary, for instance, to make an occasional adjustment in the alignment of the cushion, to check and tighten pipe and tube fittings, or to replace these if they become damaged. Replacement of the packing in the cushion, while not a frequent requirement, will nonetheless be necessary on occasion, as well as replacement of internal greasing components that may sometimes become damaged as a result of excessive vibration, or the like. Hence, any installation arrangement that makes maintenance and repair excessively difficult is in general unsatisfactory, and is to be avoided unless circumstances leave no alternative.

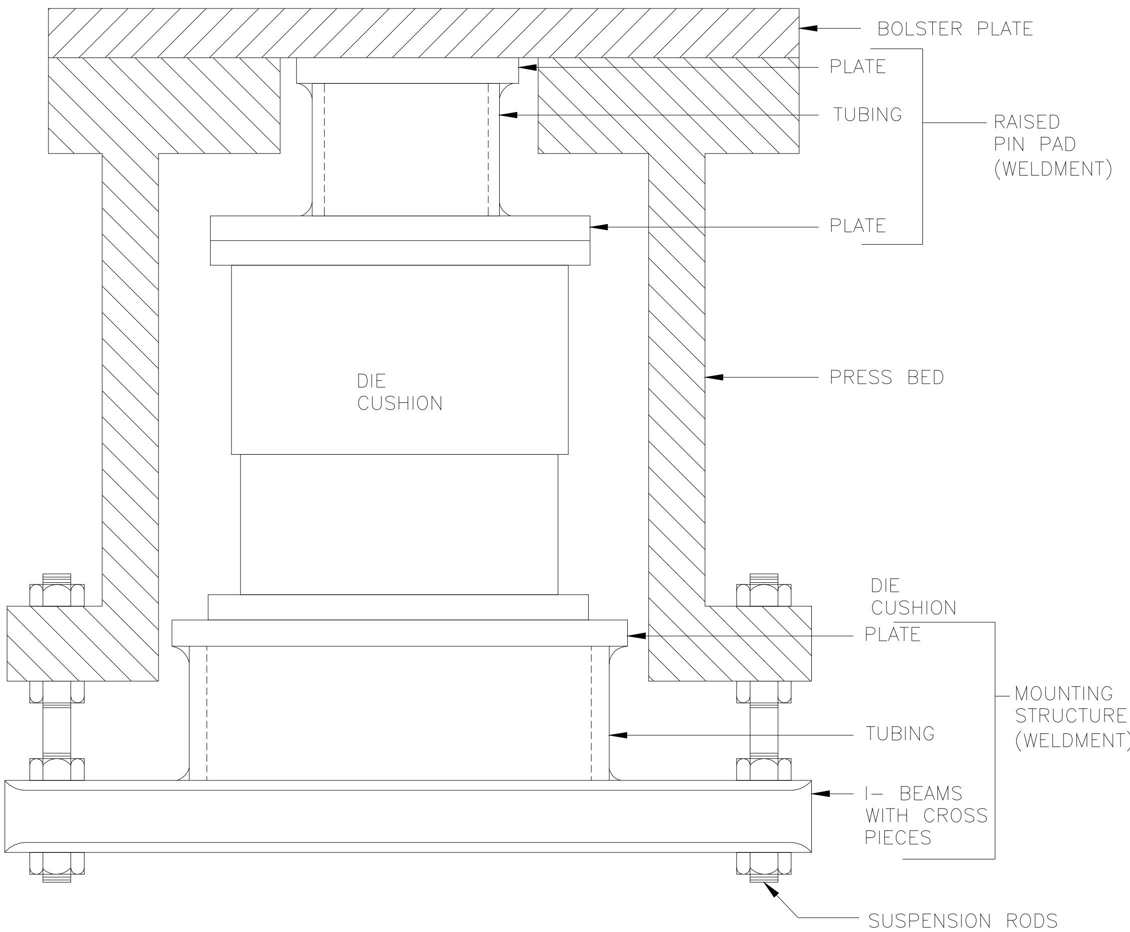


Fig. 1-1-16 Special structures

1-1-5: DIE CUSHION INSTALLATION PACKAGES

As we have noted in previous sections, the installation of a die cushion on a press requires in addition to the cushion itself a number of other items. Collectively, we may refer to these as the installation package. Figure 1-1-17 illustrates a typical installation package, and contains the following items:

a. Die Cushion

b. Pin Pad

c. Mounting Plate

d. Suspension Rods and Nuts

e. Air Supply Line — This consists of one or more pieces of hose, and connects to the shop air line; furnishes pressurized air for the cushion

f. Surge Tank

g. Surge Line (piping from surge tank to cushion)

h. Globe Valves — Basically these are off-on devices used to open and close air lines. They are to be furnished by the customer, and placed in the supply and surge lines

i. Regulator — Placed in the supply line, and usually mounted to the frame of the press or some other suitable place; used to regulate the air pressure in the system, and hence the force exerted by the cushion

j. Pressure Gauges — one for the regulator, one for the surge tank

k. Pop-off Valve — Located on the surge tank; serves as a safety device, and is set to blow when the pressure in the system reaches a predetermined maximum.

l. Drain Cocks — one on the cushion, one on the surge tank; used to remove moisture from the system, or to relieve the air pressure rapidly

m. Grease Tubing

n. Header Block

Those items not discussed above have been discussed in previous sections.

The important thing to remember concerning any installation package is that a die cushion is not complete in itself; the accessory items that go with it are essential to its proper functioning. While some installations will contain more items than those listed above, and others less, this depends upon the relative complexity and application of the cushion, and not upon arbitrary choice.

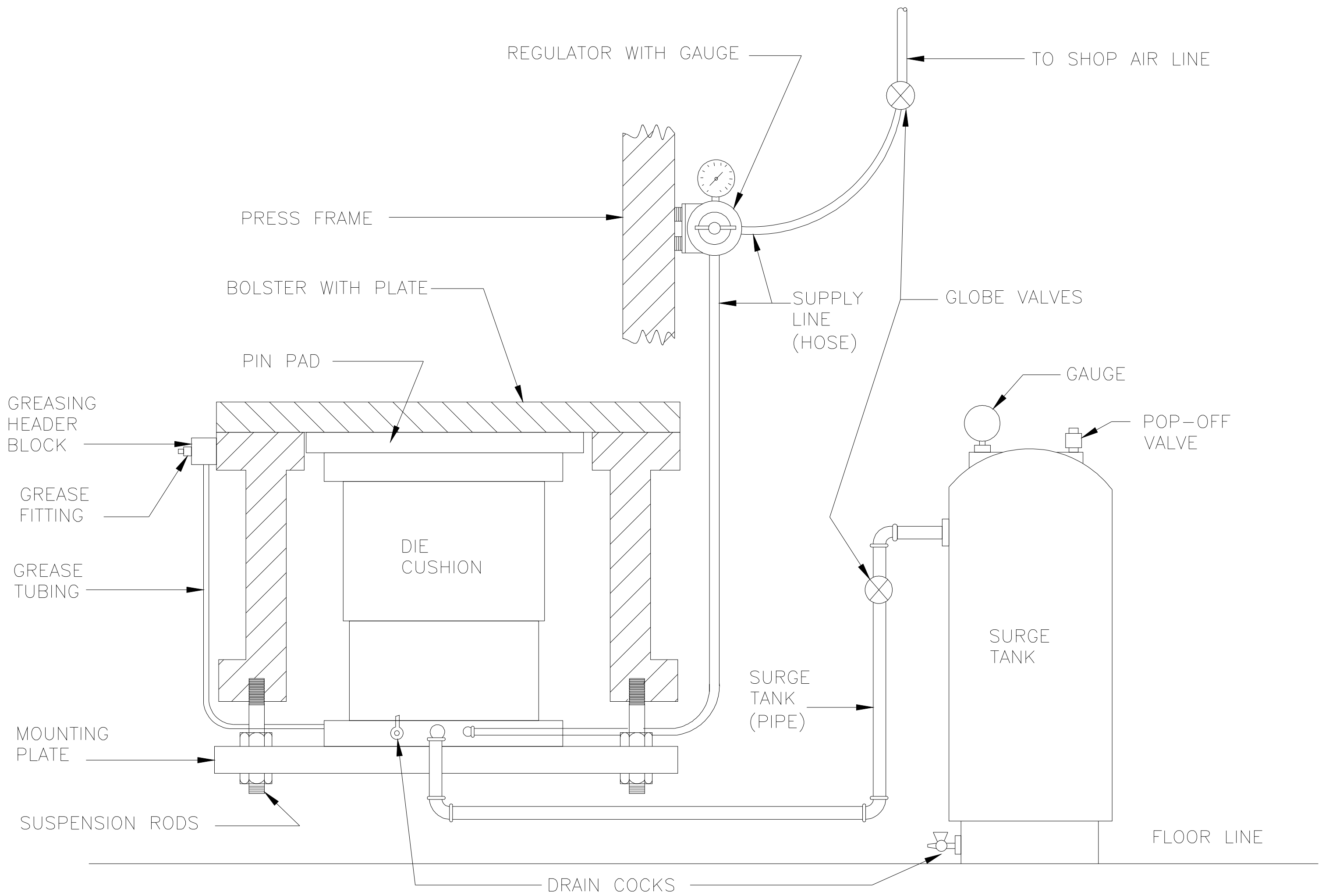


Fig. 1-1-17 Installation package

1-2-1. DIE CUSHION DESIGN FROM A FUNCTIONAL POINT OF VIEW

Before beginning this chapter, the reader should have read Chapter 1-1, and thus should have a general idea as to what die cushions are and what they are intended to do. In the present chapter, we are going to concentrate our attention on the principles and features of basic die cushion designs, in some respects elaborating on the information presented in Chapter 1-1, and in other respects introducing material that is wholly new. In this first section, we will be concerned with various of the operational factors involved in cushion application, and the ways in which they affect cushion design, These factors involve requirements imposed by the work to be done by a press, the space available for cushion mounting, and certain mechanical principles characteristic of this type of equipment.

In considering the work to be done by a press, we may take either of two approaches: we may consider specific jobs for which all basic requirements are known; or we may consider the general capacities inherent in the press design. Since a die cushion is a somewhat permanent addition, it is seldom wise to restrict its application to one or more specific jobs, even when extremely long runs are expected. Thus, we usually take the second of the alternatives, and consider the general capacities of the press. These involve consideration of the following things:

1. The rated tonnage of the press.
2. The length of the press stroke (this is the distance through which the ram moves from top to bottom during a press cycle).
3. The dimensions of the top of the bed.

While these three things do not by any means completely describe a press, they do serve to determine largely the type of work that it can and cannot do and hence they are important guides in selecting a die cushion.

In using these guides, we are attempting to determine three things about a die cushion that would be satisfactory for a given press. These are:

1. The maximum force that the cushion must be able to exert - this is usually referred to as the cushion tonnage.
2. The maximum stroke of the cushion - this is the maximum distance through which the cushion may be collapsed; and for reasons to be noted below is usually referred to as the cushion draw.
3. The size of the top plate of the cushion cylinder; or the size of the pin pad when certain design requirements are met.

The reader will not be asked to understand all the fine points involved in these determinations but there are a number of general ideas with which he should be familiar; and we will present these here.

Since drawing is categorically the most important variety of presswork where die cushions are concerned, the capacities of the press, and hence the capacities of the die cushions, are usually determined with respect to the maximum sizes of drawn shells that the press can produce. This approach is particularly useful in that the requirements for other types of work are usually somewhat less, and we do not therefore place any significant restrictions on other kinds of die cushion applications by concentrating our attention on drawing.

In considering the relationship between press tonnage and cushion tonnage, the first thing we should notice is that cushion force opposes press force, and hence subtracts from the force available for drawing. This has the effect of dividing the force exerted by the ram into two parts: one part depressing the draw ring, and the other performing the drawing operation. This relationship is indicated in Fig. 1-2-1. Next, and closely related, is the fact of the continuing variation of potential press force throughout the press cycle. As the ram descends, this potential force rises from its minimum at the mid-point of the stroke to its maximum near the bottom. Since cushion force is nearly constant, the force required to depress the draw ring is also nearly constant, and thus this variation in press force is for all practical purposes a variation in that component that is available for drawing. This means that the ratio between the force required to depress the draw ring, and the force available to draw the shell, is not constant. While the force used to depress the draw ring might be only a small fraction of the total potential drawing force near the bottom of the stroke, it might well be as great or even greater further up. Thus, our determination of maximum cushion tonnage (which is equal to the force required to depress the draw ring) must be based upon a division of the press force at the point in the press stroke where drawing begins.

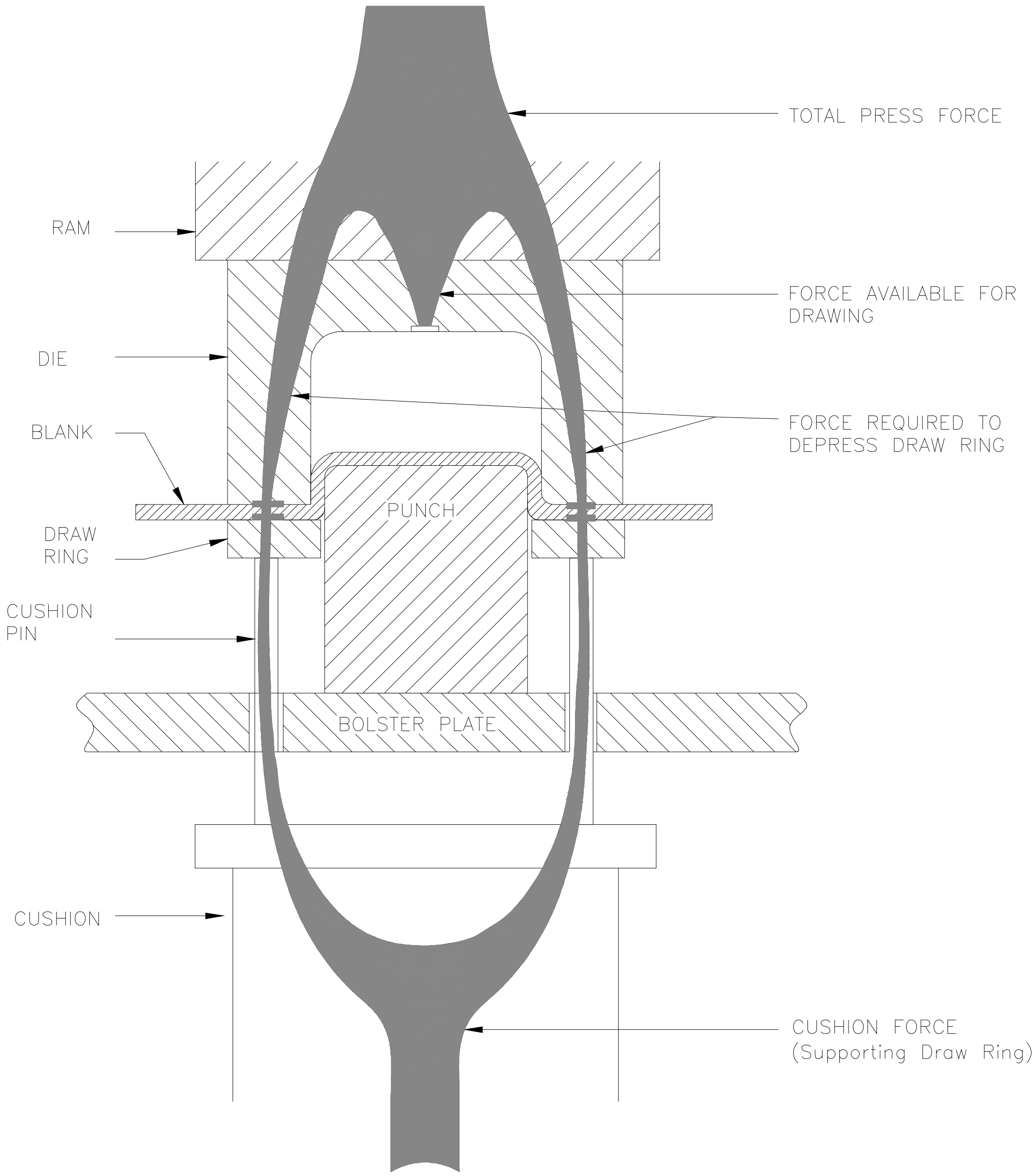


Fig. 1-2-1 Relation between cushion tonnage and press tonnage

As the reader can easily surmise, such determinations can become very complicated, and difficult to make. However, we are relieved of much of this difficulty by two things. First, we are, as mentioned above, concerned only with maximums; this means that calculations need be made for only a relative handful of dimension-material combinations. Second, upon making a goodly number of such calculations for a wide range of situations, we find that the great majority of these yield cushion requirements in the range from one-sixth to one-fourth the rated press tonnage. And these results have been well substantiated by years of experience in practical application. Such a range is not excessively large, since certain flexibility in cushion choice is necessary if we are to establish standard cushion sizes within various standard lines. This provides us with a simple rule of thumb, which states that a cushion should have a tonnage of approximately one-fifth the press, tonnage; and that this can vary from a minimum of one-sixth to a maximum of one-fourth.

This rule of thumb is of course by no means foolproof, and in particular may not always yield completely satisfactory results where a press is to be used to its ultimate capacity. Nevertheless, the reader should never have any hesitation in its use. The many variables and uncertainties involved in press application preclude anything like a perfect match between press and cushion. The only useful suggestion that can be offered for special cases is that it is better to over cushion rather than under cushion. However, we should caution that a cushion tonnage of one-fourth the press tonnage is actually quite large; and the frequent use of large cushion tonnages tends to have detrimental effects on the working parts of the press. Thus tonnages in excess of the maximum of one-fourth the press tonnage should be provided only if there is a very good reason

In considering the relationship between the press stroke and the cushion draw, we must direct some of our attention to the mechanics involved in the removal of shells after the drawing operation has been completed. As noted in Sec. 1-1-3, after drawing has been completed, and the ram and die retreat, the shell is held inside the work area by the punch, which extends up inside of it. Before it can be removed, it is necessary that it be pushed off the punch by the draw ring, an operation known as "stripping". This means of course that there must be space above the punch and below the die to accommodate the shell. This requirement is illustrated in Fig. 1-2-2. Noting the distance through which the die must travel, we see that the minimum press stroke must be approximately twice as long as the height of the shell. Or conversely, that the maximum height of the shell is approximately one-half the press stroke. On the other hand, the distance through which the cushion must retreat is equal exactly to the height of the shell; this is in fact the reason why we call it the cushion "draw". Putting the two together, we see that the maximum draw of the cushion need be no more than one-half the press stroke.

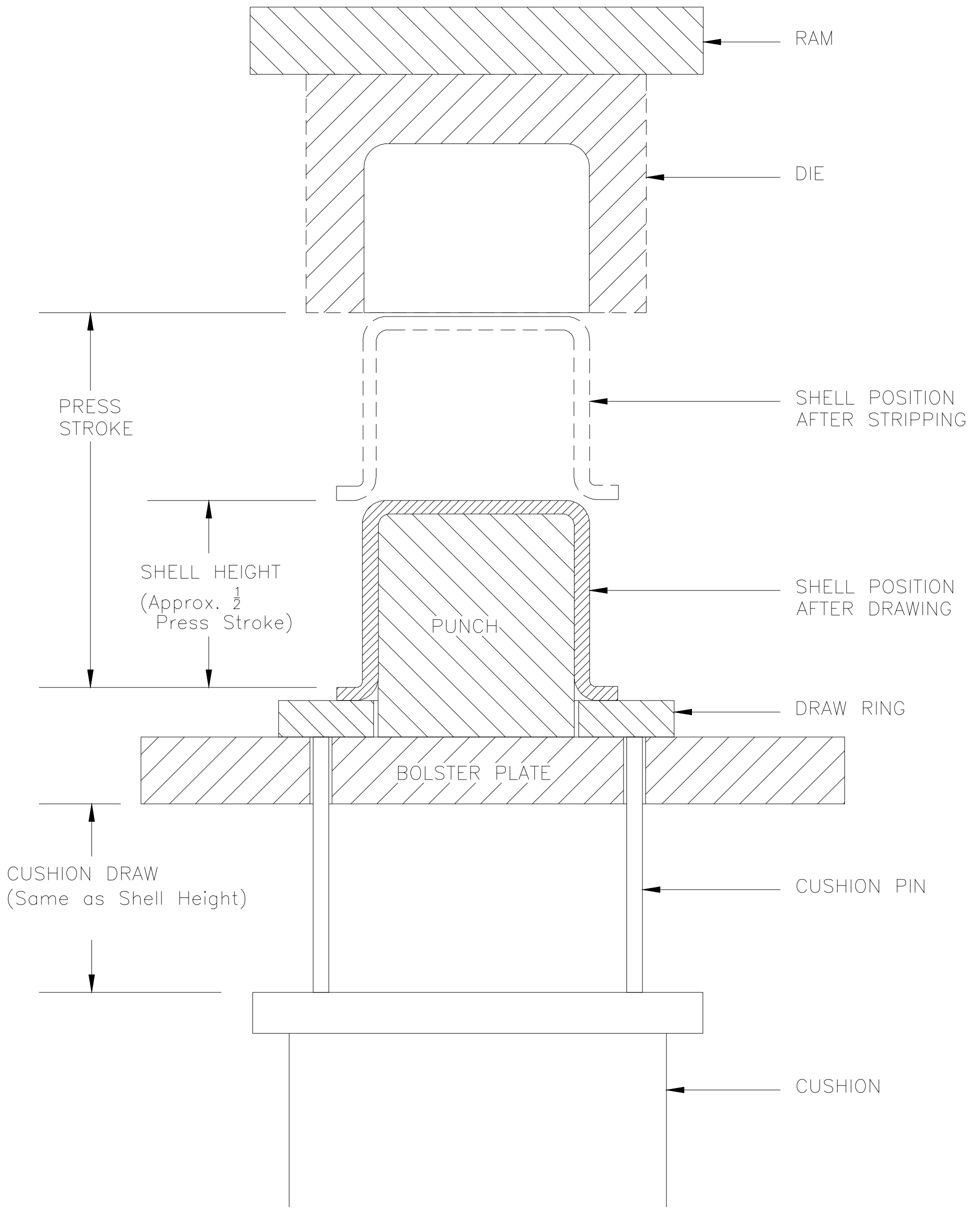


Fig. 1-2-2 Relation between cushion draw and press stroke

This is our second rule of thumb, which we add to the rule for tonnage given above. It should be noted that in practice it is seldom feasible to produce a shell with a height of fully one-half the press stroke; and hence the cushion draw under this rule will perhaps be a little excessive. Nevertheless, it is in keeping with our principle of dealing with maximums, and presents a simple method of calculation. If the customer desires a shorter draw, or if a shorter draw is more in line with installation requirements, it is usually possible to reduce the results of this rule by one quarter, i.e., to make the draw three-eights of the press stroke. However, reductions of more than this should be made only after a careful appraisal of the customer's requirements; and he should be clearly advised of the limitations this will place on the use of the press.

While the dimensions of the top of the bed do not have as important a bearing on cushion design or choice as do the press tonnage and stroke, they are worth considering. If the reader will refer to the illustrations in Sec. 1-1-3, in particular Figs. 1-1-10, 1-1-11, and 1-1-12, he will note that the cushion pins bear on the top of the cushion (or on the pin pad) at some distance from its center. Obviously, the diameter of the pin pad must be sufficiently large that all such cushion pins will bear against it. Hence, if the dimensions of the top of the press bed are large, it is possible for the presswork done by the press to be correspondingly large, and thus require a cushion-pin arrangement over too great an area to be accommodated by a small diameter cushion. Of course in theory it is only the diameter of the pin pad itself that is important; but in practice it is not wise to use a pin pad that overlaps the top plate of the cushion cylinder by more than a few inches. Hence the diameter of the cushion can be an important consideration. Since this particular problem is perhaps more of an installation design problem than a cushion design problem, we will not carry our discussion any further at this time; however, the matter should be kept in mind, particularly when analyzing the relative advantages of two or more different-sized cushions.

Leaving for the time being the consideration of press capacities as regards press-work, the next thing to consider is the space available for cushion mounting. The importance of this to cushion design hinges on the fact that the cushion is to be mounted inside of something, namely the press bed; and this means that all of the outside dimensions of the cushion are important, rather than just a few. Since press bed cavities vary widely in size and shape, it is not possible to hit upon one set of cushion dimensions that will be satisfactory for all cases. However, whatever the specific conclusions we may draw for any particular case, the most obvious general conclusion is that the cushion should be as small and compact as practical. And the reader will note that all successful cushion designs conform to this, within the limits imposed by operational requirements, particularly those mentioned above. It is certainly obvious that however well a cushion meets any other requirements, it is of no use if it cannot be mounted in the press!!

The mechanical principles referred to above pertain primarily to the following:

1. The strength of the walls of the cylinder and piston.
2. The strength of plates and flanges.
3. The overlap of the cylinder and the piston.

These are only a few of the many that we will discuss during the course of this chapter but they have particular significance in the present section in that they are closely related to the conditions under which the cushion will have to operate.

First, and perhaps foremost, the walls of the cylinder and piston must be able to withstand the greatest pressure that could possibly develop in the internal air system; otherwise the cushion might explode, doing damage to the press mechanism, and possibly injuring operating personnel. In order to be sure that the walls are strong enough; a safety factor of at least six is introduced, with a safety factor of ten being more common. This means that for most cushions the walls are designed to be strong enough to withstand ten times the greatest pressure that could possibly develop. Since this greatest possible pressure is determined with respect to the worst possible operating situation, the safety factor is usually a bit more than ten in actual practice.

In addition to the strength necessary for cylinder and piston walls, it is also necessary that all plates, flanges, stop and guide rods, etc. be strong enough to withstand a variety of forces. In particular, we may recognize forces of three types:

1. The direct force from the ram, which collapses the cushion via the cushion pressure pins.
2. Twisting and bending forces exerted on plates and flanges due to deflection and

distortion of the press bed during the maximum-force portions of the work cycle.

3. Shock and vibration resulting from the movement of the press mechanism, and the impact between tooling and work pieces; which is transmitted to the cushion by the cushion pins, and by the suspension rods and mounting plate.

To be assured of sufficient strength, plates and flanges are also designed with a safety factor of at least six, and preferably ten.

As to the matter of overlap of cylinder and piston, this relates directly to the ability, of the cushion to withstand an unbalanced load. Any unbalanced force acting off the centerline of the cushion will tend to cock the cushion cylinder; this is illustrated in an exaggerated form in Fig. 1-2-3. While in general we strongly discourage all such loading of the cushion, since any off-center loading tends to cock the cylinder and cause damage to cushion parts, all cushions will withstand a little of this, particularly if it occurs only once in a while; and the greater the overlap the greater the off-center loading the cushion can withstand. Thus we must allow some overlap in all cases, and must increase this, often by a considerable amount, when any other than very minor off-center loading is anticipated. This subject will be further discussed in the next section.

As a final point, we note that presses often operate continuously for long periods of time, and their schedules of operation are generally closely integrated with more extensive manufacturing activities, which may involve considerable numbers of other presses and related machines, and hundreds of operating personnel. A die cushion that requires excessive service or frequent maintenance and repair will not in general be a satisfactory piece of equipment.

Summing up, we may list the following requirements that any die cushion would be expected to meet:

1. It must be able to develop a force equal to approximately one-fifth the rated tonnage of the press it is installed on.

2. It must have a draw one-half as long as the press stroke.

3. It must be as compact a possible.

4. It must be strong and rugged.

5. It must be dependable and easy to service.

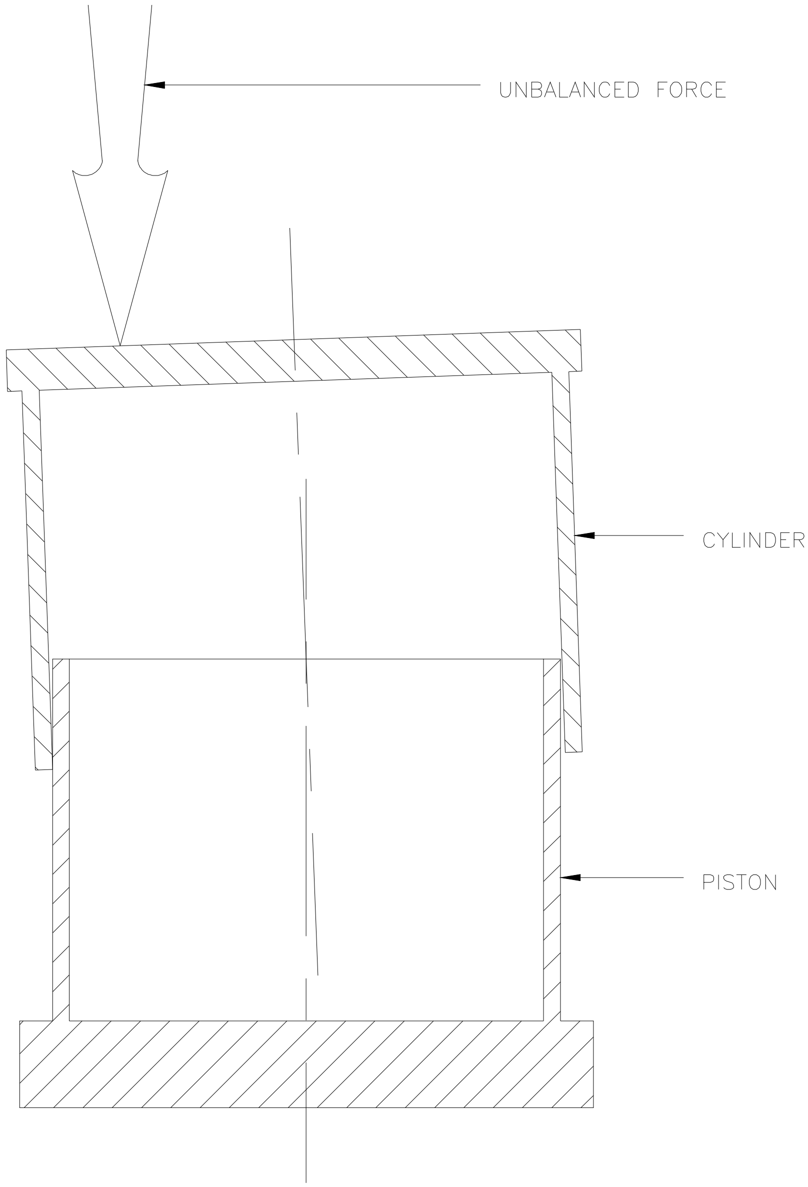


Fig. 1-2-3 Cocking of the die cushion cylinder

1-2-2. DIE CUSHION PARTS AND DIMENSIONS

As the reader will have noted, the criteria given in the preceding section for use in the design of a die cushion may also be used in the selection of a die cushion. This is a natural consequence of the attention given to press requirements in the designing of standard cushion lines, and in the determination of standard sizes and capacities. In this second section, we are going to take a close look at one of these standard lines, examining the component parts and basic dimensions of a typical cushion model. This will be a Model "C" cushion, and the comments to be made will apply to all sizes.

Referring to Fig. 1-2-4, we should familiarize ourselves with the following basic die cushion parts:

1. **Cylinder** — On Model "C" cushions, this part is a weldment; i.e., it consists of two pieces welded together. One of these pieces is a flat steel plate and is usually referred to as the cylinder top plate (or cushion top plate). The other is a cylindrical piece formed by rolling a flat steel plate into a cylindrical shape and welding the joint; it is usually referred to as a cylinder ring or cylinder ring weldment. These two parts are welded together so that the plate encloses one end of the ring.
2. **Piston** — For Model "C" cushions, this part is made of cast iron, and is somewhat the same shape as the cylinder, except that it has a smaller diameter. On some cushion lines this part is also a weldment, being fabricated in the same way as the cylinder is. In assembly, the open end of the piston is inserted into the open end of the cylinder, so that the two parts together enclose a large cylindrical cavity.
3. **Quad Ring Packing** — In order that they may be manufactured for a reasonable cost, both the cylinder and piston are manufactured to, “commercial" tolerances; which means that the outside diameter of the piston is ten or more thousandths of an inch smaller than the inside diameter of the cylinder. This means that when the two are fitted together there will be a gap between them; and although this gap is quite small, it is sufficiently large to permit the escape of the pressurized air inside the cushion, unless some means is provided to prevent this.

The closing of this gap is the function of the packing. While older versions of Dayton die cushions used cup packings for this purpose, all newer models use quad-ring packing. This is a type of O-ring packing, with a special cross section that prevents twisting of the ring during operation. For more information on the functioning of this type of packing, the reader should consult standard packing handbooks, or manufacturer's catalogs.

4. **Lubrication System** — The lubrication system on the Model "C" is basically a means of getting grease into the gap between the piston and cylinder walls. This grease serves two purposes: it reduces friction between the packing and the cylinder wall; and between the piston wall and cylinder wall. With regard to lubricating the packing, this is necessary not only to reduce wear, but also to prevent turning and twisting. As regards contact between the piston and cylinder, this occurs in spite of the gap between them because the packing will not support much side thrust, and hence does not to any degree act to hold them apart. Lubrication throughout the entire area of overlap is thus necessary. In lubricating a Model "C" cushion, grease (usually a soft compound prepared especially for die cushions) is forced into grease fittings on the lower flange of the piston, and is conducted through copper tubing into a grease groove located just below the packing groove on the piston. High-pressure hydraulic hose is used in place of copper tubing on some of the other cushion models.

5. **Stop and Guide Rods** — These are to be found only on cushions that are to be press bed mounted, and are intended to serve two purposes: first, they keep the cushion from flying apart when it is filled with pressurized air; and second, they prevent the cylinder from turning. This latter function varies in importance with the application of the cushion, but in as much as neither the top plate of the cylinder, nor the pin pad is circular (with rare exceptions) it is usually a necessary one. Should the pin pad for instance be depressed below the top of the bed, and subsequently turn a few degrees in one direction or the other, it is quite likely that it would not be able to move back through the bed opening as the cushion returns. Correcting this could on some presses involve a complete dismantling of the installation; and of course it could conceivably happen with every stroke of the

press. In assembling the cushion, one end of the stop and guide rod is passed through a hole in the cylinder flange and screwed into the lower surface of the pin pad; it is locked in with a jam nut against the cylinder flange. The other end extends down through a hole in the piston flange (this hole extends also through any mounting plates or structures that might be used), and is prevented by hex nuts from passing back through it. As the cushion is collapsed the lower end of the rod is pushed toward the floor. When the cushion recovers, the nuts limit the upward movement of the cylinder.

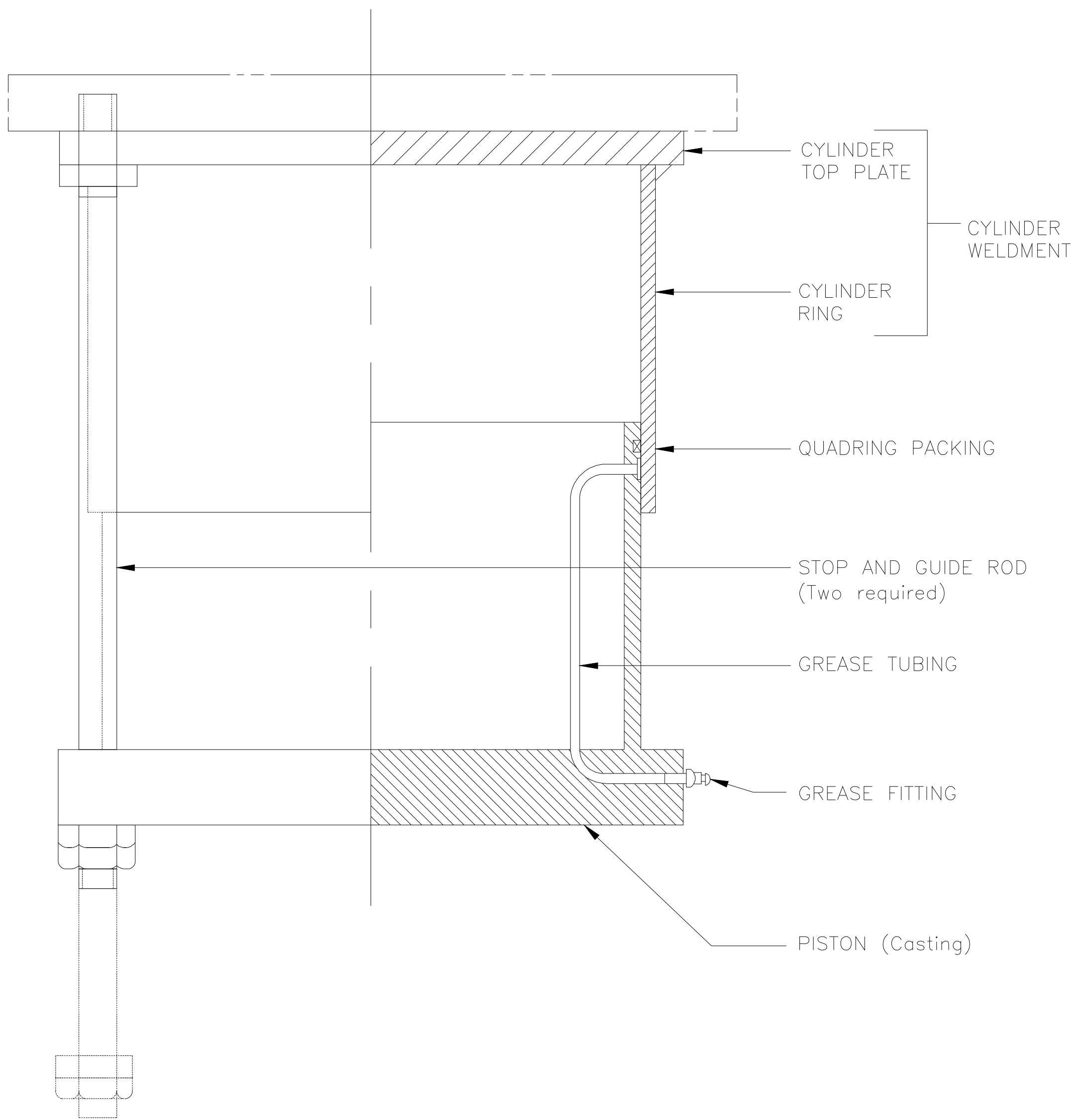


Fig. 1-2-4 Standard parts of model “C” die cushion

While there are a few small parts, such as tube fittings, that we have not included in the above list, the parts covered constitute all of the major parts of a Model "C" die cushion. Thus it is easy to see that this line of cushions has a simple, straightforward design; no parts have been included that are not absolutely necessary to the basic functioning of the cushions. Other die cushion lines, which we will discuss later, are essentially sophistications of this one, and their designs can best be understood in terms of the addition of special parts that improve or extend the cushion application. We will explore this approach In Section 1-2-5.

Leaving for a while the subject of cushion parts, let us consider now a few of the specific aspects of cushion size, namely certain important cushion dimensions. Referring to Fig. 1-2-5, we should note the following:

1. **Cushion Size** — Usually when we speak of the size of a die cushion we are referring specifically to one particular dimension - the internal diameter (I.D.) of the cushion cylinder. Alternatively, since the difference between them is negligible, we may substitute the outside diameter (O.D.) of the cushion piston. The importance of this particular dimension stems from its relation to the tonnage of the cushion. The tonnage of a cushion (or the force it is able to exert) depends upon the pressure of the air inside, and upon its cross-sectional area. Its cross-sectional area in turn depends upon the diameters we are here discussing) hence, the tonnage of a cushion increases as the I.D. of its cylinder increases, and conversely.
2. **Overall Height** — This is of course one of the more important cushion dimensions to consider when designing a cushion installation. It is the distance from the top of the cylinder top plate to the bottom of the piston.
3. **Cushion Draw** – cushion draw is the maximum distance through which the cushion may be collapsed. Examining the figure, we note that this is essentially the distance from the bottom of the cylinder ring to the top of the piston flange.
4. **Overlap** — This is the distance through which the piston and cylinder rings overlap when the cushion is at the top of the draw (uncollapsed).

5. **Alternative Draw Dimension** — This dimension, given as Dimension "A" on the figure, is the distance from the top of the piston ring to the bottom of the cylinder top plate. Note that this must be at least as great as the cushion drag, or the piston ring will make contact with the cylinder top plate before the cylinder ring makes contact with the piston flange; hence limiting the actual draw to less than that given above.

6. **Flange Thicknesses** – These are, as indicated, the thicknesses of the flanges on the top of the cylinder and the bottom of the piston.

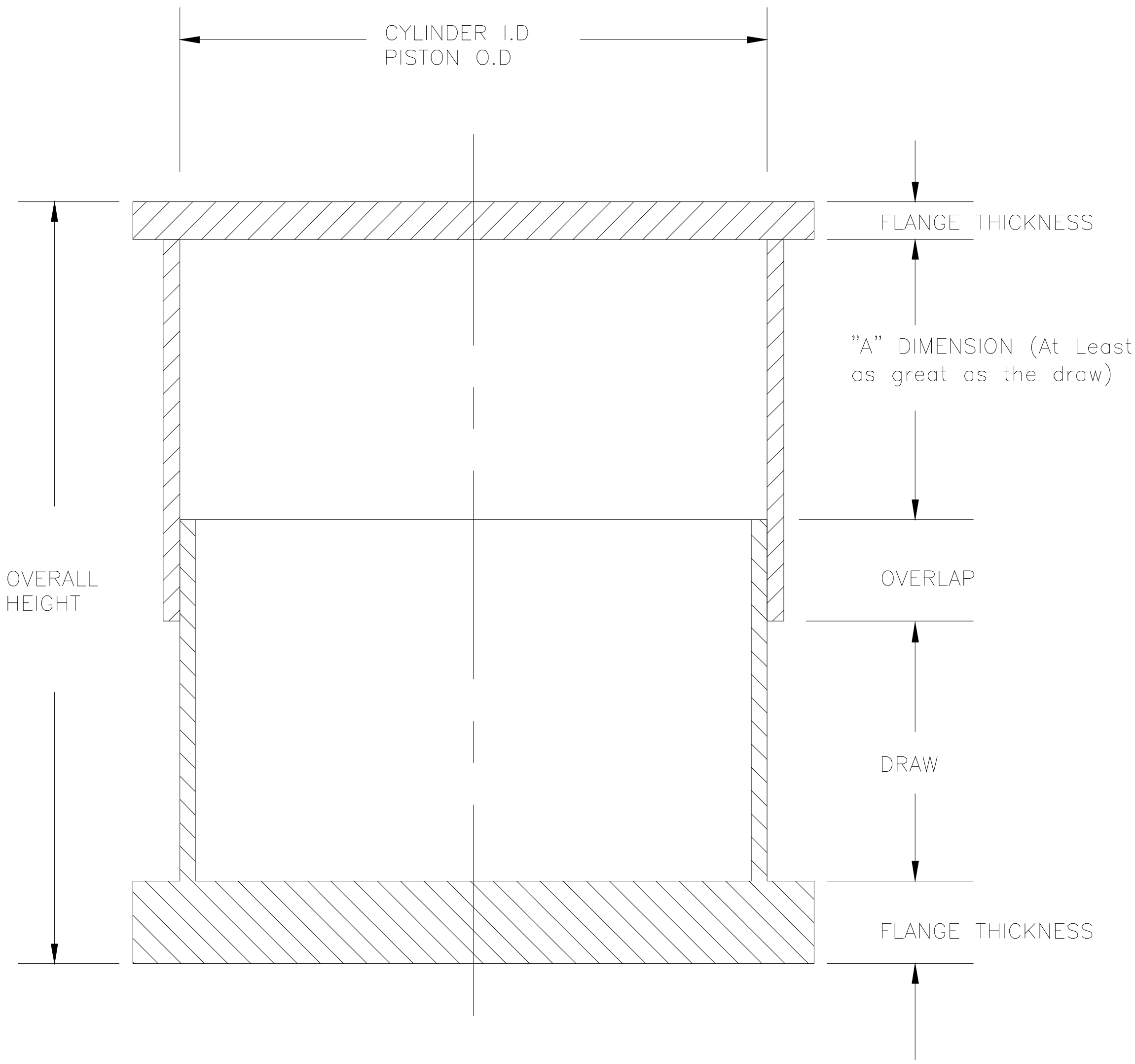


Fig. 1-2-5 Basic dimensions of a model "C" die cushion

Examining the figure closely, it is easy to see that the overall height of the cushion is equal to the sum of five other dimensions; the draw, the overlap, the "A" dimension, and the thicknesses of the two flanges. Since the "A" dimension must be at least as great as the draw, we may conclude that the minimum overall height of a Model "C" cushion is equal to twice the draw, plus the overlap, plus the flanges. This fact has a very important bearing on cushion design. Specifically, if it is necessary to keep the overall height as small as possible, we must minimize each of the component dimensions. Since the draw is determined by the press stroke, there is little we can do here, except to be sure that we do not allow any extra. The flange thicknesses are of course predetermined by strength requirements. Thus, the only dimension over which we have significant control is the overlap. The outcome of this is that most standard cushions incorporate a bare minimum of overlap, and hence most cushions have little resistance to off-center loading, a point that we made previously. It is of course quite easy to remedy this deficiency for any particular cushion, simply by increasing the overlap; but this must obviously increase the overall height, and thus make the cushion unusable for many presses. Since it is usually more feasible to take care in arranging tooling so that off-center loading does not occur, than to accommodate very high cushions in small press beds, the short overlap of the standard cushion designs represents a sound approach. Nevertheless, it does have this important drawback, and the reader will note that in our cushion literature we make a constant effort to warn against the off-center loading of standard cushions.

Another point along similar lines that we should cover concerns the use of collapsed stock-draw cushions. Here we are dealing in part with the economics of stock items. Essentially it is a matter of trying to cut the number of items tied up in inventory, while at the same time maintaining a sufficient capacity to sell cushions "off the shelf" to the majority of customers. The approach is to stock a single "long-draw" cushion in place of a variety of draws, and then to partially collapse the longer draw to meet short-draw requirements. For instance, Model C-12 cushions are stocked in 3" and 6" draws only; if a customer wants a 4" or 5" draw cushion, he is sold a 6" draw cushion that has been partially collapsed so that only 4" or 5" of draw remains. The only disadvantage in this is that in collapsing the 6" draw cushion to the shorter draw we increase the overlap; the reader can verify this with a glance at Fig. 1-2-5. Thus a C-12 with a 4." draw obtained in this manner has an overall height of 2" more than a C-12 with a 4" draw that is built to the standard convention of using only a minimum overlap. For the majority of applications, this difference is of minor significance, and the customer probably benefits by the increased resistance to off-center loading resulting from the longer overlap. However, if height is critical, it might sometimes be necessary to eliminate the extra overlap. This, of course, requires a certain amount of machining, and will raise the price of the cushion; a condition that it might sometimes be difficult to explain to a customer.

Looking at the cushion from the top, what we see is essentially a circle with two squared-off flanges, 180° apart; this is illustrated in Fig. 1-2-6. The flange width is taken as the dimension indicated on the drawing. Because of the two flanges, the top view has a "short" dimension, and a "long" dimension, the short dimension being the same as the outside diameter of the cushion cylinder. This means that the cushion requires more room one way than the other; and this must be taken into account in designing an installation. There are two holes in each flange, making a total of four. For bolster plate mounting, these are used for the bolster mounting rods. For press bed mounting, two of them, 180° apart, are used for the stop and guide rods; and the other two are used for bolts fastening the pin pad on. The piston flanges are similar, with the holes serving much the same functions; except that for press bed mounting the two holes not used for stop and guide rods are used for bolts fastening the cushion to the mounting plate. We will discuss this subject in more detail in the next chapter.

As a final consideration for the present section, we should take particular note of the system we use to identify die cushions. Each standard cushion line is identified by a characteristic letter, such as the Model "C" cushions we are discussing here, or the Model "MC" cushions we will discuss later. In identifying a particular cushion, this letter is usually followed by two numbers, separated from each other and from the letters by dashes. The first of these gives the I.D. of the cushion cylinder, and is regarded at a designation of cushion size; the second gives the draw. Thus, a cushion identified as a "C-12-6" is a cushion from the Model "C" line, with a 12" cylinder I.D., and a 6" draw. When no draw is specified, or the draw is not significant to the discussion, the last number is usually dropped, and we would refer to the cushion simply as a “C-12”.

While this system is simple enough as to cause no appreciable difficulties, a few words of caution are nevertheless in order. The literal designations of standard cushion lines will sometimes be used for cushion designs that are somewhat different from standard, but for which there is no readily available identification. Thus, while "C-16" usually means a standard Model "C" cushion with a 16" cylinder I.D., it is quite possible that on occasion a cushion so designated will have special features. These will usually be noted directly after the literal-numerical identification, such as "C-12-6, with air-exhaust hold down"; however, this is by no means a guarantee against mistaken identity, and it is wise to be on the lookout for variations from standard. Also, as mentioned above, the draw may or may not be arrived at through the collapsing of a longer-draw cushion; the resulting variation in overall height can sometimes be quite important, and it is best not to assume that this dimension will be consistent from cushion to cushion. Finally, for some cushion types, there is actually no real standard design at all, and the literal designation has a much more flexible meaning. Such cushion lines, for example, as the Model "RM" ram-mounted cushions may vary so widely in design that each individual cushion is virtually custom designed.

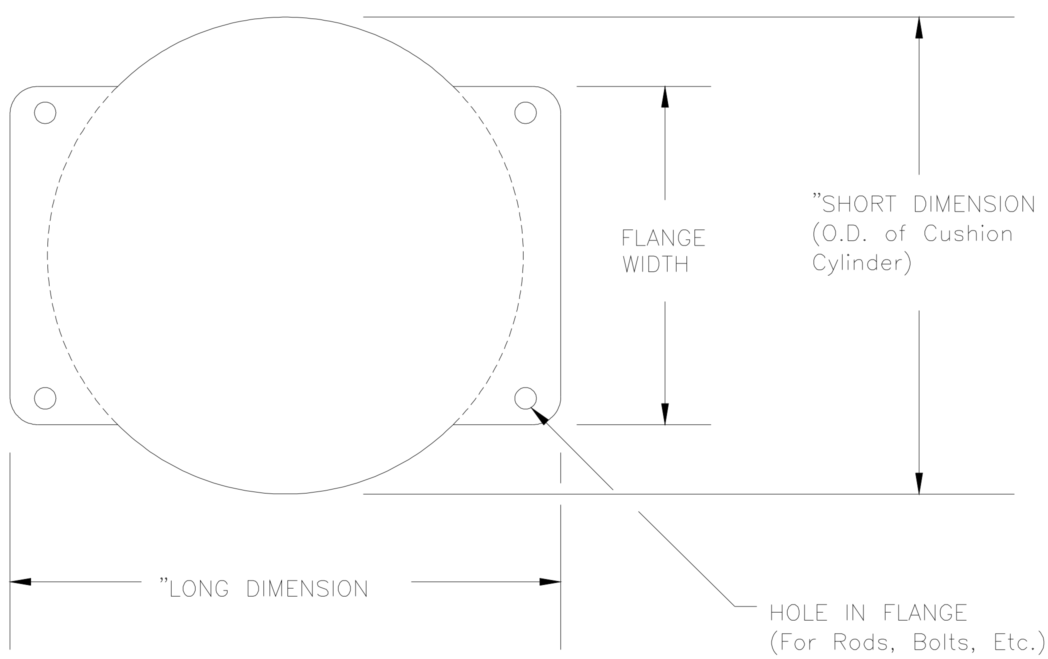


Fig. 1-2-6 Top view of model “C” die cushion

1-2-3. ELEMENTARY THEORY OF PNEUMATIC SYSTEMS

Die cushions are devices whose primary functioning involves the containment, compression, and movement of masses of air; and questions regarding pressures, volumes, and air flow conditions are both numerous and varied. In this, and the following, section of the present chapter, we will discuss some of the more important principles involved, explain their application to the particular problems that arise in the design and use of die cushions, and provide some simplified methods for making necessary calculations and for determining practical configurations. Much of the material to be presented in these two sections will of necessity be somewhat beyond the immediate needs of the general reader; and for this reason he may skip over them, if he so desires, and go on to Section 1-2-5. However, the subjects to be covered here have a certain value in themselves, and a familiarity with the things to be explained here will act to simplify and clarify later discussions; hence, the reader is urged to go over this material if at all practical.

Fundamentally, the role played by air in the mechanisms we are to be concerned with involves its use for transmitting force and for storing energy. In order, however, to understand the practical significance of these phenomena, there are a number of basic principles that we must first become acquainted with. To the extent that we must study them here, they are not at all difficult; nevertheless, a thorough understanding of them is essential to an understanding of the ins and outs of pneumatic systems, and we will consequently go over them very carefully.

To begin with, the air of which we speak is a mixture of several chemical substances in a gaseous state. The most important of these are: Oxygen, which constitutes about 20% of the mixture; and Nitrogen, which constitutes a little less than 80%. Other substances, such as Carbon Dioxide and water vapor, are present in small amounts, but their exact amounts or effects do not concern us here. All of these substances are present in the form of aggregates of very small particles called "Molecules". By the term "gaseous state" we mean that these constituent molecules are not bound tightly together, such as are the molecules in liquids and solids; and consequently, they are relatively free to move in all directions. Figure 1-2-7 illustrates the somewhat random motion of the molecules in a quantity of air. These move in straight lines, and are not under any restraint except that resulting from a collision with other molecules. These collisions are assumed to be perfectly elastic, which means that when two molecules collide they bounce apart much the same, as would two rubber balls. If a quantity of air is not confined, the motion of its constituent molecules will cause it to expand, or disperse, as is indicated in Fig. 1-2-8. This tendency of air to expand is an important phenomenon, as we will see below.

A good proportion of the mechanics of pneumatic devices and systems deals with the relationships between the tendency of a quantity of air to expand and the various restraints that are put on that expansion. This leads to a number of basic concepts dealing with the idea of the confinement or containment of air. Perhaps the first of the restraints that we should discuss is that resulting from the force of gravity. In spite of their basic tendency to move in straight lines, very few molecules of the air (or atmosphere) that surrounds the Earth ever attain sufficient velocity to overcome gravity and escape entirely into space. Sooner or later, virtually all molecules that move outward from the Earth are slowed to a standstill, and then fall back. In this respect they behave much the same as any other objects propelled into space, such as a stone thrown into the air. Now, while the molecules on the outer fringes of the Earth's atmosphere are subject to no restraint other than the force of gravity, as they fall back toward the Earth they collide with other molecules j so that air molecules below this outer fringe are subject to restraint not only from gravitation, but also from collision with fringe molecules falling back to Earth. Thus the fringe molecules act as a sort of porous "container" for molecules closer in. This particular effect is cumulative as we move closer to the Earth's surface; so that air becomes increasingly denser, and the pressure it exerts becomes increasingly greater. The overall result of this is indicated in Fig. 1-2-9. We note especially the difference in air pressure; at the outer fringes of the atmosphere this is zero pounds per square inch; and at sea level it has risen all the way to fourteen and seven-tenths pounds per square inch, which we usually round off to 15 psi. This pressure is known as "atmospheric pressure"; and as we will see in the next section, it has an important role to play in various calculations dealing with the relationship between pressure and volume.

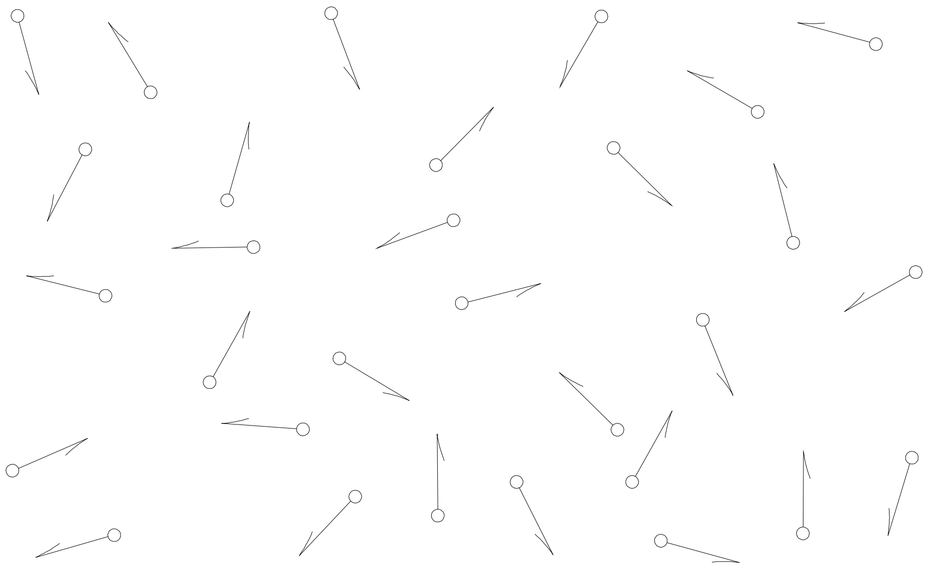


Fig. 1-2-7 Random motion of air molecules

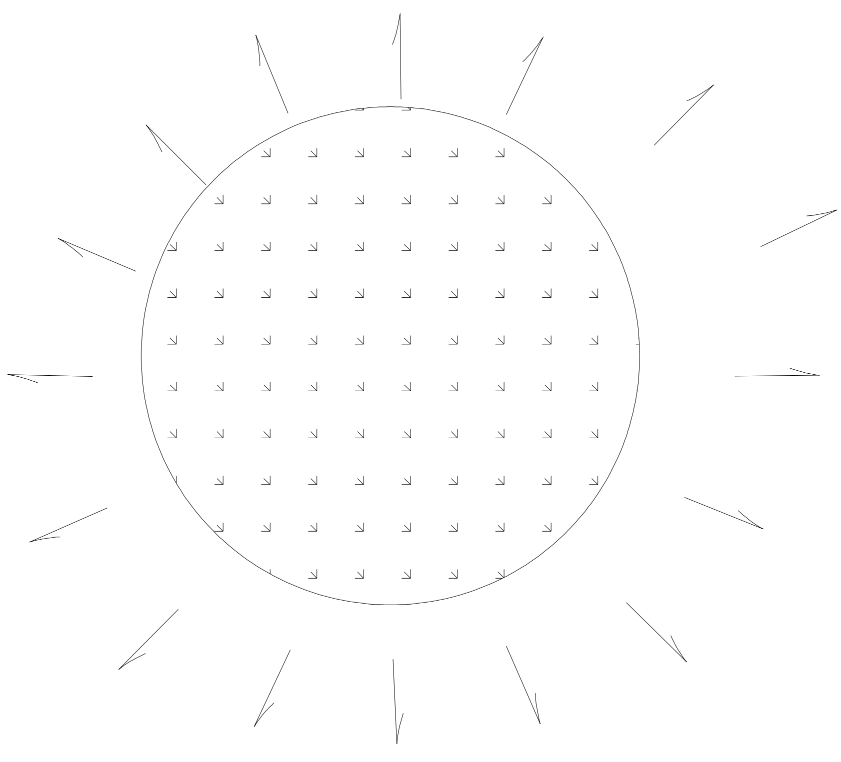


Fig. 1-2-8 Dispersion of an uniformed quantity of air

We should at this point pause for a moment to consider the meanings of two rather important terms; these are force and pressure. In Physics, a force is usually defined simply as a push or a pull; and since we are dealing primarily with forces deriving from confined air, the notion of a push will be the most useful here. The magnitudes of these pushes are measured by comparing them to weights, by which we mean the effects of gravity. Thus, if we say that a given object weighs one pound, we mean that because of the effect of gravity on it .it exerts a push against a supporting surface, such as a table; and the magnitude of that push is equal to a standard push (the weight of a standard object), which we signify by the term pound. In engineering, the comparison of forces (pushes) with gravitational forces is the standard convention, and all magnitudes are therefore given in ounces and pounds (or grams and kilograms in the metric system).

In particular, the forces exerted by air derive directly from the collisions of air molecules with other objects. Thus, as a molecule in a quantity of air collides with the wall of a container it exerts a push against it. The total force exerted by a confined quantity of air is simply the sum of the pushes exerted by all of its molecules. The magnitude of this total force will obviously depend upon the number of molecules involved, and the speed at which they are traveling. The first of these involves a quantity called density, which we will discuss later; the second involves a quantity called temperature, which is in a sense a measurement of average molecular speed. In simple terms, the denser the gas, and the higher its temperature, the greater will be the total force it exerts.

While the concept of total force is a very useful and important one, there is another concept that is just as useful; this is the concept of the amount of force exerted against a specified area of a container wall. This area is usually taken as a unit area, such as one square inch or one square centimeter; and the quantity of force involved is referred to as pressure. Thus, if the force exerted by the air in a container against each square inch of the container wall is fifty pounds, the air is said to exert a pressure of fifty pounds per square inch, or 50 psi. In comparing these two concepts, we note that force involves only magnitude, while pressure involves both magnitude and area. We will treat this subject a little further in the next section, where we will deal with the rules and methods of making calculations.

Conventionally, when we think of restraint on a quantity of air we think of a container having solid walls. These walls may be rigid, or they may be flexible and elastic. Two such containers are illustrated in Fig. 1-2-10, a balloon having elastic walls on the left, and a steel box having rigid walls on the right. Each has an opening by means of which air may be allowed to enter or exit. Considering the balloon first, let us examine what happens when air is forced into it. Immediately this air spreads out to fill whatever cavity exists initially, and begins to exert pressure against the walls; these, being of an elastic substance (rubber) will begin to stretch. This process continues as additional air is forced in, and may be carried to the extreme of bursting the balloon. Let us consider, however, what happens if the supply of air is cut off before this happens, and the opening is closed. The expansion (stretching) of the container walls will continue only momentarily, and then it will cease. What has happened is that the tension in the balloon wall, created by stretching the rubber, has counterbalanced the pressure of the air inside. Obviously, if this were not so the expansion would continue.

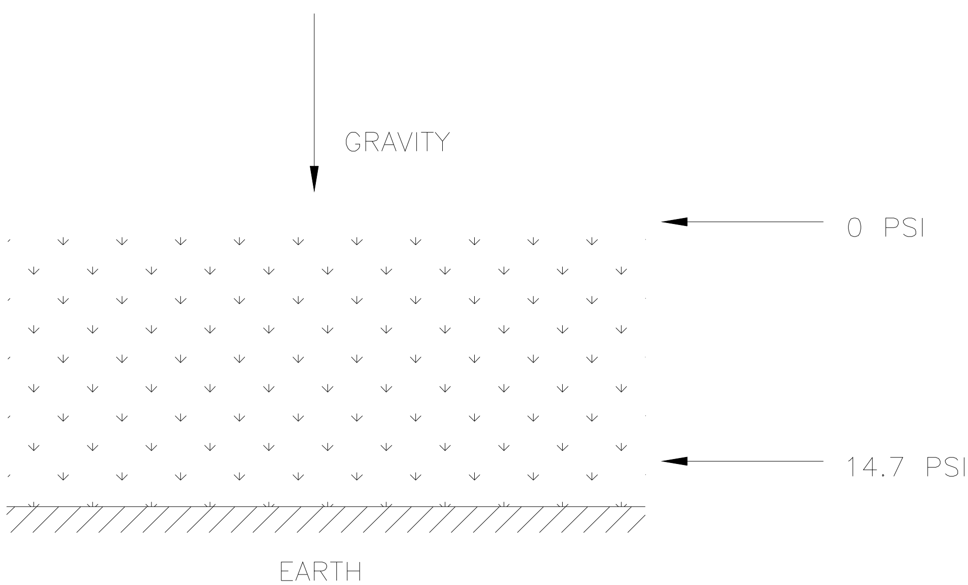


Fig. 1-2-9 Atmospheric pressure

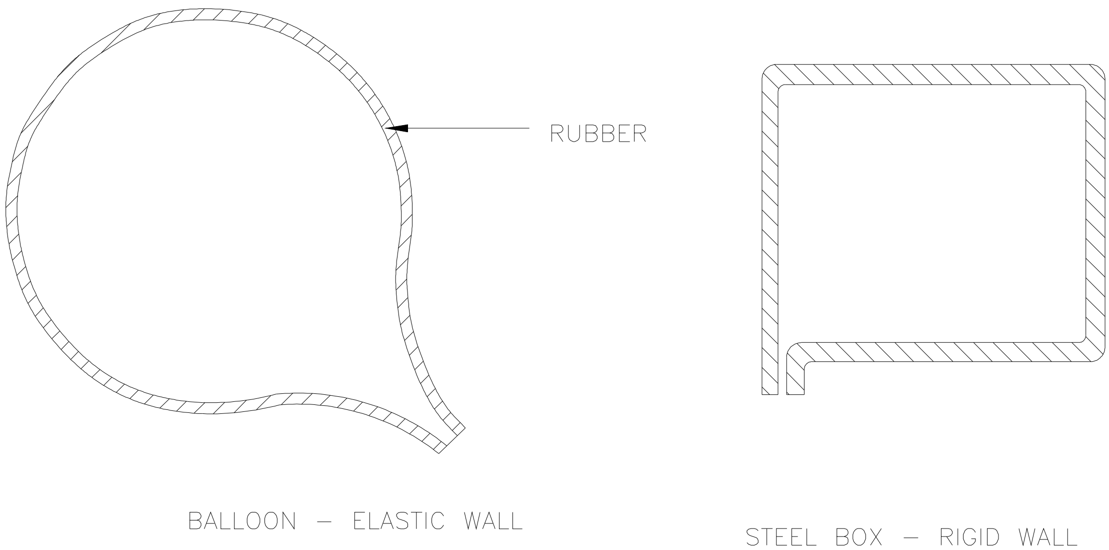


Fig. 1-2-10 Solid wall containers

Surprisingly, much the same thing happens when we force air into the steel box. The important difference is that in this case the stretching of the container walls occurs on such a small scale that it is virtually unnoticeable. In fact, it could be measured only in millionths of an inch. This is of course due to the difference between rubber, which stretches easily, and steel, which does not. Nevertheless, it is important that we recognize that a certain degree of stretching does occur. This stretching creates a tension in the container walls that counterbalances the pressure of the air. Hence, we say that not only does the air push against the walls, but the walls also push against the air. This will result in an equilibrium condition, just as it does in the case of the balloon.

Generally, we are concerned with two classes of containers: variable-volume, and non-variable-volume. Of course, as we noted above, the volumes of all containers vary at least a little when air is forced into them; however, there is a vast difference between the multifold increase in the volume of the balloon and the infinitesimal increase in the volume of the steel box, and we tend to think of the volume of the box as non-variable. While the balloon is obviously a variable-volume container, we find that for the most part this type of container has little direct industrial application. Where we intend to utilize variability of volume in the design of machinery, we find that it is necessary to use a different type of container, in which variability of volume is a function of the movement of separate parts, one of which is called a cylinder, and the other a piston. Two versions of this type of container are illustrated in Fig. 1-2-11. Henceforth, when we speak of a variable-volume container it will be this type of container that we mean.

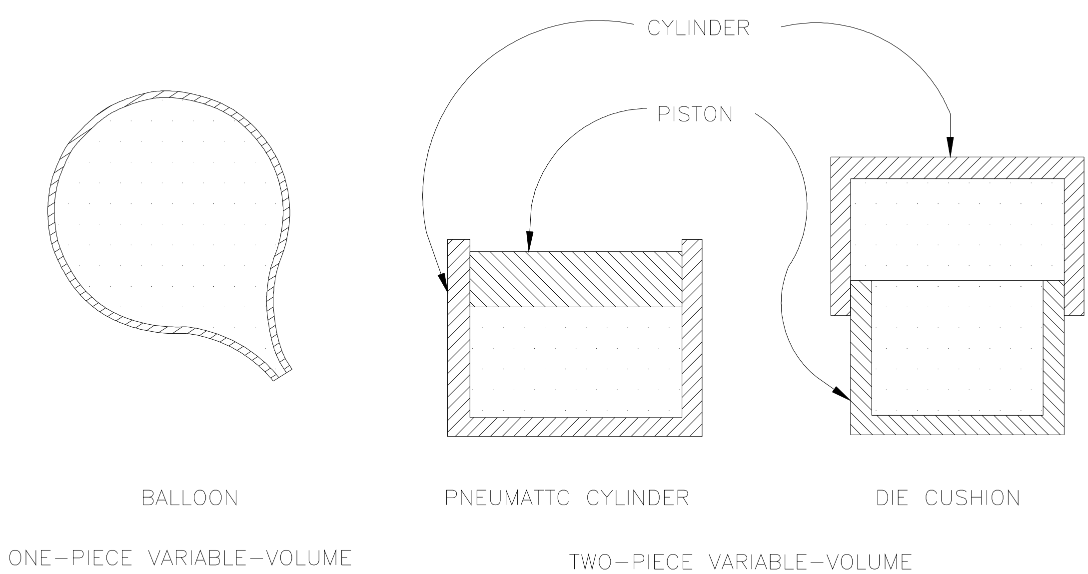


Fig. 1-2-11 Variable volume containers

The variability of the volume of this type of container is an important feature for two basic reasons: one - this variation involves the movement of one of the container parts (i.e., the piston or the cylinder), which may be employed in conjunction with the movement of parts of machinery; and two - it corresponds to an important fundamental property of gaseous substances (air in particular), namely, that they may be expanded and compressed. To be sure we understand what this latter reason entails, let us refer to Fig. 1-2-12; this figure illustrates a typical variable-volume container in which the piston is shown at three different positions. In each case the container holds the same quantity of air. If we take the first case as the normal condition, then we note that as the piston is moved up, the air in the container expands until it fills the new volume. If the piston is moved down, the air is forced into a smaller volume, or is compressed. Certain important things happen when air is expanded or compressed, and we will examine these below; however, we must first examine a special condition known as equilibrium, to which we made brief reference above.

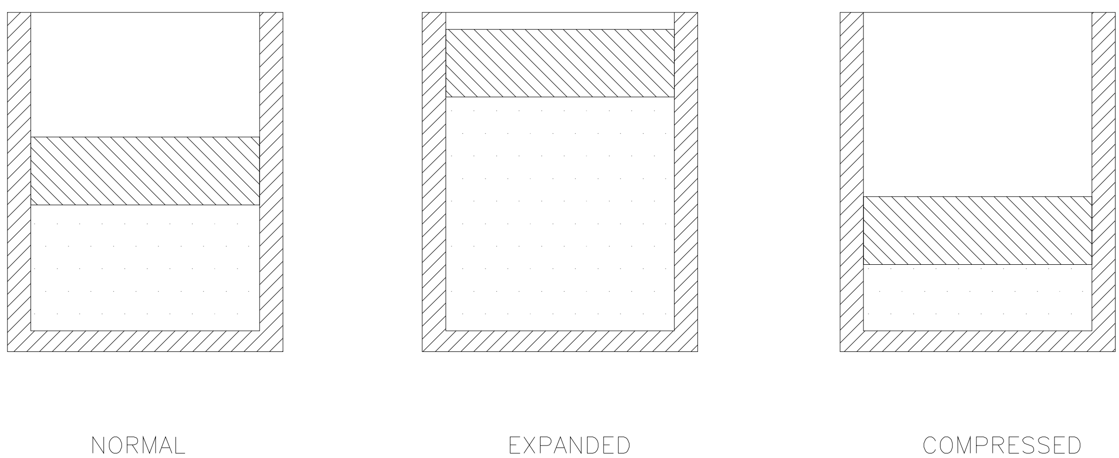


Fig. 1-2-12 Expansibility and compressibility of air

Equilibrium, in its simplest sense, refers to a balance of forces. Thus, if two forces are in direct opposition, and they are of equal magnitude, they will cancel each other out, and we say that they are in equilibrium. By this we mean that they continue to exist, but that they do not cause movement. As we noted above, when air is pushed into a container it exerts force against the container walls; this is illustrated in Fig. 1-2-13a. If the container is of the non-variable-volume type, this force will be counterbalanced by the resulting tension in the walls. However, if it is of the variable-volume type, some of the force exerted by the air will tend to change the volume, since it will be exerted against moveable members. This situation is illustrated in Fig. 1-2-13b, where the forces exerted by the air are indicated by the lower-case letters a, b, c, and d; and the forces acting to counterbalance these are indicated by the upper-case letters A, B, C, and D. If we examine the figure carefully, we will see that only the forces c and d (which represent all the forces exerted by the air in a sideways direction) can be counterbalanced by tension in the container walls, because the surfaces against which they act are part of the continuous surface of the cylinder, and are hence firmly connected together. Since the piston is not connected to the cylinder, forces a and b can be counterbalanced only by forces external to the container. If such forces do not exist, the cylinder and piston will fly apart much the same way as a rifle and a bullet.

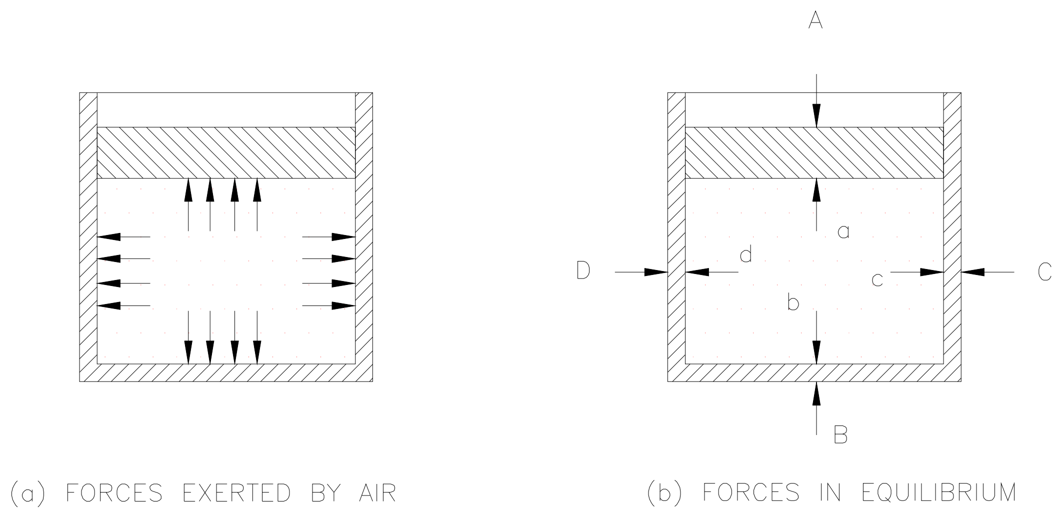


Fig. 1-2-13 Equilibrium condition

In practice, external forces A and B are supplied by the machinery or other equipment of which the container is a component part. In this arrangement, one of the moveable parts is firmly attached to the machinery, and the other part is limited in its movement either by the position of other components of the machinery, or by devices provided especially for this purpose. Ultimately, whatever the exact mechanical arrangement involved, the machinery acts as an external connection between the piston and cylinder, and the forces exerted by the air against these are counterbalanced by tensions in various machinery parts. Either of the two moveable parts of the container may be held stationary, as is indicated in

Fig. 1-2-14. The right-hand alternative, where the piston is held stationary, is typical of the die cushion installation arrangements discussed in the previous chapter. With the stationary member of the container firmly attached to the machinery, the question of equilibrium, that is, of the counterbalancing of all the forces exerted by the confined air, is reduced (or simplified) to the problem of counterbalancing only the force exerted against the moveable member. In particular, the moveable member will remain stationary if the internal and external forces are equal and it will move if they are not. Hence, the use of the container revolves around processes that destroy or restore the equilibrium condition. While this is a little abstract a view for the present discussion, it is nevertheless a very useful one.

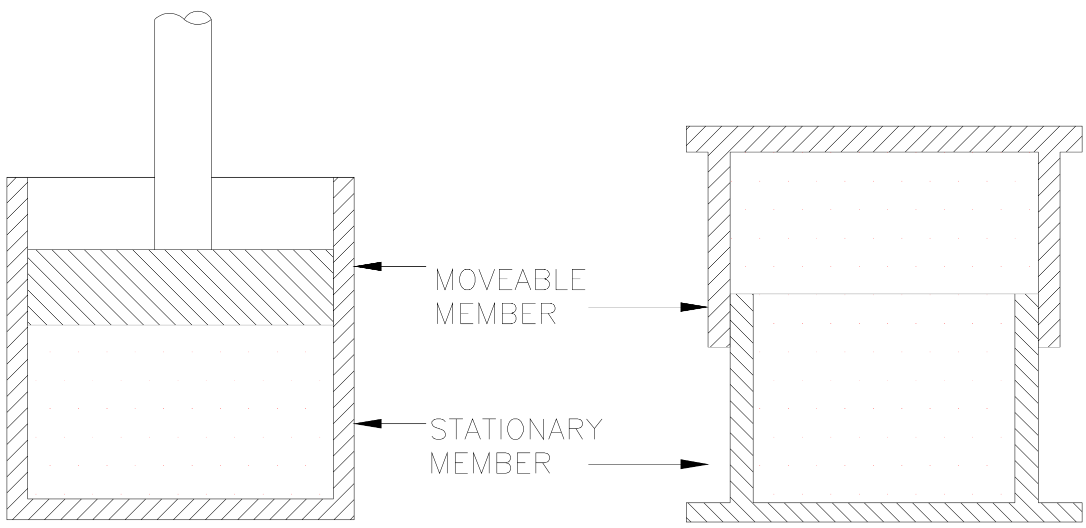


Fig. 1-2-14 Containers as machinery components

Returning now to our discussion of expansion and compression, let us consider what happens when the quantity of air in a container is kept constant (no air is added or removed), but the volume of the container is changed. As we indicated above, the force exerted by air against its container is due to the collisions of its molecules with the container walls; and the magnitude of the force depends upon the number of molecules, and the speed at which they are traveling. If we leave for a while the concept of total force, and consider only the concept of pressure, we will see that as the volume of container is increased, and the air in it expands, the number of molecules colliding with, a given portion of the container wall will decrease. Hence, the pressure decreases. Conversely, if the volume of the container is decreased, the number of molecules colliding with a given portion of the wall will increase; and the pressure increases. Thus we see that these two quantities, volume and pressure, vary in an inverse manner, one increasing while the other decreases, and vice versa. All of this provided, of course, that the quantity of air (total number of molecules) remains the same.

At this point we should take a moment to comment briefly on the distribution of the internal forces exerted by the confined air. Essentially, this is the question of the uniformity of the air pressure, and hence pertains to the distribution of the air molecules, both in quantity and in speed. Since space does not permit us to engage in lengthy discussions, we will take time to consider only the common-sense aspects of this. First of all, let us consider two pertinent facts: one - at room temperature, and at atmospheric pressure, a single cubic inch of air contains approximately 800,000,000,000,000,000,000 molecules; and two - the average speed of these molecules is approximately 1800 feet per second. Because of these, two important consequences are inevitable: first, because of the extremely great number and extremely small size of air molecules, all of the effects that we will be concerned with will involve vast aggregates - the actions of individual molecules can thus be evaluated only in terms of averages, and all the laws and assertions that we can make will have only statistical meanings; second, because of their relatively great speed, the molecules in an air mass will mix extremely rapidly, and will not therefore support phenomena that require appreciable permanence of location at any point, or in any region of the container

Applying these ideas to the question of pressure uniformity, we arrive at the following conclusions, which we state without further arguments:

1. If there is no change in the volume of the container, or in the quantity of air, the distribution of the molecules, and hence the air pressure, will be uniform throughout the container, and particularly against all portions of the container walls.
2. If there is a slow change in either the volume of the container, or the quantity of air, the corresponding change in pressure will also be uniform throughout the container.

3. If there is a rapid change in either the volume of the container, or the quantity of air, this change will create an unequal distribution of molecules, and hence a condition of non-uniform pressure; however, uniform pressure will be reestablished very quickly if this change becomes slow, or ceases altogether.

The determination of what is slow change and what is rapid change, and of how much practical uniformity differs from theoretical uniformity, requires that we consider the actual application of the container, and will thus vary from one case to another. Hence, we will not attempt to present any criteria at this time. We can only caution that this can sometimes be quite important in what are often unexpected ways.

Before we investigate the mathematical relationships involved, let us consider in very simple terms the relationships between the basic phenomena of equilibrium, expansion and compression, and uniformity of pressure. Referring to Fig. 1-2-15, we may note the following:

1. In Fig. 1-2-15a, we have represented a variable-volume container for which the internal and external forces acting against the piston are equal; i.e., they are in equilibrium. If the external force is not altered, and nothing is done to change either the volume of the container, or the quantity of air, the piston will remain stationary.
2. In b, the external force has been increased by a fixed amount. Since this made it greater than the internal force, the piston was caused to move down. However, as the piston moved down, it compressed the air in the container, gradually increasing the pressure, and thus gradually increasing the internal force on the piston. Eventually a point was reached where the internal and external forces were again equal, and the movement of the piston stopped.
3. In c, the external force was decreased. Since this made it smaller than the internal force, the piston was caused to move upward. However, as the piston moved upward the volume of the container increased, the air expanded, the pressure of the air decreased, and the internal force against the piston decreased. Again, a point was eventually reached at which the external and internal forces were equal, and the movement of the piston stopped.

4. In d, we used a different procedure to destroy the equilibrium. Here we added air to the container, thereby increasing the pressure, and hence increasing the internal force on the piston. This caused the piston to move upward until equilibrium was again restored.
5. In e, we removed air from the container, thereby decreasing the internal pressure, and hence decreasing the internal force on the piston. This caused the piston to move downward until equilibrium was again restored.

In all of these cases, one important phenomenon stands out: whenever we destroy the equilibrium, whether we change the external force or the internal force, the piston, moves In such a direction as to restore the equilibrium (i.e., to equalize the opposing forces).

In attempting to arrive at a mathematical statement relating the quantity of air in a container and the force it exerts against the walls, we are led first to consider the notion of density. In simple terms, density is the amount of a substance in a particular volume, usually a unit volume, such as one cubic inch. We note how this concept relates to the notion of pressure, which deals with force on a unit area. We may put these two together in the following manner: First, let us note that the number of air molecules in a given cubic inch of space, and the number that collide with or pass through one of its inch-square surfaces will vary in direct proportion; i.e., if we double the number of molecules within the volume, we will double the number at any of its surfaces, etc. Since the first of these is a measure of density, and the second a measure of pressure, we may state that density and pressure vary in direct proportion, provided the average speed of the molecules remains the same. The condition of constant average speed for the air molecules is satisfied if the temperature remains constant; and calculations made on this basis are called isothermal calculations. Since matters dealing with temperature change are rather involved, and since for most of the cases that will concern us calculations allowing for temperature change do not yield results greatly different from those of isothermal calculations, the latter are usually adopted as standard when dealing with die cushions.

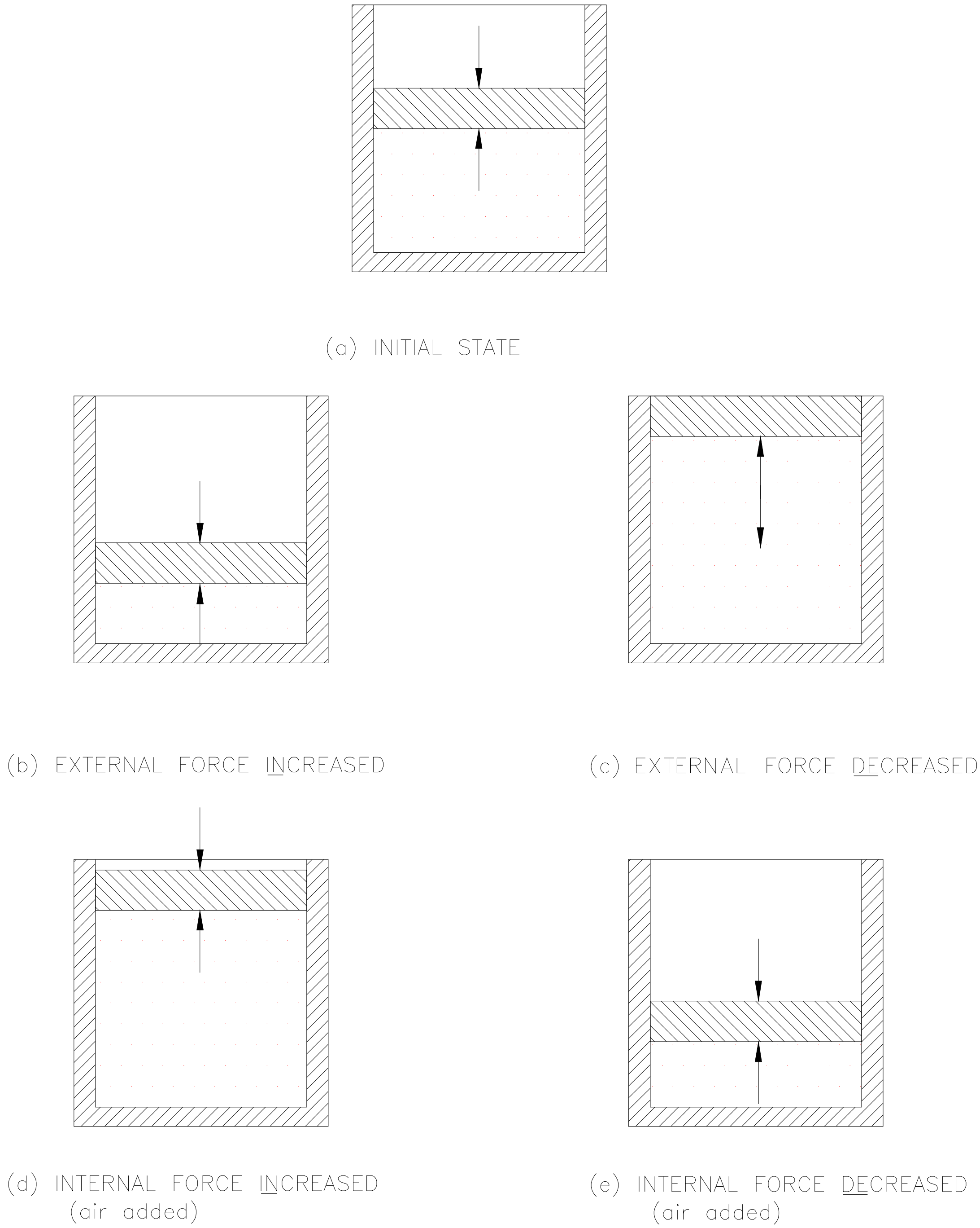


Fig. 1-2-15 Restoration of equilibrium

If the quantity of air in a container remains the same while the volume is being changed, we see that density and volume vary in inverse proportion: i.e., if we double the volume we halve the density, or if we halve the volume we double the density, etc. Since density and pressure vary directly, we may substitute pressure for density in the above statement, and say that pressure and volume vary in inverse proportion. This means that using the mathematical values for these quantities, the product of the volume of air in cubic inches by the pressure of the air in pounds per square inch is a constant value. Thus, for a variable-volume container, regardless of the position of the moveable member, we will always get the same value if we multiply the volume by the pressure. If we know the pressure and volume at one particular position of the moveable member, we can calculate either one in terms of the other for any position. The only special conditions that we must assume are a constant temperature (isothermal calculation), and that no air is added or removed.

Translating this into a mathematical formula, we have the following:

If V1 is the initial volume of the container,

P1 is the initial pressure of the air in the container,

V2 is a second volume of the container, and

P_2 is the pressure of the air corresponding to that second volume,

then,

$$V1 \times P1 = V_2 \text{ X } P_2.$$

This formula is the basis for many of the calculations we have to make in designing die cushion pneumatic systems.

In cases where isothermal calculations are not sufficiently accurate, the above formula is modified as follows:

If T1 is the initial absolute temperature of the air in the container, and T2 is the absolute temperature corresponding to the second volume, then,

$$(V1 \text{ X } P1)/T1 = (V2 \text{ X } P2)/T2.$$

The absolute temperature of the air is obtained by expressing its temperature in centigrade degrees, and adding 273. We note that if there is no reliable means available for measuring the second temperature (the initial temperature may be assumed to be room temperature), then it will be necessary to calculate the second temperature in terms of the change in pressure. This requires us to assume that there is no heat entering or leaving the system (i.e., that the compression (or expansion) is an adiabatic process), and that all work done against (or by) the air is converted into internal (or external) energy. The calculation methods that this entails are beyond the scope of the present discussion.

While we explained the relationship between pressure and force towards the beginning of our discussion, it would be well for us to restate it at this time in a more precise manner. Very often when we know the pressure of the air in a container it is desirable to compute the total force exerted against some portion of the wall, or conversely, particularly when the wall surface is that of a moveable member. Since pressure is the force per square inch, the total force against a portion of the container wall is simply the pressure times the total area involved; i.e.,

If F is the total force, P is the pressure of the air, and A is the area of the container wall, then,

$$F = P \times A.$$

Or, if we want to calculate pressure when we know the total force

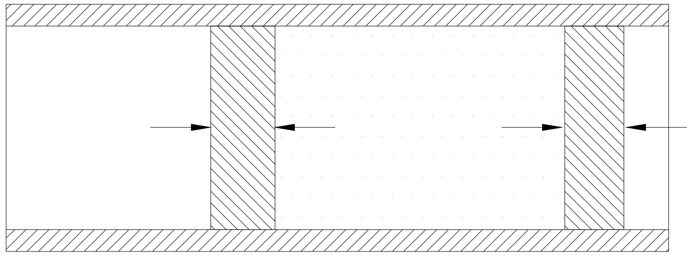
$$P = F/A.$$

Thus, if the pressure inside a container is 50 lbs per sq in., and the area of the inside surface of the moveable member is 20 sq in., then the total force exerted by the air against the moveable member is 1000 lbs. We note particularly that this force depends solely upon the air pressure and the area against which it acts} and is not influenced by forces exerted against other portions of the container wall if these do not affect the pressure.

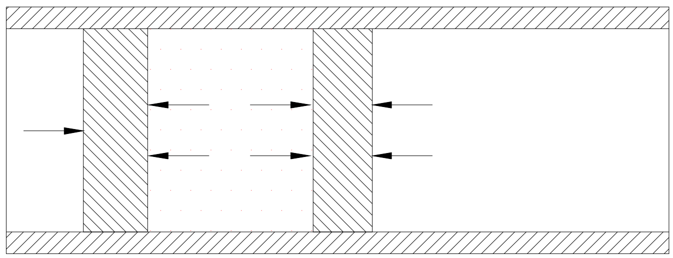
At the beginning of our discussion, we stated that the role played by air in the mechanisms we are to be concerned with involves its use for transmitting force and for storing energy. How that we have some idea as to the way air behaves when confined in a variable-volume container, let us examine briefly each of these basic phenomena. In Fig. 1-2-16a we show a variable-volume container consisting of three parts, one cylinder and two pistons. In this illustration the external and internal forces against each of the pistons are equal, and no movement occurs. In b, we have provided an additional external force against the right-hand piston. This causes the right-hand piston to move to the left, compressing the air and raising the air pressure between the pistons. This increases the internal force acting against the left-hand piston, and moves it to the left. Thus the force applied against the right-hand piston has in effect been transmitted to the left-hand piston through the air between them. Aside from the bare fact of this transmission, a number of other things should be noted:

1. It is necessary for the air to be compressed, at least a little, before the movement of the left-hand piston begins.
2. If there is no addition of external force against the left-hand piston, the rate of movement of both the pistons will be the same as soon as a condition of uniform pressure is established between them.
3. Assuming no friction, the rate of movement will be proportional to the size of the added force (the conditions for #2 still assumed).
4. Movement of the left-hand piston may be stopped if an additional external force is applied against it, which is equal in magnitude to that applied to the right-hand piston.
5. If the additional force applied to the right-hand piston is removed, the pistons will move apart until the initial separation is reestablished.

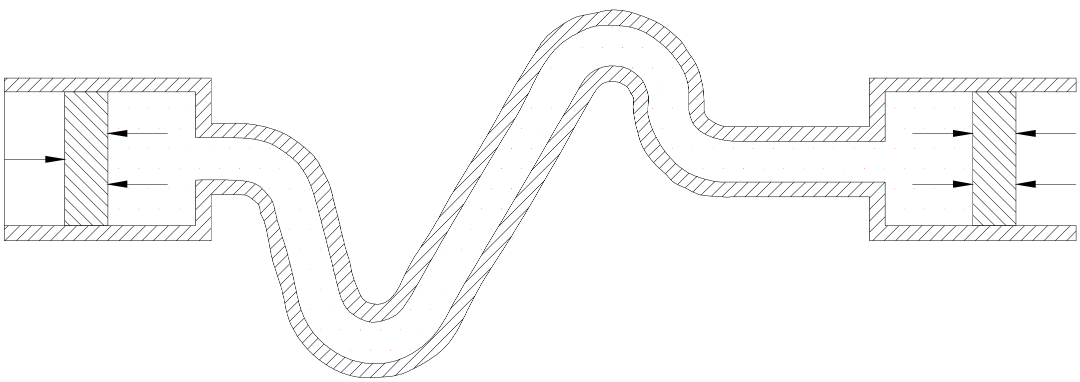
While none of these things has been stated in a particularly precise manner (i.e., the statements are not sufficient for mathematical calculations), we have indicated some of the more important phenomena that can be expected; and on the basis of these we may acquire some degree of visual insight into the transmission phenomenon, which is not always easy to understand.



(a) INITIAL EQUILIBRIUM



(b) TRANSMITTED FORCE RESULTING IN MOVEMENT



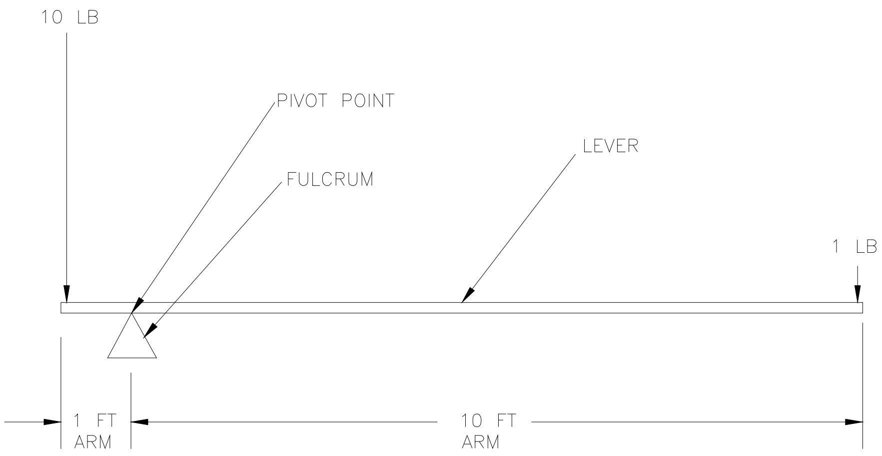
(c) TRANSMISSION OF FORCE AN IRREGULAR PATHWAY

Fig. 1-2-16 Transmission of force by air

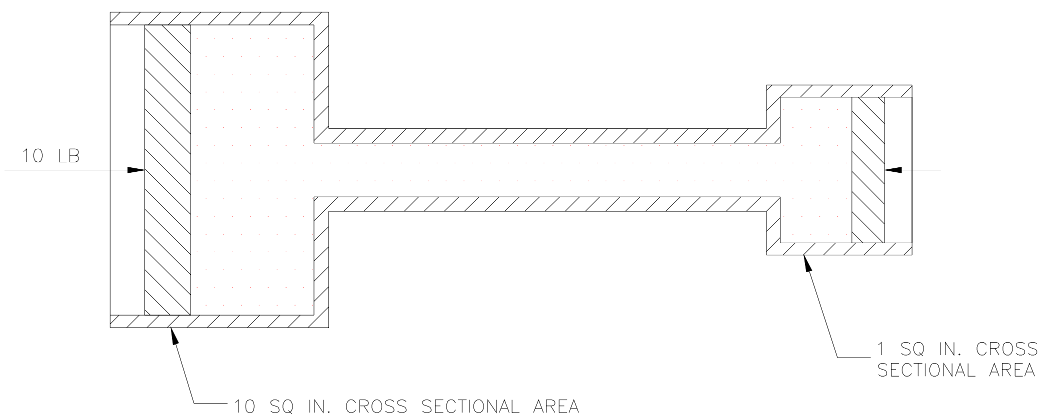
In Fig. 1-2-16, we show that much the same situation exists when we have two separate containers connected together by an irregular arrangement of pipe. So long as there is clear passage for the air to flow through, and the range of movement of the pistons does not exceed the regular areas in their respective cylinders, the transmission of force from the right-hand piston to the left-hand piston is essentially the same as in b. This makes it feasible to transmit force over long and irregular pathways with relatively little difficulty or loss.

In Fig. 1-2-16, all the pistons have the same diameter, and hence the same cross-sectional area. However, there is no general reason why this should be necessary (baring the special case of a common cylinder), and for most pneumatic systems the normal situation is in fact just the opposite; i.e., the diameters of all the pistons are different. This has a number of simple, but nonetheless important consequences. The best way to acquire an understanding of these is to examine an analogous situation in a mechanical system. In Fig. 1-2-17a we have illustrated a mechanical device known as a lever. For this particular case it is a rigid piece of steel 11 feet long, supported by a fulcrum at a point 1 foot from one end. The primary function of a lever in a mechanical system is the multiplication of force, distance, or speed. In the example shown, a force of 1 pound applied to the end of the long segment is able to counterbalance a force of 10 pounds applied to the end of the short segment. And if we assume movement of the long end downward, we see that the system would enable a force of 1 pound to lift a weight of 10 pounds. Essentially, this is the most common type of lever-application. We note, however, that the distances moved by the two ends of the lever, and the speeds at which they move, are in direct proportion to the lengths of the arms (segments); i.e., if the long end moves 10 feet the short end moves only 1 foot, and the time of movement necessarily being the same the speeds are in the same ratio.

Conversely, if we apply force to the short arm in order to lift a weight on the end of the long am, the force must be ten times the weight moved; but the weight will move ten times as far, and ten times as fast.



(a) MECHANICAL LEVERAGE



(b) PNEUMATIC "LEVERAGE"

Fig. 1-2-17 Multiplication of force, distance and speed

The pneumatic counterpart of this arrangement is shown In Fig. 1-2-17b. The small cylinder on the right has a cross-sectional area of 1 sq in., and the large cylinder on the left has a cross-sectional area of 10 sq in. By and large, the fundamental relationships between forces applied to the right- and left-hand pistons is similar to that between forces applied to the right- and left-hand lever arms in the mechanical arrangement above. However, the fact that air may be compressed does tend to complicate the matter. The

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analogous situation with the lever would involve a flexible lever, which as we can see would tend to complicate the distance and speed relationships, and introduce a time-delay between the application of force on the right and the lifting of the weight on the left (or vice versa). Analogous inexact speed and distance relationships, and a time delay are in fact important characteristics of pneumatic systems of this type. However, they do not alter the relationship between the magnitudes of the forces applied to the pistons; hence this is usually the best starting point for the evaluation of such a system.

The explanation of this force relationship is relatively simple, and represents and important application of the principles discussed above with respect to the calculations of forces and pressures. If the system is in equilibrium, the internal and external forces against each of the pistons are equal (though opposite in direction); yet the pressures are also equal for all parts of the system. If we calculate the magnitudes of the individual forces in terms of these (i.e., multiply the pressure by the areas of the pistons), we find that the magnitudes of the forces are proportional to the areas of the pistons. If we change the external force against one of the pistons, this causes a change in the internal pressure, which is proportional to the change in force. However, this change in pressure has the same relationship to the two piston forces as the initial pressure had; and hence the change in force will also be proportional to the piston areas. Since movement of the pistons will involve a flow of air from one cylinder to the other, and since the capacities of the cylinders are in proportion to their cross-sectional areas, distance and speed will also vary; but in a ratio that only tends to be the inverse of the areas, due to the complications introduced by the compression and expansion of the air.

Since the functioning of a pneumatic system invariably involves the flow of air from one part of the system to another, it is necessary for us to investigate this flow in and of itself; i.e., there are a number of important consequences that modify the simple pressure effects that we have been primarily concerned with up to now. We touched on these above as they relate to compression and expansion, but we have not yet viewed them independently. To this end, let us refer to Fig. 1-2-18, which illustrates the basic situation in terms of two containers connected by a piece of pipe. We should notice first that if we apply force to the right-hand piston in order to cause movement of the left-hand piston, our overall objective involves the movement of air from the right-hand container to the left-hand container. And this air must flow through the connecting piece of pipe. This requirement for the movement of air gives rise to the following:

1. Instantaneous response is impossible. The second piston will move only after an interval of time has passed, and it will continue to move for a comparable interval after the first piston has stopped.
2. The movement of the second piston will tend to be smoother than that of the first (assuming that irregularities exist); shocks and irregularities will not be as pronounced because of the rapid mixing of air molecules.
3. There will be friction between the air and the connecting pipe, which will slow down the air as it passes through, and reduce its capacity to move the second piston.

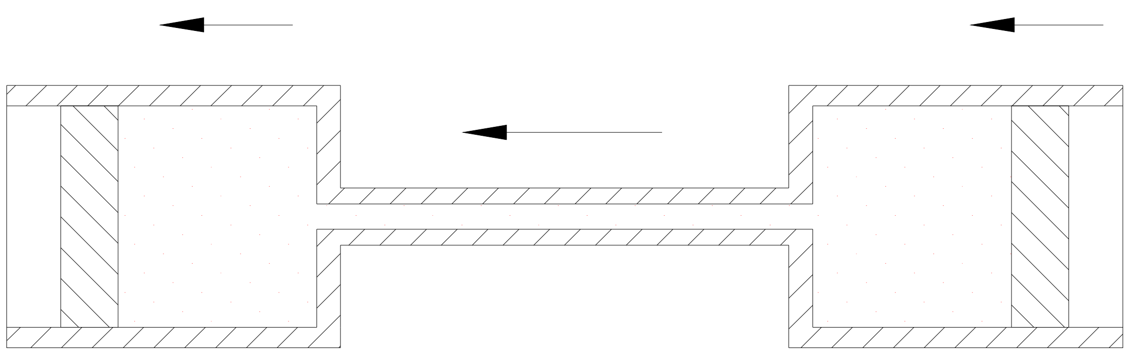

Fig. 1-2-18 Flow and friction

As regards this latter situation, while we cannot think of friction between air and its container in precisely the same manner as that between, say, two pieces of metal, we must not forget that both the air and the container wall are substances, and must in some sense rub against each other as the air flows through. Further, the pressure of the air in the pipe is an obstruction in itself; molecules colliding with the pipe wall will rebound crossways to the flow of air, and become obstacles for molecules moving along the pipe axis. A remotely analogous effect will be seen if we attempt to pass a rag through a piece of pipe by inserting one end of it into the pipe and attempting to push the rest behind; we will find that it is much easier to pull the rag through from the other end, perhaps by using a piece of string. Of course air moves much easier, but this "stuffing" effect does exist, and increases with the pressure. Other factors are also involved, but their complexity prohibits their treatment here.

In view of the above remarks, we can see that air is not always an ideal medium for the transmission of force, even though it is the best medium for a large and important category of operations. In particular, we note that pneumatic systems operate best when they operate slowly; here the bad effects of flow phenomena are minimized. As the speed at which a pneumatic system must operate is increased, the conditions of airflow give rise to what we may regard generally as reduced-effect phenomena. What we mean by this is that a transmitted force (which may be multiplied or divided via the leverage principle) may be intended to move a particular machine element a specific distance, at a specific speed, and with a specific amount of force. Flow phenomena may reduce any one, or all three of these; and at rapid operating rates the system can become ineffective. For instance, it may be desired to move a given piston a distance of, say, 10 inches; but at high speeds it may move only 8 inches. If distance is critical enough, this may not be satisfactory. Or, a given operation may require a force of, say, 400 pounds, but receive only 300 pounds; or perhaps it must be accomplished in 1/2 second, but actually takes 3/4 second. For die cushion systems, flow phenomena act to decrease the effectiveness of external surge volumes, a problem that we will go into further in later sections.

Our next topic is energy, which we will treat only briefly. Energy as a concept is generally rather abstract, and usually admits of rigorous treatment only in terms of higher mathematics. Our purpose in discussing it here is merely to familiarize the reader with those ideas pertinent to the clarification of certain inconsistencies implicit in an independent discussion of force and movement; such as we have engaged in thus far. These do not invalidate previous conclusions, but they do serve to limit their application to complex systems; hence it is advisable that the readers have some notion as to the origin and derivation of such limitations. Usually the essential ideas can be satisfactorily treated in terms of simplified concepts of storage and conservation.

Storage of energy, so far as we are concerned here, involves the relatively straightforward notion that energy is absorbed by a quantity of air whenever it is compressed, and released or given up by it when it is expanded. In more formal language, we say that work is done on or against air when it is compressed; and work is done by air when it is expanded against a resisting force. These two processes are illustrated in Fig. 1-2-19. The fact that storage occurs revolves around the fact that if its energy is not lost through the radiation of heat, then the compressed air acquires a capacity to expand the container (if the compressing force is removed or reduced); or the capacity to cause some other commensurate movement. Since energy is most often defined as a capacity to do work, the acquired capacity is actually acquired energy. While our approach obviously lacks scientific precision, it does serve to point out the most significant aspect of this as it affects the systems we are investigating. Hence, when a variable-volume container is collapsed, and the air in it compressed, the air absorbs and stores energy; which is than released when the compressed air is allowed to expand the container, and do work by moving machine elements.

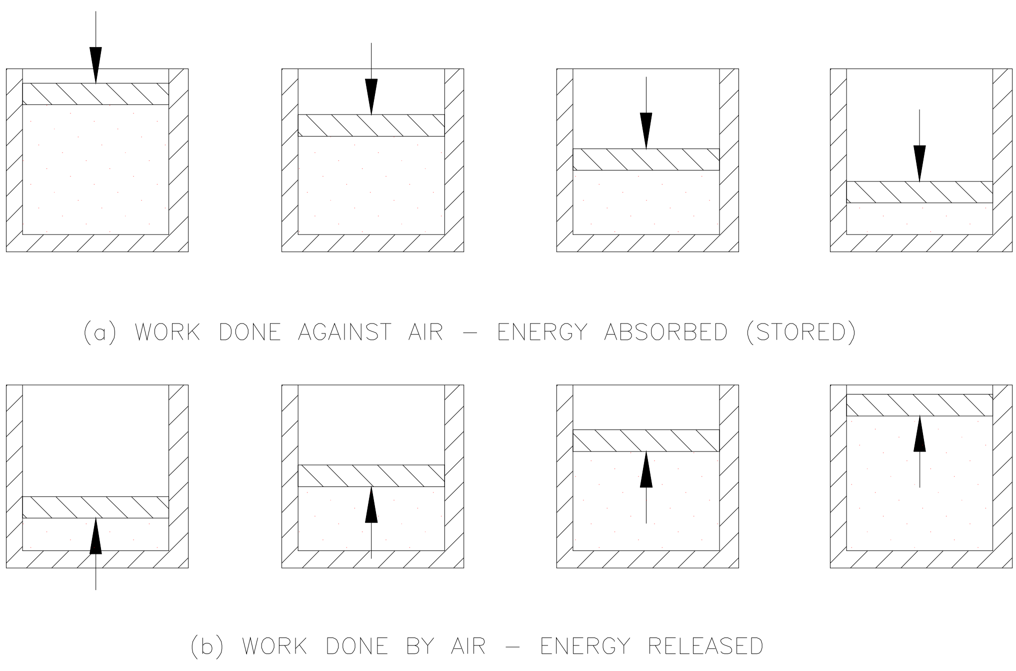

Fig. 1-2-19 Storage and release of energy

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The idea of conservation of energy is a slightly different matter, since this pertains to a basic law of Physics used extensively in quantitative evaluation. Unlike storage, which is a process, conservation is a result. And the generality of this result enables us to predict and understand the specific results of a great variety of otherwise inexactly related physical processes. For the present application, conservation can best be understood in terms of the resisting-force definition of energy (or work). In this, the energy acquired (or lost) is equal to the magnitude of the force resisting movement times the distance through which movement (or displacement) occurs. To apply this here, we must modify slightly some of our previous viewpoints, To begin with, when we apply force in collapsing a variable-volume container, the magnitude of the external force causing the displacement of the moveable member is equal to the sum of two other forces: One of these is the internal force exerted by the pressurized air, which we have dealt with extensively above. The other, which we have not previously mentioned, is the force of inertia of the moveable member; this varies as the mass (weight) of the moveable member, and its acceleration (i.e., it tends to zero when speed (or the lack of it) is constant). When the container is being expanded, the inertial force opposes the internal force exerted by the air, and adds to the external in a manner analogous to that occurring during compression. If we simplify the matter by regarding the inertial force as always being part of the force resisting movement, we may state flatly that whatever the position or direction of displacement of the moveable member, the internal and external forces are always equal. As the air is compressed these forces increase; and as it is expanded, they decrease.

We note that these statements are in apparent conflict with earlier statements in that they postulate constant equilibrium; however, this is because we are now considering the force of inertia, which we ignored earlier, and the general validity of the earlier statements is not impaired if we do not apply them to situations other than those they were developed for. It is interesting, nevertheless, to observe the difficulties that arise when we attempt a rigorous general treatment; and we can better appreciate the advantages of simplification.

The heart of the conservation notion for the present context is that the amount of energy stored or released in any given instant is dependent upon the magnitude of the forces (internal -and external) at that instant, and the distance through which they cause displacement. It is independent of the direction of displacement. Hence, at any position of the moveable member, there is a corresponding potential for work. If we divide the total distance through which the moveable member is displaced into short segments, and use for each of these segments the average magnitude of the internal (or external) force, each segment then corresponds to a work element. as is illustrated in Fig. 1-2-20. This element will be absorbed by the air in one direction, and released by it in the other. However, either way, the total is the same. Thus, energy is neither created nor destroyed by the displacement of the moveable member; it is merely transferred to and from the air. Of course, in practice this idealized situation does not exist; friction transforms a portion of the energy involved into heat, and this is subsequently conducted or radiated away. And while this does not actually constitute the distraction of energy, it does entail its loss by the pneumatic system. However, conservation does tend to be approximated by a well-constructed system operating at a slow or moderate rate of speed; and this situation is the basis for many of the principles used in pneumatic system design.

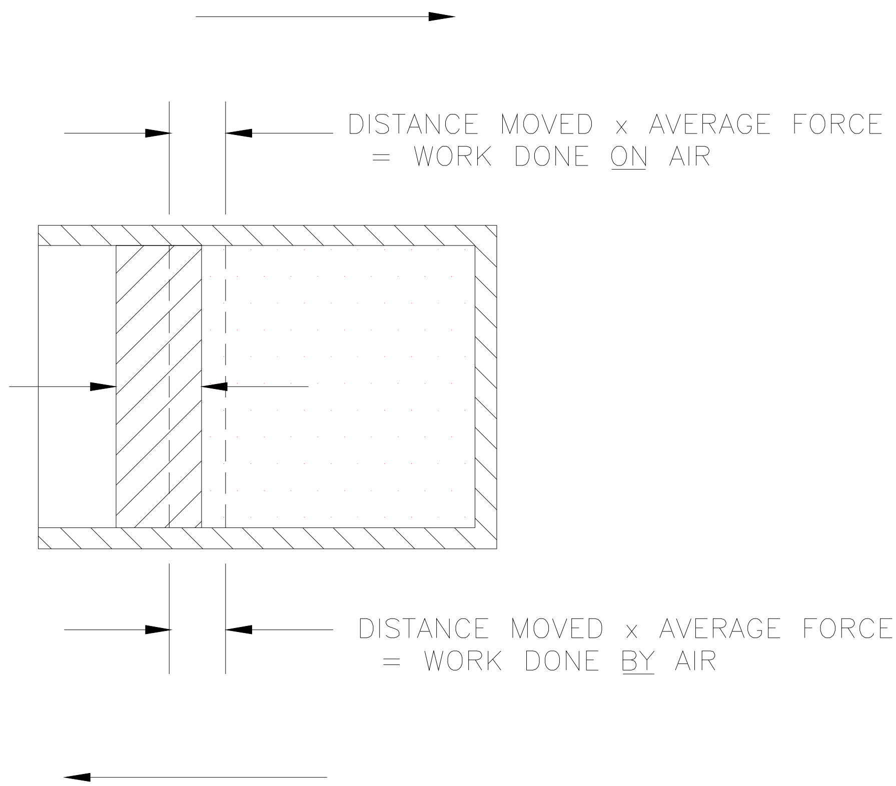


Fig. 1-2-20 Conservation of energy

Before concluding the present section, we want to discuss briefly a simplified pneumatic system, such as would be encountered in die cushion applications.However, we must first go back and pick up a subject that we discussed at the beginning of the section, that of atmospheric pressure. As we noted then, the effect of gravity on the atmosphere of the Earth results in an air pressure of approximately 15 pounds per square inch at sea level; and this is the value we usually take for all normal situations, Now, very few if any of the pneumatic systems we will have to deal with will operate in the absence of this atmospheric air pressure; i.e., we will not find machines working in a vacuum. Thus, all the pneumatic systems we can expect to encounter will be in a sense immersed in air having a pressure of 15 psi, as indicated in Fig. 1-2-21. For most situations, this has the effect of adding to the external force against the moveable member; which thus reduces the effect of the internal force just as though the internal air pressure had been reduced by 15 psi. Hence, in evaluating the effectiveness of internal air pressure, we normally use this decreased value, and then ignore the atmospheric pressure altogether. The decreased value of internal air pressure, which Is the difference between the actual value and atmospheric pressure, is commonly known as gauge pressure, because this is what our pressure gauges are designed to measure; and we designate it by the abbreviation psig (although the simpler psi is more common when this is the only pressure we are concerned with). The actual pressure is then indicated by the abbreviation psi abs, with the abs meaning absolute. We note that it is this actual (absolute) pressure that must be applied when calculating volume-pressure relationships, so that the differentiation can sometimes be quite important. We will discuss this matter further in the next section.

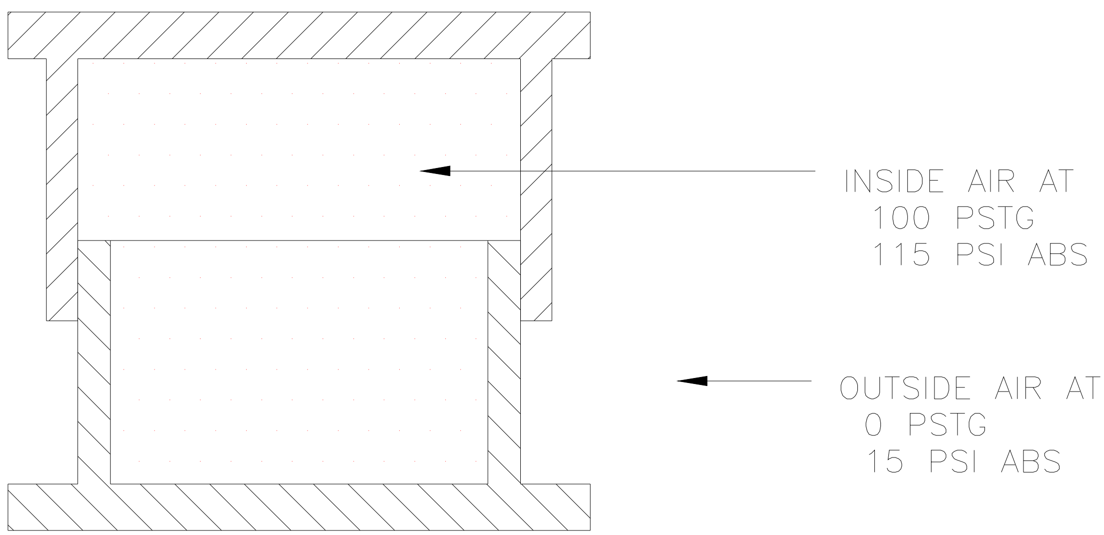


Fig. 1-2-21 Immersion of a container in atmospheric air

While we have up to this point discussed containers in a general and often abstract sense, it is obvious that we cannot continue this policy when actual pneumatic systems are involved. Hence, we must begin to be more specific, and devote more attention to the individual features and characteristics of the many different container types. The first step in this direction involves the separation of containers into three primary categories, defined in terms of the functions of containers as elements of systems. These categories are:

1. **Suppliers** - The function of a supplier is, as the name implies, to supply pressurized air for use by other containers. Usually the quantity supplied is expressed in terms of pressure, and volume per minute.
2. **Storers** - As the name implies, this category includes those containers used to store air received from suppliers for future use.
3. **Users** - This category includes the largest variety of containers, all of which are designed to perform some particular kind of operation in conjunction with other machinery elements.

In order to show how these containers relate to each other, we have illustrated a simple pneumatic system in Fig. 1-2-22, which includes in addition to the basic containers several other devices necessary to the functioning of the system. We will discuss the parts of this system beginning at the top and working down.

The uppermost container in the figure is a standard supplier commonly known as an air compressor. Generally speaking, most suppliers are variations of this particular device, although the mechanical principles involved sometimes differ appreciably. The right-hand wall of the cylinder has two openings for the passage of air; each of which is occupied by a check valve. A check valve is a simple device that permits the flow of air in one direction only. In this case, the upper valve permits the flow of air into the container, and the lower valve permits flow of air out of the container. The operation of the compressor begins with the piston all the way to the right, from which position it is pulled to the left by the piston rod, which is activated by suitable mechanical devices. As the piston moves to the left, the pressure of the air inside the compressor drops rapidly, since there was very little inside to begin with. Since the compressor is immersed in atmospheric air at 15 psi abs (as we mentioned above), the force exerted by this air against the upper check valve causes it to open, and air is forced into the compressor. This process continues until the piston reaches the left extreme of its stroke, and the pressures inside and outside the compressor are equalized at 15 psi abs.

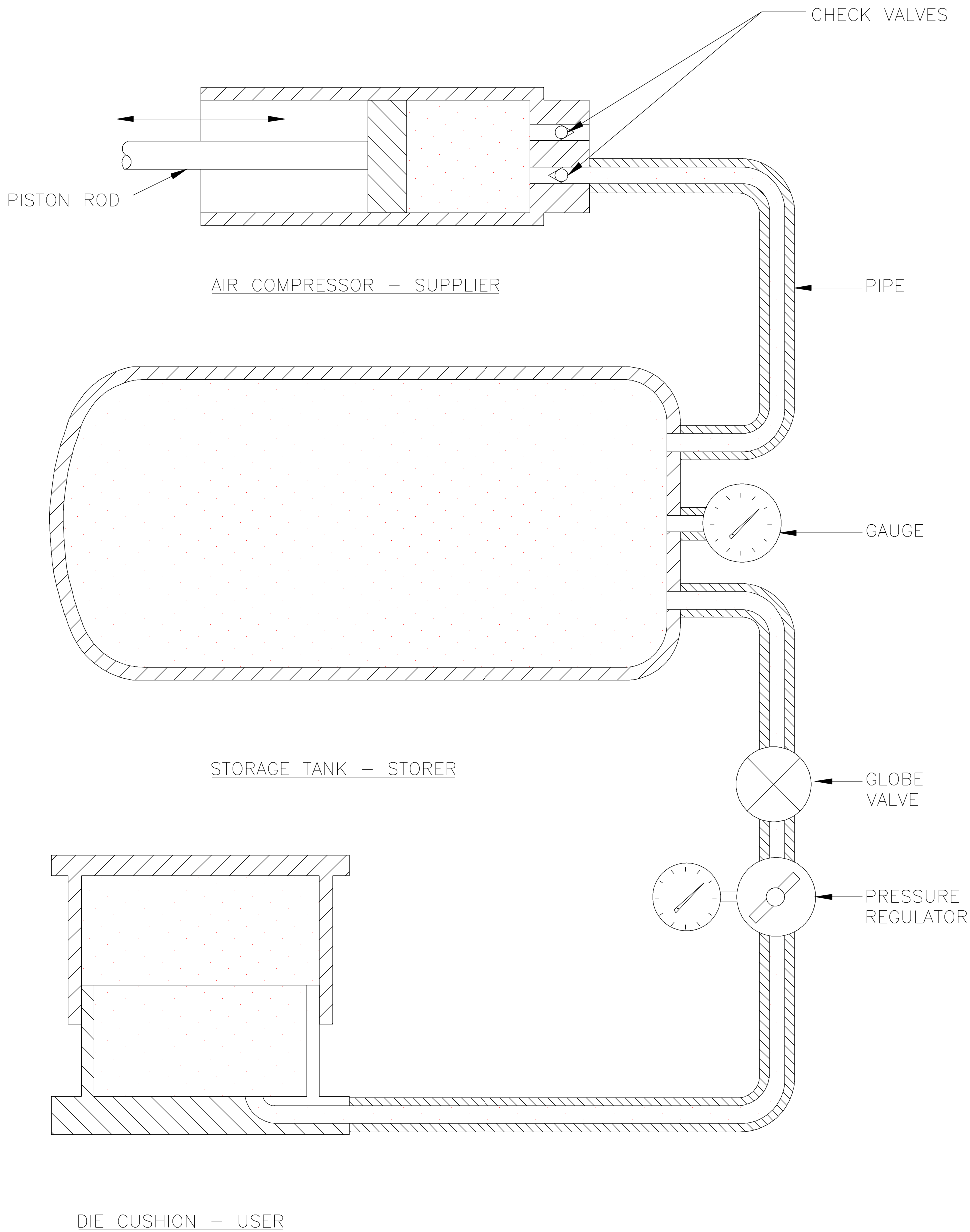


Fig. 1-2-22 Elementary pneumatic system

In the next phase of the operation, the direction of movement of the piston is reversed. This begins to increase the internal air pressure; which closes the upper check valve, and opens the lower check valve. With the lower valve open, the air in the compressor is able to flow into the pipe that connects it to the second container, the storage tank. Note that air will open the lower valve and flow into the tank only if the pressure in the compressor is greater than the pressure in the tank. This process of pushing air into the storage tank continues until the piston again reaches the right extreme of its stroke. Here it again: reverses direction, the lower check valve closes and the upper one opens, and the cycle begins all over again.

With each successive cycle of the compressor, the air pressure in the storage tank increases a small amount. This pressure is indicated on the gauge, which measures the difference between it and the pressure of the atmosphere outside (gauge pressure). When this pressure reaches a certain point, say 100 psig, the compressor is shut off, and no further air is supplied. The system is usually constructed so that this occurs automatically. As air is used from the storage tank, the pressure in the tank will begin to drop so that it will be necessary, eventually, to turn the compressor back on. Usually this is also done automatically so that in a typical system the compressor might be turned off when the tank pressure reaches 100 psig, and turned back on when it drops to 90 psig.

The pipe leading from the storage tank to the air user, in this case a die cushion is blocked by two devices. The uppermost of these in the diagram is a globe value, which is simply an off-on device for opening and closing the line. Below this is a more sophisticated device known as a pressure regulator. The reason for the regulator is that we may want to operate the cushion at a pressure less than the pressure in the storage tank. The regulator i3 so designed that it will allow air to pass from the storage tank to the cushion only when the pressure in the cushion is less than a predetermined amount, say 50 psig; when the pressure in the cushion reaches this point, the regulator closes the line. If the cushion pressure subsequently drops below this point, the regulator then reopens the line until the 50 psig is reestablished.

In calling a die cushion a user of air, we should remark that it is not to any degree a continuous user. It requires air initially to charge it to the proper pressure; but afterwards it requires only enough to compensate for leaks. Some devices, such as air motors, operate in such a manner that the air they use is allowed to expand until the pressure is reduced to near 15 psi abs (0 psig); then the air is exhausted into the atmosphere. Such devices require a constant flow of air from the storage tank; and often a constant, or nearly constant, operation of the compressor.

1-2-4. DIE CUSHION PNEUMATIC SYSTEMS

As we indicated at the beginning of the previous section, the material to be presented here is a little beyond the immediate needs of the general reader, and if he has not read Section 1-2-3, he should skip this section as well. Our purpose here will be to enlarge upon our discussion of basic pneumatic system theory by treating specifically those systems, or portions of systems, built around die cushions. With reference to the concluding discussion of the preceding section, and to Fig. 1-2-22, we will consider the treatment there of air compressors and storage tanks as adequate for our present purposes, and will refer to these only indirectly by designating the pipe from the storage tank as the supplier for a die cushion system, and labeling it the shop air line (refer to Fig. 1-1-17).

Before we can investigate die cushion systems in any detail, we must first consider in a general way what these systems are supposed to accomplish. It is in terms of these objectives that they are designed and constructed; and hence these must provide the basis for study and evaluation. We will assume at this point that the reader understands the basic purpose of die cushions as relates to press operation if he is unsure of this, he should go back at this time and reread Section 1-1-3 on The Function of Die Cushions. In view of the applications discussed there, a die cushion system has the following general objectives:

1. It is to provide support of sufficient magnitude for cushion pins to assure proper operation of tooling.
2. It is to provide for the adjustment of die cushion force, over an adequate range, to suit the cushion-force requirements of a variety of different jobs.
3. It is to maintain constant cushion force, within reasonable limits, as the cushion is collapsed, so that the original adjustment may be optimum, rather than merely a compromise.

While the importance of these objectives can best be understood in terms of actual die cushion application, the effectiveness of a system in achieving them can be treated independently; and this will be our basic approach here.

In order that we will be able to determine whether or not a given system provides sufficient cushion force, we must first be able to calculate the actual force provided. This is accomplished via the procedure indicated in the preceding section; i.e., multiplying the air pressure in psig times an area in square inches. However, there are a number of questions that often arise concerning the selection of this area, and before proceeding with some actual calculations we want first to enlarge upon the previous discussion by investigating the surfaces involved. — A preliminary point that we should males is that pressure, which is force per unit area, is directed at right angles to the surface against which it acts. This does not imply that the molecules involved are in any way restricted to right-angle collisions, but rather that when the collisions are averaged out, all force components other than those perpendicular to the confining surface, are cancelled out.

With this point in mind, let us refer to Fig, 1-2-23, which shows four types of piston surfaces. In a, we show a simple, flat piston surface, perpendicular to the axis of the cylinder, and hence to the direction of piston movement. This means that all of the force exerted by the internal air against the piston will tend to move it to the left. And the magnitude of this total force may be determined simply by multiplying the area of the inside surface of the piston by the gauge pressure of the air; i.e.,

$$F = P \times A.$$

This is fairly straightforward, and entails no difficulty. However, when the inside piston surface is not of this type, such as the surface in b, the matter is not quite as simple. Here the inside piston surface is not in all places at right angles to the direction of piston movement, and the air acting against it is not at all points as effective a mover as in a.

In order that we may understand the nature of the problem, and the method of solving it, we have constructed the inside piston surface in b in such a manner that there are only three basic situations: one - where the pressure acts in the direction of piston movement; two - where the pressure acts perpendicular to the direction of piston movement and three - where the pressure acts at a particular angle to the direction of piston movement. Note that this angle, given in the figure as Ø, is the same as the angle between the actual piston surface and a hypothetical surface identical with the surface in case a. Each of these basic situations is treated as follows:

1. Where the air pressure acts in the same direction as the piston movement, we calculate as in case a. Thus we determine the total area of those portions of the piston surface perpendicular to the direction of movement, and multiply this by the pressure of the air.

2. Where the air pressure acts at right angles to the direction of piston movement, it has no effect whatsoever on displacement; and we simply ignore it and the areas against which it acts.

3. Where the air pressure acts at a particular angle to the direction of piston movement (other than a right angle), we resolve it into two components - one in the direction of movement, and one perpendicular to it - given as the a-component and the b-component in the figure. We then treat these as above, ignoring the b-component, and multiplying the a-component times the area of the surface involved.

The total force acting to cause movement is then the sum of the forces determined in #1 and #3 above.

At this point we take note of two important relationships: one - the magnitude of the a-component given above is P (the air pressure) times the cosine of the angle that P makes with the direction of piston movement (angle Ø, as noted above); and two - the area of the piston surface involved is the quotient of the corresponding portion of a hypothetical surface identical with that in a by the cosine of angle Ø. Thus:

If A is the hypothetical surface (as in a), and P is the air pressure, then the area against which P acts is

$$A/\cos Ø,$$

and the effective component of P (the a-component) is

$$P \cos Ø j$$

DAYTON DIE CUSHIONS

so that the effective component of the total force involved is

$$F \, Z \, (A/\cos Ø) \times (P \cos Ø) = A \times P.$$

If we add this quantity to the quantity obtained as in #1 above we find that the total force acting to cause movement is the same as that obtained if we multiply the actual air pressure times the area of a hypothetical piston surface identical with that in case a. Hence, cases a and b may be calculated in the same way, and we may ignore the irregularities in the piston surface of b.

If we employ some of the simpler concepts of advanced mathematics, and divide the surface of any piston into small parts of infinitesimal size, we find that each of these parts may be treated under one of the three situations of case b. Hence, if the air pressure is the same at all points, the result stated above is a general result, and applies to all pistons. A minor simplification is possible if we note that the internal piston area in case a is the same as the internal cross-sectional area of the cylinder. Thus, we may state as a general rule that the total force acting to cause movement is equal to the product of the cross-sectional area of the cylinder and the pressure of the air. Hence, if the cylinders are the same size, the total force in case c, where the internal piston surface is arbitrarily irregular, is the same as in cases a and b.

This result is in some respects rather intuitive for the above cases, and the explanation may seem at least partially superfluous. However, while the intuitive notion yields correct results in this instance, it can just as easily yield incorrect results, particularly where the situation is more complicated. We have included the above explanation specifically so that the reader may understand the mathematical reasoning involved, and thus be in a better position to solve more difficult problems. In particular, the general case above yields two total forces: one - the total force against the piston surface; and two - the total force acting to cause movement; the latter being a portion (or component) of the former. In case a these two forces are identical; but this represents a rather rare occurrence.

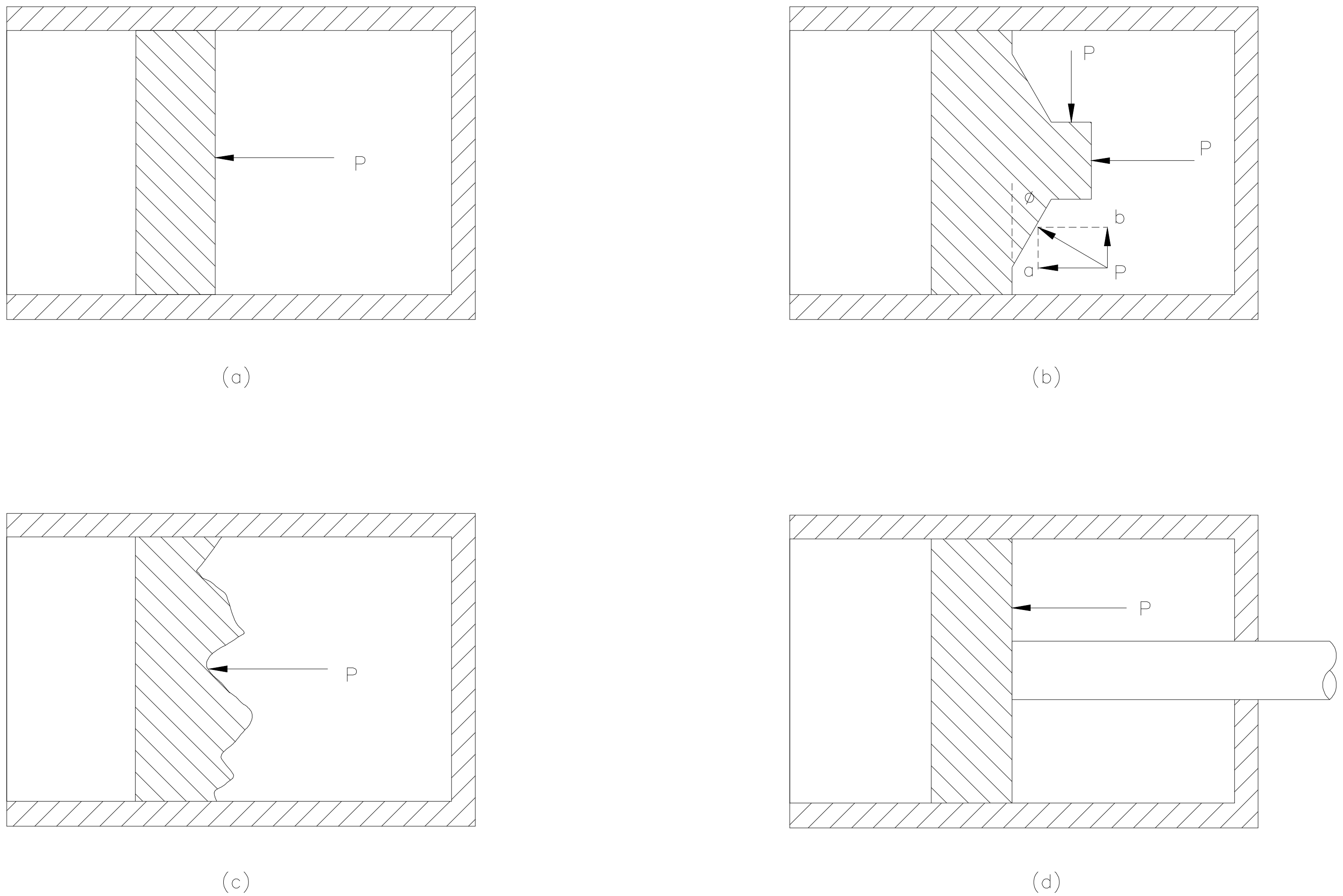


Fig. 1-2-23 Piston surfaces

A slightly different case is illustrated in Fig. 1-2-23d, where part of the irregularity involved is a piston rod extending through the cylinder wall. Here the pressure acting against the outside end of the rod is different from that inside the cylinder, and this case does not reduce to as simple a form as cases a, b, and c. In essence, we have a situation where the effective internal area of the cylinder is reduced by the area of the rod at the point where it passes through the cylinder wall. If then, in the above formula for total force, we substitute for A the difference between the internal cylinder area and the rod area, we will have the total force exerted by the internal air. To this we must of course add the force exerted by the rod, which may be in either direction, or may be zero, depending upon the circumstances outside the cylinder. We will find that this subtraction of rod area from internal cylinder area is typical of problems where a rod of one kind or another is involved.

Referring now to Fig. 1-2-24, we may apply these results to the standard Model "C" cushions discussed in Section 1-2-2. In Fig. 1-2-24a, we note that the piston has an irregular internal surface, but that the air pressure against all parts of that surface is the same; hence we may apply the general result derived above, and use the internal cross-sectional area of the cylinder. Since this cross section is circular, we may use the familiar formula

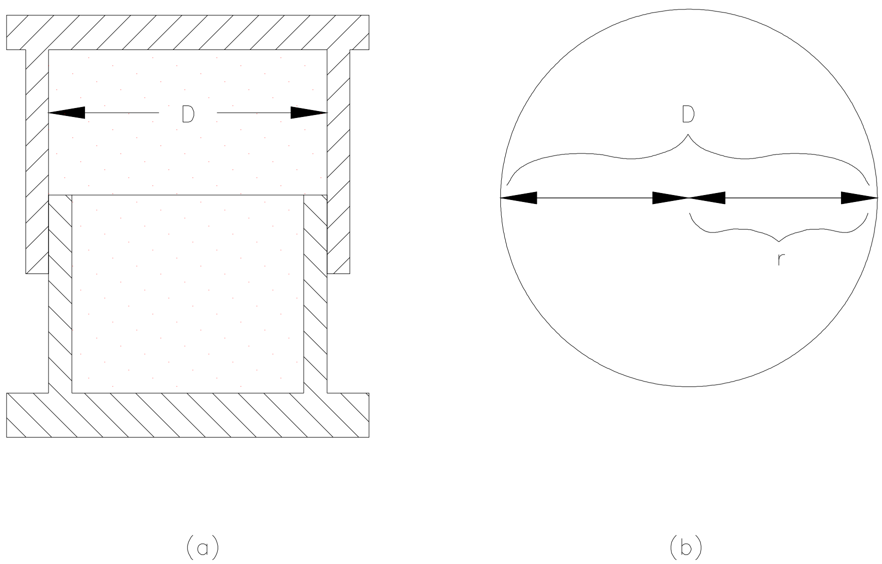
$$A = \pi r^2,$$

noting that r (the cylinder radius) is one-half the internal diameter (D), which is the dimension used in designating cushion size (Fig. 1-2-24-b). For instance, a C-14 cushion has a 7-inch internal cylinder radius.

In determining the total force provided by a cushion, we must multiply the cross-sectional area of its cylinder by the actual pressure of the air inside. However, for determining maximums we use the maximum air pressure that the pneumatic system can deliver, as long as this does not exceed the maximum pressure for which the cushion was designed. For instance, if the maximum air pressure the system can deliver is 80 psig, and the cross-sectional area of the cylinder is 100 sq in. then the maximum total force the cushion can provide is 8000 pounds. However, since the capabilities of different pneumatic systems are peculiar to the individual systems themselves, this figure applies only to the particular system under discussion, and is subject to change as the system changes. In order to achieve a more general application, we prefer to rate cushions in terms of the maximum pressures for which they were designed, rather than that of any particular system. This permits a fairer evaluation of the capability of the cushion itself, and permits us to formulate standard reference tables. An example of such a table is given in Fig. 1-2-24c. This covers all standard sizes of Model “C” cushions, which are rated at 100 psig maximum air pressure. Note that we have given the cushion forces both in pounds and in tons, so that the reader may compare them. However, for most applications the rating in tons is the more practical, and is the only one given on our standard forms and tables.

While the maximum tonnage that a given cushion can provide is obviously important, since this must be at least as great as the maximum that will be required, it is still the actual tonnage being supplied for any particular job that is ultimately of primary importance. That is, the maximum potential tonnage depends upon the cushion selected and the air pressure provided by the pneumatic system (shop air line); once this cushion has been installed on a press, its maximum potential concerns us only in that it either is or is not adequate. When we consider the actual operation of the press, which is what the customer will be primarily concerned with, the important question concerning the die cushion is how well (and how accurately) it satisfies the cushion-force requirements of the tooling. Thus we must consider the adjustability and constancy of its internal air pressure; with the understanding that it is with regard to these that the customer will judge its performance.

To begin with, we should distinguish between adjustment and regulation. Of course no complete delineation is possible, since the meanings of the terms do tend to overlap; but there are significant differences in general application that the reader should be familiar with. Adjustment usually involves the selection of the correct air pressure from the range of possible values, and the setting of regulating devices to achieve this. On the other hand, regulation usually involves the maintaining of the value arrived at during adjustment. Thus, the regulator that we provide with our cushion installations does not determine the correct air pressure; it merely maintains whatever pressure the press operator sets it for. The reason we stress this point is that the adjustability of a die cushion installation must of necessity involve the methods whereby the correct air pressure is determined and subsequently set at the regulator, as well as the accuracy with which the regulator is able to maintain it.



Force equals pressure time area

Cushion Model	**Cylinder Area (Sq. In.)**	**Force @ 100 psig**	
		Pounds	**Tons**
C-6	28.2	2,820	1.4
C-8	50.2	5,020	2.5
C-10	78.5	7,850	3.9
C-12	113	11,300	5.7
C-14	154	15,400	7.7
C-16	201	20,100	10.0
C-18	254	25,400	12.7
C-20	314	31,400	15.7
C-22	380	38,000	19.0
C-24	452	45,200	25.6

(c)

Fig. 1-2-24 Standard calculations for Model “C” line

There are two devices in a die cushion system that are used for air-pressure adjustment; these are the regulator and the drain cock. The regulator is used to add air, and the drain cock is used to remove it. The reason we use the drain cock (which is also used to remove moisture from the system) in this way is that the regulator is essentially a one-way device; i.e., it permits high-pressure air to flow into the cushion if the cushion pressure is below the regulator setting, but it cannot cause air in the cushion to flow upstream into the shop air line if the cushion pressure is too high. While normal leakage will eventually reduce the cushion pressure if the regulator is adjusted to a lower pressure, it is almost always more practical to reduce cushion pressure rapidly by opening the drain cock.

Roughly, the adjustment procedure is as follows:

1. If the cushion force is too small, we turn the regulator handle until the gauge indicates a higher pressure (Note that the gauge indicates the pressure in the cushion, and not the pressure in the shop air line.). This upward adjustment should be made in small increments until the correct cushion force is obtained.
2. If the cushion force is too large;
    a. First, we reduce the regulator setting, noting that this will not cause the gauge to register a lower pressure.
    b. Second, we open the drain cock momentarily, and allow a little of the air inside the cushion to escape. If our initial adjustment was arrived at correctly (i.e., by small upward increments), it will not be necessary to allow much air to escape in this manner.
    c. We then adjust the regulator upward a small amount until proper cushion force is obtained (essentially as in #1).

This procedure, of course, indicates a trial-and-error determination of the correct cushion air pressure; otherwise it would only be necessary to drain out all the air, and then adjust the regulator until the gauge indicated the correct setting. This is the procedure followed for reruns of jobs where the correct cushion pressures have been previously determined and recorded. However, trial-and-error determination is the general method. Usually past experience will indicate an approximate regulator setting, but only actual trials can practicably indicate the exact one. The usual procedure is to set up the tooling, adjust the regulator to slightly less than the setting indicated by experience (or by calculation, if experience is not sufficient), and then run off sample parts, one by one, slowly adjusting the pressure upward until the desired results are obtained. The correct cushion air pressure should then be recorded for future use. Information regarding, the calculation of cushion tonnage for drawing work will be found in the Installation Instruction and Service Manual. Determining air pressure from cushion force, or vice versa, is essentially a matter of applying the formulas and procedures discussed above (i.e., either F = P x A, or P = F/A).

The range of adjustment has as its upper limit the maximum air pressure that the shop air line can provide (within the limits of cushion design and rating). Its lower limit, however, is not quite so clearly defined. Theoretically it is possible to use the cushion at very nearly zero pressure (0 psig), but if accurate cushion tonnage is necessary this may not be practical. Oar pressure regulators are normally accurate to within a few percent; but as the regulated pressure drops below 10 psig this accuracy can no longer be maintained; and for this reason very low pressure settings are not recommended. If it is absolutely necessary that such low settings be used, the standard regulator may in some instances be temporarily replaced by a special low-pressure regulator; however, this should be done only with the approval of Engineering.

In the preceding section we discussed the fact that as the volume in a container is decreased, the pressure of the air inside is increased. Since the depression of the pin pad by the pressure pins as the ram passes through the lower part of its stroke decreases the volume of the die cushion, we must expect that the pressure of the internal air will increase; and hence the cushion force will increase. The amount of pressure increase will depend upon the ratio between the volumes at the top and the bottom of the draw; which follows from the formula

$$V1 \times P1 = V2 \times P2,$$

or

$$P2=(V1/V2) \times P1$$

This increase in cushion force means that the adjustment we discussed above is not for one particular pressure, but rather for a range of pressure. This range is:

from P1 to P2 ,

or if we express it in terms of cushion force,

from F_1 to F_2.

Therefore, in designing a cushion system, we design it in terms of these pressure ranges; which as we will see considerably complicates an otherwise simple arrangement.

In order to avoid confusing this range of pressure variation with the range of adjustment explained above, we usually treat it in terms of the pressure at the top of the draw and the amount of pressure increase, the increase being called the buildup, and expressed as a percentage of the initial pressure. Thus, if the initial pressure is 100 psig, and the pressure at the bottom of the draw is 125 psig, the buildup is 25%. This percentage expression is especially useful, because in evaluating the acceptability of a particular range of pressure variation the percentage of increase applies directly to cushion force, and requires no intermediate calculation.

While the formula relating pressure and volume is fairly simple and straightforward, actual calculation can become involved and confusing. There are two reasons for this: First, the calculation of volume, which is necessary before the pressure-volume formula can be applied, is not usually simple; generally the shapes of the interiors of containers are somewhat irregular, and this irregularity must be accounted for in one way or another. And second - the pressures called for in the formula are absolute pressures; if we desire accurate results, we must convert our gauge pressures to absolute pressures before using the formula, and then convert the answers back into gauge pressures before applying them to actual die cushion systems. Failure to do this will lead to error; and while this error la often small, it can nevertheless be important.

In order to understand the nature of this error, let us examine a special example, which deliberately overemphasizes the significance of the error for the purposes of illustration. In Fig. 1-2-25, we show a die cushion with the cylinder in two positions; the internal volume of the cushion in (a) is by definition exactly twice the internal volume in (b). The initial pressure in (a) is 0 psig, or 15 psia (psi abs). Our problem is to calculate the gauge pressure in (b). While this pressure is given in the figure, let us run through the actual calculations, using gauge pressure first, and absolute pressure second. The formula is:

$$P2 = (V_1/V_2) \times P_1$$

and since (as given above) Vi = 2 x V_2, and therefore $(V_1/V_2) = 2$, we have

$$P_2 = 2 \times P_1.$$

Using gauge pressures all the way through, we have,

$$P_2 = 2 X 0 = 0,$$

so that our calculations yield a final pressure of 0 psig. On the other hand, if we use absolute pressures in the formula we have,

$$P_2 = 2 \times 15 = 30,$$

so that our calculations yield a final pressure of 30 psia, or 15 psig. Obviously the answer is not the same in both oases; and since we know the gas laws must yield correct answers with absolute pressures, we must conclude that calculations using gauge pressures will be in error; which is what we stated above.

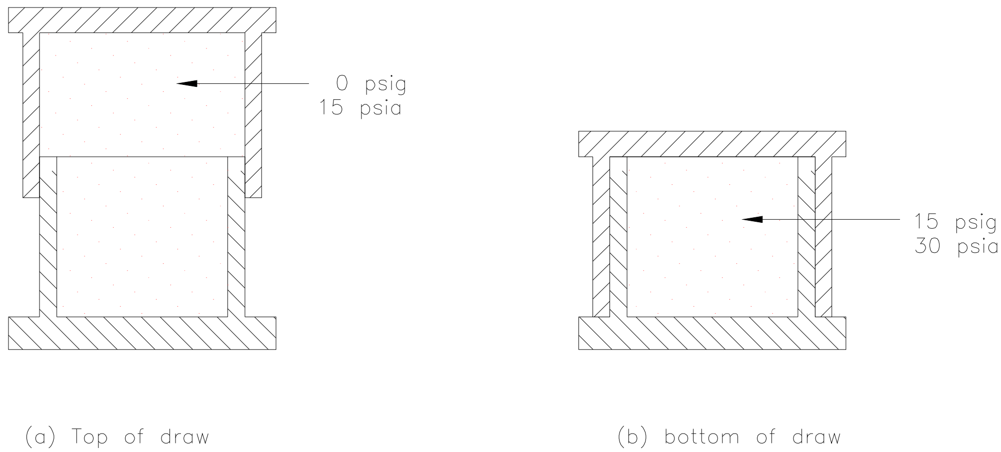


Fig. 1-2-25 Pressure build-up

This error is not usually, however, as significant as this example indicates. To put the matter in better perspective, let us perform these calculations for a more realistic situation. Let us suppose that we have a Model C-14-4 cushion, being used at a pressure of 80 psig. For this particular cushion, the volumes at the top and bottom of the draw are, respectively,

$$V_1 = 7.1 \text{ gallons, and}$$

$$V_2 = 4.4 \text{ gallons.}$$

Hence, the ratio (V_1/A_2) is 1.6, so that the formula becomes,

$$P_2 = 1.6 \times P_1.$$

Taking the value of the atmospheric pressure to be 15 psi, the initial pressure (V_1) has the values 80 psig, and 95 psia. Using gauge pressure first, we have,

$$P_2 = 1.6 \times 80 = 128 \text{ psig}$$

However, performing this calculation with absolute pressure, we have,

$$P_2 = 1.6 \times 95 = 152 \text{ psia} = 137 \text{ psig}$$

Thus, using gauge pressure, we have an error of 9 psi. This means that the actual build-up is 71%, rather than the 60% that the gauge-pressure calculations yield. Whether or not this error is appreciably significant will depend upon the application of the cushions however, we will generally assume that such errors are significant, unless we are clearly sure that they are not.

We should point out that the need to use absolute pressures in the volume-pressure formula does not change the situation with respect to the use of gauge pressures in the calculation of effective cushion force. Actually, it is the absolute pressure inside the cushion that tends to expand it, but as we mentioned in the preceding section, this is opposed by the atmospheric pressure outside the cushion, which tends to collapse it. And the result is that the 15 psi outside counterbalances the extra 15 psi inside, so that only the pressure difference is effective pressure. And this effective pressure is what we have defined as gauge pressure. Thus we use gauge pressure when calculating cushion force, and absolute pressure when determining pressure changes by means of the pressure-volume formula; and convert one to the other by adding or subtracting 15 psi as the situation warrants.

As we noted above, in order to use the pressure-volume formula, we must be able to determine initial and final volumes. For die cushions these are, respectively, the internal volumes at the top and bottom of the draw. In the general case of an arbitrary pneumatic device the calculation of volume is potentially extremely difficult, because of the many possible irregular shapes involved; but with die cushions there are only a few irregularities to contend with, and with a little thought and care the matter can usually be dispensed with fairly easily. We begin with the simple notion that volume is area times height; or expressed as a formula,

$$V = A \times H.$$

This formula can be most easily applied to a simple pneumatic cylinder, such as that illustrated in Fig. 1-2-26a. Here the internal cross-sectional area of the cylinder is uniform at all points, and we need merely multiply this times the internal height to get the internal volume. Since

$$A = \pi\,(D^2/4)\,,$$

the formula above becomes

$$V = \pi\,(D^2/4) \times H.$$

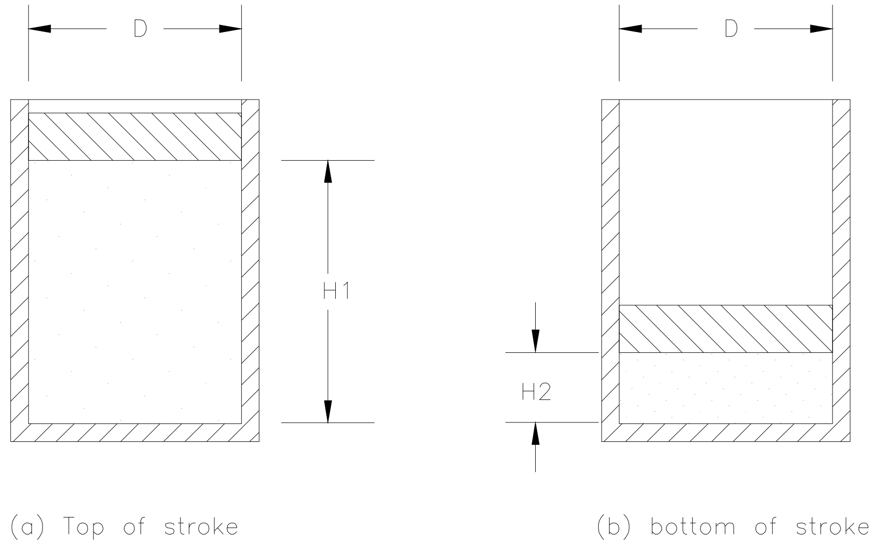


Fig. 1-2-26 Simplified pressure calculations

If we desire, we may use this formula to calculate both of the volumes required, and then substitute these into the basic formula. However, a considerable savings in effort and time can be affected if we substitute first, before making any calculations. Thus, applying the volume formula to the formula

$$P_2 = (V_1/V_2) \text{ X } P_1$$

we have

$$P_2 = (\pi\ D^2/4)\ H_1 / (\pi\ D^2/4)\ H_2 \text{ x } P_1,$$

where H_1 and H_2 are the internal heights at the top and bottom of the stroke, as given in the figure. Noting next that the common factor (π $D^2/4$) in the numerator and denominator of the right-hand expression cancels out, we may state the equation more simply as

$$P_2 = (H_1/H_2) \text{ x } P_1.$$

Hence, for this simple case we are able to substitute the internal heights of the cylinder for the respective volumes.

The question immediately arises: can we apply this procedure to standard die cushions? The answer is a qualified yes; meaning that we can apply a similar procedure if we are willing to forego a certain amount of accuracy. Depending upon the degree of accuracy required, we may use one of three methods for determining a satisfactory substitute for volume. In Fig. 1-2-27a we illustrate the least accurate approach, and label it a "first approximation". Here we simply use the actual internal height of the cushion} substituting this directly into the basic formula. Thus for a first approximation the pressure ratio is the inverse of the actual internal height ratio. However, since this introduces a sizable error, we will often find it necessary to use a more accurate approach. Such an approach must make an allowance for the volume of the piston walls (since we are using the cylinder diameter). The method of doing this is illustrated in (b). Here we first divide the internal height into two parts: $H_{P,}$ which is the internal height of the piston, and H_C, which is the remainder (or, the height of the cylinder portion of the internal cavity). The next step is to reduce the value of Hp by an amount sufficient to account for the piston-wall volume. The "second approximation" involves making a careful guess at this value, and subtracting it from H_p. The sum of this reduced value of H_p and the cylinder height (H_C) is known as the "adjusted internal height" H_a. H_a, may then be used in place of volume in the basic pressure-volume formula.

If we feel the need for a still more accurate value of H_a, we may use the method of the "third approximation". This method makes use of the fact that the piston-wall volume in terms of height (respective of the cylinder area) is given by the formula

$$(\text{Wall thickness}) \times 2\, H_P / (\text{cylinder radius}),$$

i.e., the wall thickness times twice the piston height, divided by the radius of the cylinder (internal). Noting that the cylinder radius is one-half the diameter, we may put this formula in the form:

$$4W \times (H_p/D),$$

where "W" stands for the wall thickness. Thus if we reduce the actual internal height by a value equal to four times the wall thickness (piston) times the ratio of the piston height to the cylinder diameter, we will arrive at a very accurate height value for substitution into the basic formula.

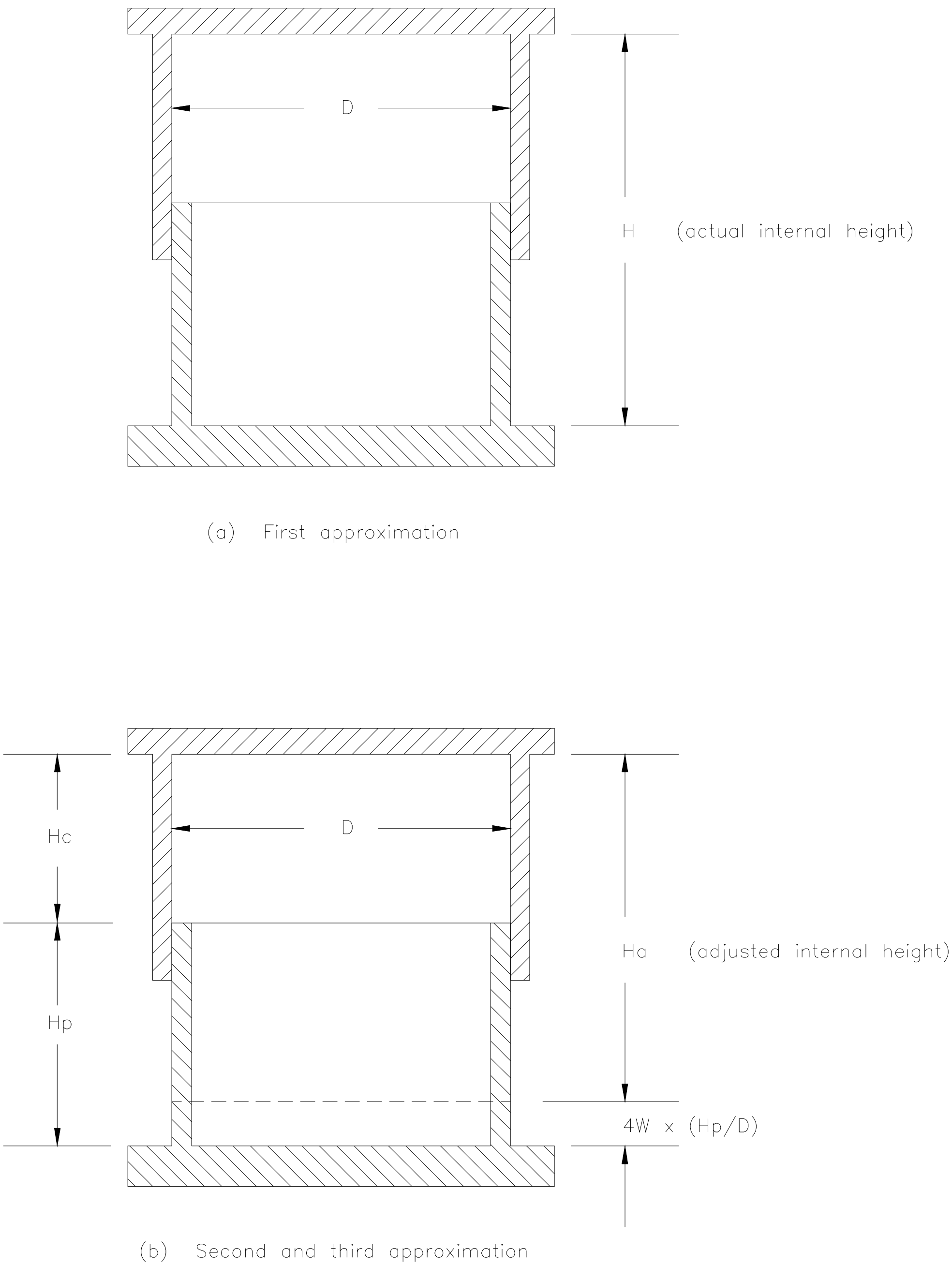


Fig. 1-2-27 Internal cushion height approximations

We note in this latter method that our approximations, in becoming more accurate, have also become more complicated; and for general use the method of the third approximation has little to offer over the direct calculation of volume. However, where we are dealing with standard products, and base most of our calculations on tabulated values or various charts and graphs, we find it much easier to deal with heights than with volumes. This will become a little clearer as we proceed. Where there are no irregularities in the internal surfaces of the piston and cylinder, the third approximation actually yields highly accurate results. However, we will often (particularly in the case of cast-iron pistons) find reinforcing ribs and other protuberances inside the cushion that cut down a little on the internal volume. Most tabulated internal heights will be reduced a little to allow for these (where they occur), so that we will seldom have to worry about them in practice.

The actual use of adjusted internal heights is quite simple, as we can see by referring to Fig. 1-2-28. Here the difference in the internal heights of the cushion at the top and bottom of the draw is given directly by the draw itself. Thus Ha is the internal height at the top, and (Ha - S) is the internal height at the bottom, S being the draw. Substituting these values for the volumes in the formula

$$P2 = V1/V2 \times P1,$$

we have,

$$P2 = Ha / (Ha - S) \times P1.$$

While this equation appears a little more involved, it is actually very easy to use; and as we noted above, yields very accurate values.

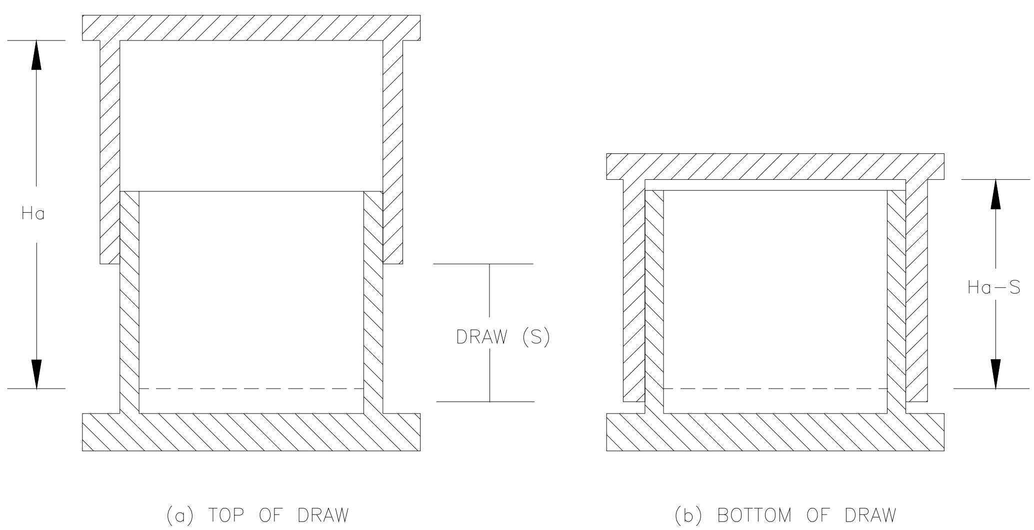


Fig. 1-2-28 Practical use of Ha

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Now that we have at our disposal a practical means of calculating the pressure at the bottom of the draw, let us consider the problem of cushion-force uniformity. It is of course obvious that the cushion force will increase as the cushion is collapsed; this we have In fact stated above. The question is: how does this affect the operation of the tooling? We should note in this context that none of the cushion applications discussed in the first chapter require the delivery of an exact force. The most critical application, drawing, requires only that the force exerted by the cushion be sufficient to prevent wrinkling of the blank, but not so much that it prevents the necessary inward sliding of material as the shell is formed around the punch. Thus we have a range of pressure from a minimum on the one hand to a maximum on the other. Our question thus becomes more specifically: how large a range of cushion-force variation (increase) is permissible. And of course: how does this compare to the actual buildup. Obviously the minimum and maximum forces actually exerted must fall between the minimum and maximum permissible forces. This means that the force at the top of the draw (which is the actual minimum) must be greater than the permissible minimum; and that the force at the bottom of the draw (which is the actual maximum) must be less than the permissible maximum.

If we recall that the cushion force at the top of the draw may be adjusted by the air-pressure regulator in the supply line to very near the permissible minimum, then the question reduces essentially to a comparison of the permissible percentage increase with the actual buildup. This buildup is the difference between the force at the top of the draw (F1) and the force at the bottom of the draw (F2); i.e.,

$$\text{buildup} = F2 - F1.$$

As this is normally expressed as a percentage of the initial force (top of draw) we have:

$$\text{percent buildup} = (F2 - F1)/F1.$$

Or, if we use the gauge values of the internal air pressure, we may state this as

$$\text{percent buildup} = (P2 - P1)/P1.$$

Satisfactory determinations of the maximum permissible buildup may be arrived at through pencil-and-paper calculations; however, this is generally very difficult, and the most practical approach is to rely on past experience. Our experience with drawing operations tells us that a buildup of about 25% is acceptable in the vast majority of cases. There are extremes in both directions, of course; 50% or more buildup is permissible for some types of work, while on the other hand we will sometimes find critical applications where buildup must be limited to 10% or less. Nevertheless, 25% makes a good rule of thumb, and is hence used as a practical maximum for all cushion installations. We should point out in this context that a given cushion installation will be used for a wide variety of work over a period of many years, and cannot (with rare exceptions) be designed for individual jobs. Thus we cannot allow the occasional extreme cases in either direction to dictate design features. The choice of 25% should not, however, mislead us into believing that an installation with a 25% buildup is of optimum design; the minimizing of buildup is desirable in all; cases, and is always the basic objective. We usually try to hold buildup to less than 20% or even to less than 15% where practical; it is only where these lower values present major problems that we invoke the 25% upper limit. The rule in this case is that buildups greater than 25% must be subject to careful engineering review; and the customer must be explicitly informed of the possible limitations this will place on the general use of the cushion.

In applying the 25% maximum to the equations developed previously, we begin with the equation

$$P2 = 1.25 \times P1,$$

but we note that in this equation the pressures are gauge pressures (since the 25% must also be the cushion force buildup). Hence, before we can use this we must convert to absolute pressures by adding 15. psi to each. Thus,

$$P2 + 15 = 1.25 \times P1 + 15.$$

Since we are restricted in the present circumstance to using both P1 and P2 as gauge pressures, the adjusted-internal-height formula given above for pressure buildup (which requires absolute pressures) must be given in the form,

$$P2 + 15 = Ha / (Ha - S) \times (P1 + 15).$$

Substituting the right member of this equation into the left of the equation immediately above, we have,

$$Ha / (Ha - S) \times (P1 + 15) = 1.25 \times P1 + 15.$$

Performing the indicated operations, this gives,

$$Ha \times P1 + 15 \times Ha = 1.25 \times Ha \times P1 + 15 \times Ha - 1.25 \times P1 \times S - 15 \times S.$$

Transposing and combining terms yields,

$$0.25 \times Ha \times P1 = 1.25 \times P1 \times S + 15 \times S.$$

At this point we can go no further until we assign a value to P1. Assuming this to be the maximum pressure rating of the cushion, 100 psig, we then have,

$$25 \times Ha = 125 \times S + 15 \times S = 140 \times S.$$

Simplifying this, we end up with,

$$Ha = 5.6 \times S,$$

or, the adjusted internal height is 5.6 times the draw. This particular derivation, as is obvious, is somewhat on the involved side, and not especially practical in itself. However, it can be greatly simplified if we but note that the final factor (i.e., 5.6) was in effect obtained by dividing the maximum absolute pressure by the difference in pressure; i.e., dividing 140 psia by 25 psia. This approach is straightforward, and fairly easy to use.

What we have derived here is a simple numerical factor relating the adjusted internal height and the draw for a particular pressure buildup. Such factors are extremely useful, and for this reason we give them a special name, total surge factor. Which we will usually abbreviate as T.S.F. The significance of this name will become apparent as we proceed. We note here that a T.S.F. is dependent basically upon the two absolute pressures involved; i.e.,

$$T.S.F. = P_2/(P_2 - P_1)$$

(as we stated above). Hence, it will vary when either one or both are changed. What this means in practical terms is that a T.S.F. is determined not only by the buildup desired, but also by the initial pressure chosen. We found above that when the initial pressure is 100 psig (115 psia) and the buildup is 25% of the gauge value, the T.S.F. is 5.6. Let us now choose another initial pressure, and calculate the T.S.F. for the same percent buildup. Let us suppose that the initial pressure is 50 psig (65 psia). In this case the T.S.F. (using the formula immediately above) is,

$$\text{T.S.F.} - 77.5 \text{ psia}/12.5 \text{ psia} = 6.2,$$

taking the pressure difference ($P_2 – P_1$) as 12.5 psi (25% of 50 psig, the initial gauge pressure). Thus it is apparent that the change in initial pressure gives a change in the total surge factor.

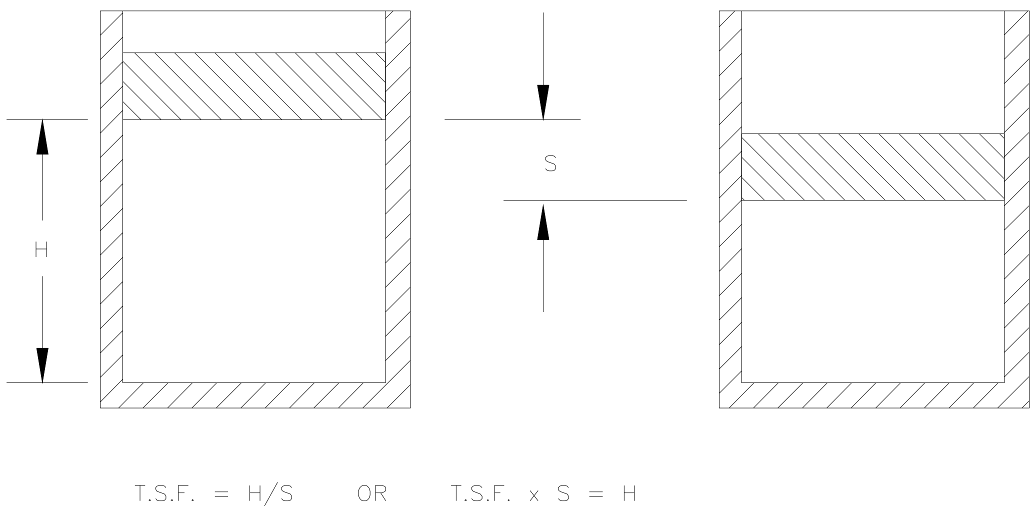

Fig. 1-2-29 Total surge factor

Before going further with the calculation of total surge factors, let us be sure that it is clear in our minds just how these factors are to be used. Referring to Fig. 1-2-29, which uses a simple piston-cylinder arrangement for illustration purposes, we see that we are dealing with two distances. One of these is the height of the internal cavity, and the other is the distance through which the piston moves. We have labeled these "H" and "S", respectively; and they are related to each other in the same way as the adjusted internal height and the draw of a die cushion. The T.S.F. is the numerical ratio between these two distance; i.e.,

$$\text{T.S.F.} = H/S,$$

or T.S.F. x S = H.

Thus, the T.S.F. enables us to find the draw when we have the height, or conversely to find the height when we have the draw. Specifically, for die cushions, we multiply the draw by_ the T.S.F. to get the adjusted internal height, or we divide the adjusted internal height by the T.S.F. to get the draw.

For more general use, total surge factors are best presented on graphs, such as we will find in bulletin 1-2-7. However, we will often find good use for a simple table such as the following:

Table 1-2-1

Initial Gauge Pressure	Final Gauge Pressure	Total Surge Factor
100 psig	115 psig	8.7
100 psig	120 psig	6.8
100 psig	125 psig	5.6
100 psig	130 psig	4.9
200 psig	230 psig	8.2
200 psig	250 psig	5.3

Factors such as these provide a basis for important system calculations, which we will now undertake to explain.

In order to keep these calculations in proper perspective, let us consider again just what it is that we are trying to achieve. Essentially, this is relatively constant cushion force during the operating cycle. Which means, as we stated above, that we want to minimize air pressure buildup as the cushion is collapsed. This buildup, in turn, is dependent upon the ratio between the displaced air and the total volume of the internal air system, which we have thus far reduced to the ratio between the cushion draw and the adjusted internal height - the T.S.F. Referring now to Fig. 1-2-30a, let us see what total surge factors tell us about standard cushions.

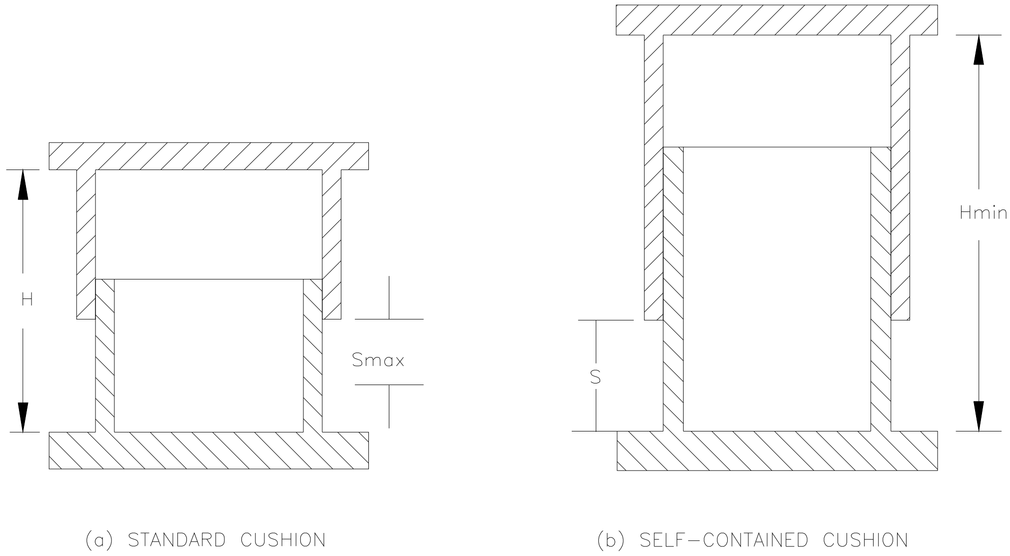

Fig. 1-2-30 Buildup limitation

For the sake of concreteness, let us assume that our T.S.F. in this case is 5.6, permitting the maximum buildup with a 100 psig initial pressure (i.e., 25%). This number, 5.6, tells us that for each inch of draw the cushion uses it must have 5.6 inches of adjusted internal height. Thus, if for example it uses 3 inches of draw, it must have an adjusted internal height of at least 16.8 inches. If it should have less than this, then it cannot use 3 inches of draw. If, for instance, the adjusted internal height is actually 14 inches, then the maximum draw that the cushion can use (and still have no more than 2% buildup) is:

$$S = H_a / \text{T.S.F.} = 14/5.6 = 2.5 \text{ inches.}$$

Since such a number is likely in most cases to be less than the available (see Section 1-2-2) draw, we have designated it in the figure as S_{max}.

The use of longer draws, without resorting to an external volume component (which we will discuss presently), can be achieved only by increasing the internal height. For example, if we needed a draw of, say, 6 inches, the adjusted internal height would have to be:

$$H_a = \text{T.S.F.} \times S = 5.6 \times 6 = 33.6 \text{ inches}$$

A cushion designed with adequate internal height to permit the use of all its available draw is usually called a self-contained cushion, such as that illustrated in Fig. 1-2-30b. The internal height in this case, since it is usually greater than that of a standard cushion of the same draw, is designated as H_{min}. The reader should compare the heights of the two cushions in this figure. While there is no specific mathematical relationship incorporated in the drawing, it is nonetheless clear that the overall height of the cushion on the right has been increased by a considerable amount over that of the cushion on the left. While the reason in this case is different, the problems that ensue are much the same as those we mentioned in Section 1-2-2 in conjunction with the overlap of cylinder and piston, and in Section 1-2-1 as part of our discussion of outside cushion dimensions. The requirements of installation design frequently necessitate the minimizing of overall cushion height; hence, the self-contained cushion does not generally represent the best solution to the use of long draws.

Since these calculations involve us with the physical dimensions of cushions, we note in passing that we should take care not to confuse actual internal height with adjusted internal height (unless of course we are using the method of the first approximation). The difference, which can amount to several inches, can introduce critical errors into an installation layout should the wrong figure be used. This problem is not particularly significant if we are merely determining pressure buildups and surge volumes for standard cushions with standard draws; but when we work in the opposite direction -that is, when we use the calculation methods discussed here as a basis for cushion design - we must be careful to add the difference between actual and adjusted heights. And of course, we might note that outside height is still greater.

Getting back to the question of long cushion draws, we can see that our basic problem, if stripped of nonessentials, is strictly one of providing sufficient volume in our air system (since the pressure-volume formula is independent of exact shape). The method illustrated in Fig. 1-2-30b, while it is the simplest and most direct, is not by any means the only way in which this can be done. In fact, the most important method of providing sufficient volume is that illustrated in

Fig. 1-2-31. Here we have used dotted lines to indicate that the additional volume provided by a self-contained cushion over a standard cushion is in this case provided instead by an externally located pressure tank, connected to the cushion by a length of pipe. Since the internal volume of a pneumatic device is commonly called its surge, the external pressure tank is usually called a surge tank, and the line (pipe) between it and the cushion is called the surge line. The total internal volume of the cushion and the surge tank should be at least as great as that required for a self-contained cushion for this particular draw.

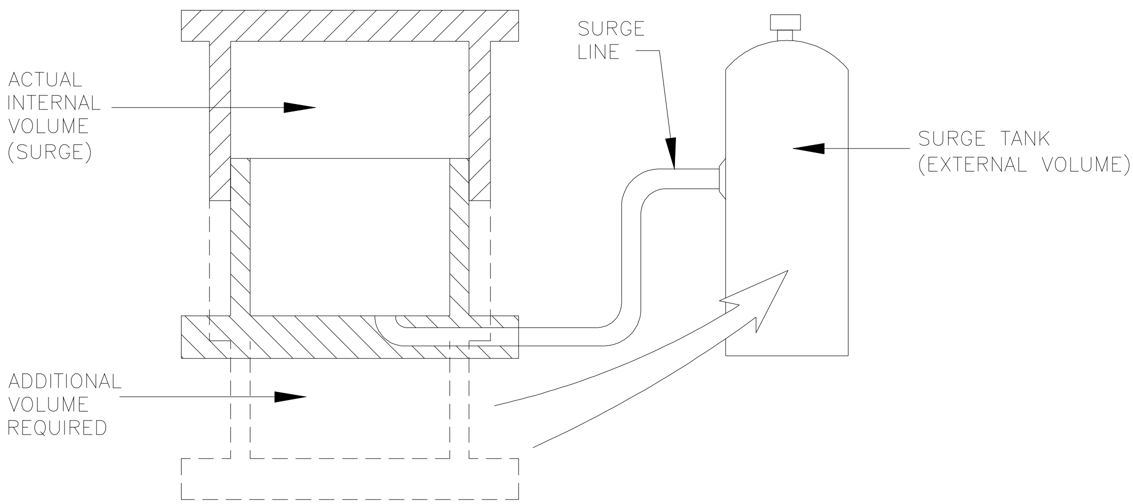

Fig. 1-2-31 External surge

Since the difference in internal volume between a standard cushion and a self-contained cushion is expressed directly as a difference in internal height, and the volume of a surge tank is normally expressed in gallons, conversion will be necessary. This is fairly simply done, and requires merely the multiplication of the difference in internal height by a volume-per-inch factor. Such factors may be calculated (being equal to the cross-sectional area of the cylinder in square inches divided by 231), or they may be obtained from tables such as the following:

Table 1-2-2

Cylinder Diameter	Gallons per Inch
6"	0.13
8"	0.22
10"	0.34
12"	0.49
14"	0.67
16"	0.87
18"	1.1
20"	1.4
22"	1.7
24."	2.0

To show how such a calculation is carried out, let us first summarize the procedure in a step-by-step form, and then make the calculation for a concrete example. The steps are:

Step 1 - Multiply the cushion draw by the total surge factor (to get the required height for a self-contained cushion).

Step 2 – Subtract the adjusted internal height from the result of step #1 (to determine how much height the standard cushions is lacking).

Step 3 – Multiply the result of step #2 by the gallons-per-inch factor (to express this height in terms of gallons).

Suppose, now, that we need to know the surge tank volume for a model C-14-6 cushion, operating on a 100 psig air line, with a maximum permissible buildup of 15% (i.e., 115 psig). The T.S.F. in this case is 8.7 (Table 1-2-1), and the gallons-pep-inch factor is 0.67 (Table 1-2-2). We proceed as follows:

Step 1 -The draw of this cushion is 6 inches, and this times the T.S.F., 8.7, gives 52.2 inches (the required self-contained cushion height).

Step 2 -The adjusted internal height of this cushion (obtained from special tables) is 14. 7 inches; and this subtracted from the result of step #1, 52.2, gives 37.5 inches (the height our standard cushion lacks).

Step 3 - The result of Step #2, 37. 5, times the gallons-par-inch factor, 0.67, gives 25.2 gallons (the gallon equivalent of the standard-cushion height deficiency).

Thus, the minimum surge tank size for this particular application is 25.2 gallons. Since there is no commercially available tank in this size, we would use the next largest size, which happens to be 33 gallons.

At this point, if we hadn't covered the material on flow phenomena in the preceding section, we might be inclined to feel that all the basic problems have been solved. However, when we consider that the introduction of an external volume component, connected to the cushion by a relatively narrow passageway (i.e., the surge line), brings up the matter of air flow and friction, we can see that there is more work to be done. Referring to Fig .1-2-32, we can see that as the ram descends, and forces the cushion cylinder down, air must flow out of the cushion, through the surge line, and into the surge tank. If the cushion were to be collapsed very slowly, there would be plenty of time for the air pressure in the cushion and in the surge tank to equalize, even with a very small surge line. However, as the speed at which the cushion is collapsed increases, the time available for pressure equalization decreases, and air must flow at increasing speeds through the surge line. Since the surge line offers resistance to this flow, which increases as the speed of the air increases, a pressure differential must exist between the cushion and the surge tank. That is, in order to move air through the surge line, say from the cushion to the surge tank, the pressure in the cushion must be greater than the pressure in the tank; and, the faster the air must flow, the greater the pressure difference must be. Thus, our surge tank size calculations, which assume equal pressure in cushion and tank, must show at least a little error. Specifically, the pressure in the cushion at the bottom of the draw will be higher than we calculated it to be, by an amount sufficient to provide the pressure differential needed to force air through the surge line.

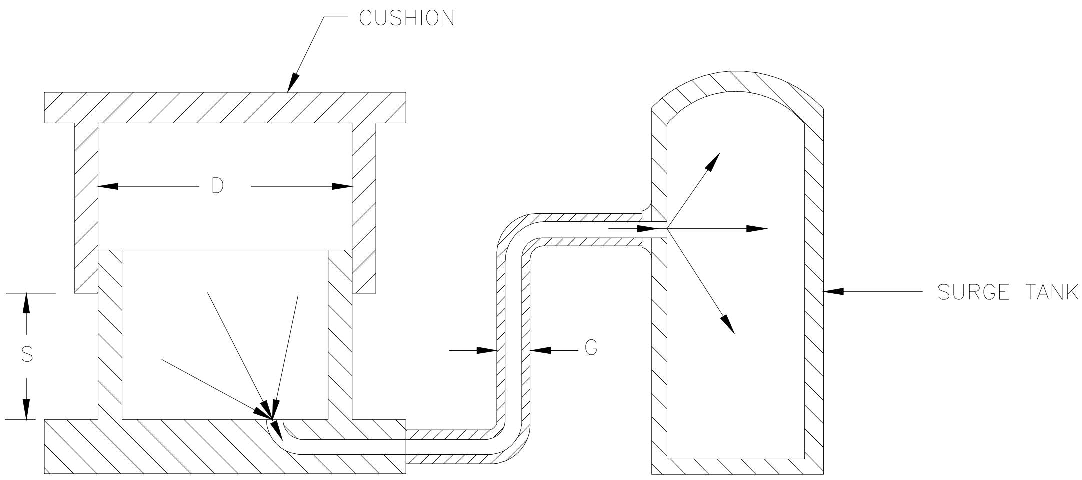

Fig. 1-2-32 Surge Line Size

Exact calculations for this three-part air system (i.e., cushion, surge line, and surge tank) are extremely difficult; and we do not wish to imply that any simplified procedure we might discuss here can supply any more than rough estimates. However, we do have available a fairly useful rule of thumb, which states that .the pressure differential between cushion and surge will be within acceptable limits if the speed of the air through the surge line does not exceed 1000 inches per second. This does not take all pertinent factors into account, but can be applied with reasonable of effect if we treat only the simplest case. To understand the nature of this simplification, let us note that the restriction of the flow of air through a line will depend upon three things:

1. The <u>size</u> of the line, being greater the smaller the size. - Specifically, this means the cross- sectional area, but we may treat it in terms of diameters.
2. The <u>length</u> of the line, being greater the greater the length.
3. The <u>straightness</u> of the line, being greater the more bends and junctions there are.

Treatment of the last two is too difficult to consider here; but if we restrict ourselves to the idealized case of a short, straight piece of pipe, then we may use the rule of thumb to determine the required size of the pipe.

Considering first the extreme case where all the air displaced by the downward movement of the cushion cylinder must be pushed through the surge line into the surge tank, we may state that the speed of flow of air in the cushion is equal to the speed of downward movement of the cylinder. Since the cross-sectional areas of the cushion cylinder and the surge line are related as the squares of their respective diameters, we may then state that:

If r is the speed of flow of air in the surge line,

R is the speed of flow of air in the cushion,

G is the internal diameter of the pipe,

and D is the internal diameter of the cushion cylinder,

then

$$r \times G^2 = R \times D^2$$

which merely states that equal volumes move past given points in pipe and cushion in unit time.

Since this equation contains four variables (i.e., r, R, G, and D), our next step is to reduce it to an equation containing three known and one unknown variable. Thus, we note the following:

1. Our rule of thumb states that r is a maximum at 1000 inches per second. Since this will yield the minimum pipe size, we may accept it as the value of r. Smaller values might be used for critical applications„
2. The cushion cylinder diameter, D, will be known to us whenever we select a particular cushion for calculation.
3. The speed of air in the cushion, R, which we have assumed to be the speed of cushion cylinder movement, depends upon the length of time for the movement, and the distance moved. The distance moved is the

cushion draw, S; which we will know from our cushion selection. The time of movement will be one-quarter of the time for a press cycle (assuming the cushion draw is selected per the standard rule of one-half the press stroke). A press cycle in seconds is: 60 divided by the strokes-per-minute rate of the press (S.P.M.). Thus, R, which is S divided by the time of movement, is:

$$R = S/(l/4 \times 60/(S.P.M)) = S \times (S.P.M.)/15.$$

This leaves only Q as the unknown, and this is the value of the pipe size that we are trying to determine.

Which means that 81 percent of the air displaced must flow through the surge line. This is compared with the 100 percent assumed in deriving the formula for pipe size.

Such factors as this will be known as tank displacement ratios, or T.D.R.'s, and may be used to correct the original formula thus:

$$\text{original formula: } r \times G^2 = R \times D^2 ;$$

$$\text{corrected formula: } r \times G^2 = R \times D^2 \times (T.D.R.) ;$$

the corrected formula simply stating that the volume of air moving past a given point in the pipe in unit time is the T.D.R. (expressed as a percent) of that moving past a given point in the cushion. When applied to the derived equation for pipe size, this correction yields:

$$G = D \times \sqrt{s} \times (S.P.M.) \times (T.D.R.)120$$

We note that this equation applies only to fairly short, straight surge lines; and if a line is long, and/or has several bends in it, the pipe will have to be larger. Calculations for this increase are, however, considerably beyond the scope of the present discussion.

In practice, the reader will probably have little occasion to use either the original or the corrected equation for pipe size, since sizes for most standard cushions are tabulated on cushion data sheets. Tabulated values should be used wherever available, since these have been proven satisfactory in actual cushion operation; and our calculated values are merely, as indicated, rough estimates.

Solutions for pipe sizes for elementary systems of one cushion, one pipe, and one surge tank, may be extended to more complicated systems, such as that illustrated in Fig. 1-2-33, through the application of two simple rules. These are:

> **Rule Number One** - The total cross-sectional area of all pipes leading out of a junction must be at least as great as the total cross-sectional area of all pipes leading into the junction. - Where a system permits flow in both directions, the direction of flow used is the most critical one. For cushion systems, this is the direction from the cushions to the surge tanks (indicated by arrows in the figure).
>
> **Rule Number Two** - The ratios of cross-sectional areas of all pipes leading in the same direction with respect to a junction (i.e., either from or to) must be the same as the ratios of the volumes to which they connect, for constant-volume containers, and the ratios of flow rates (e.g., gallons per second) for variable-volume containers.

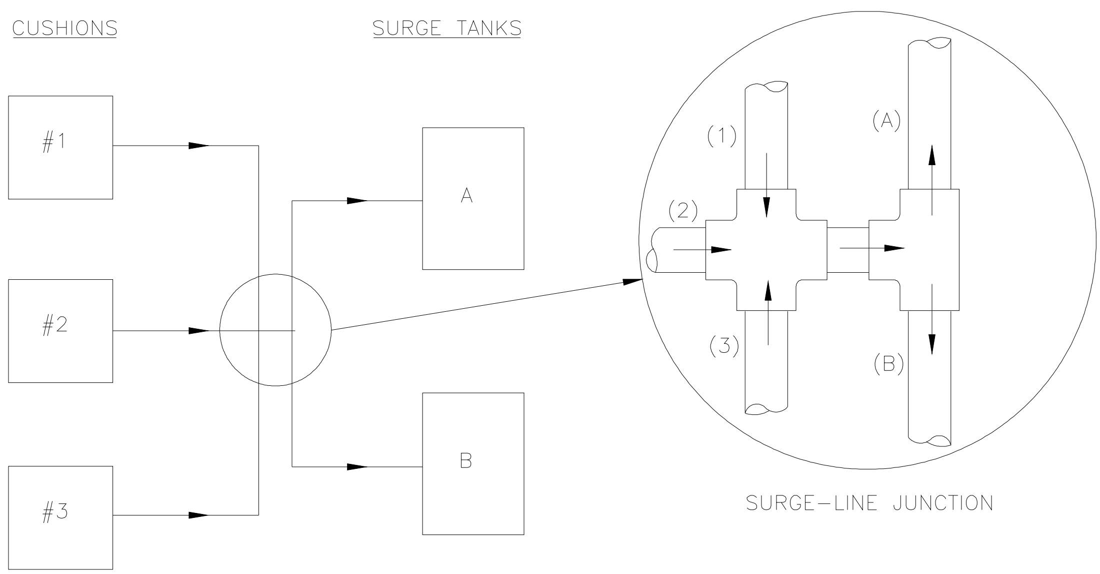

Fig. 1-2-33 Complex system

In order to see what these rules mean, and how they are to be applied, let us examine closely the two junctions in the figure.

Notice first that the left-hand junction has three pipes leading into it, and only one leading out. The right-hand junction has only one pipe leading in (the pipe from the left-hand junction), but two pipes leading out. Rule number one, applied to the left-hand junction, says that the cross-sectional area of the one pipe leading out of the junction must be at least as great as the sum of the cross-sectional areas of the three pipes leading into it. Similarly, for the right-hand junction, the sum of the cross-sectional areas of the two pipes leading out must be at least as great as the cross-sectional area of the one pipe leading in.

Applying Rule number two to the left-hand junction, the ratio of cross-sectional areas of any two of the pipes leading in, say for instance pipes #1 and #2, must be the same as the ratio of the gallons-per-second flow through each. Since a common air system assumes common air pressure, this requirement is automatically satisfied if the pipe sizes are calculated per the equation developed above. For example, if pipe #1 carried twice as much air in unit time as pipe #2, then its cross-sectional area must be twice as great. Since there is only one pipe leading out, Rule number two does not apply to it For the right-hand junction, the two pipes leading out must have cross-sectional areas in the same ratios as the volumes of the surge-tanks to which they connect. For example, if surge tank "A" is three times as large as surge tank "B", then the cross-sectional area of the pipe leading to "A" must be three times as large as that of the pipe leading to "B". Since unequal surge tank size is a fairly common occurrence, this particular application of Rule number two is an important one.

We will normally find that when two or more cushions are connected into the same surge system (such as the figure shows), they will all be of the same size and same draw; hence, the pipes from the cushions to the first junction would all be of the same size. Where different sized cushions, and/or cushions of different draws must be dealt with, the best and most common approach is to use a separate surge system for each. In fact, it is often impossible to achieve correct operation by any other means. If, however, we should require a common surge system in such a case, the simplest procedure would be as follows:

1. First calculate individually the surge tank size required for each cushion.
2. Then calculate individually, on the basis of the individual tank sizes, the pipe size required for each cushion.

3. Next, provide a total surge tank capacity (regardless of the actual number of tanks) at least as great as the sum of the individual tanks calculated in step #1.
4. Work out a piping arrangement similar to that of Fig. 1-2-33, using Rules number one and number two, as explained above.

Symbol Or Equation	Meaning Or Use	First Used On Page:	Explained On Page	See Fig:
H_a	Adjusted Internal Height	1-2-48	1-2-48	1-2-27
G.P.I.	Gallons Per Inch (conversion factor, Table 1-2-2)	1-2-56	1-2-56	---
S.P.M.	Strokes Per Minute (press)	1-2-59	---	---
T.D.R	Tank Displacement Ratio	1 -2-61	1-2-60	---
T.S.F.	Total Surge Factor (volumetric ratio)	1-2-53	1-2-53	1-2-29
$F = P \times A$; $P = F / A$ (gauge pressures)	To calculate force from pressure, or pressure from force.	1-2-38	1-2-38 thru 1-2-43	1-2-23 & 1-2-24
$V_l \times P_1 = V_2 \times P_2$; $P_2 = (V_1/V_2) \times P_1$ (absolute pressures)	**Generally:** To calculate pressure changes in terms of volume changes, or conversely (1st form). **Usually:** To calculate the maximum pressure (i.e., pressure at bottom of draw) (2nd form).	1-2-44	Previous Section: 1-2-25 & 1-2-26	1-2-15
$P_2 = (H_1/H_2) \times P_1$ (absolute pressures)	To calculate maximum pressure (at bottom of draw) - simplified version of equation immediately above (for special case only).	1-2-48	1-2-47 & 1-2-48	1-2-26
$H_a = H - (4W \times (H_p/D))$	To calculate the adjusted internal height for a simple cushion (Model "C "or equiv.).	1-2-48	---	1-2-27

$P_2 = H_a/(H_a - S) \times P_1$ (absolute pressures)	To calculate the pressure at the bottom of the draw.	1-2-50	1-2-50	1-2-28
T.S.F. = $P_2/(P_2 – P_1)$ (absolute pressures)	To calculate total surge factors.	1-2-53	1-2-52 thru 1-2-57	1-2-29 & Table: 1-2-1
S= H_a / T.S.F.; H_a = T.S.F.x S	To calculate draw from adjusted internal height, or conversely.	1-2-54	1-2-53 thru 1-2-57	1-2-30
Surge Tank Size (in gallons) = ((T.S.F. x S) – H_a) x G.P.I.	To calculate the minimum external surge tank capacity. - Given in the form of a three-step procedure in the text.	1-2-56 & 1-2-57	1-2-56 & 1-2-57	1-2-31 & Table:1-2-2
$G = \frac{Dx\sqrt{SxS.P.M.}}{120}$ $G = \frac{Dx\sqrt{SxS.P.M.xT.D.R.}}{120}$	To calculate surge line size (diameter).	1-2-60 1-2-61	1-2-57 thru 1-2-60 1-2-60 & 1-2-61	1-2-32 1-2-32